Special Topic: Reducing the Cost of Compressed Air

7-150 The total installed power of compressed air systems in the US is estimated to be about 20 million horsepower. The amount of energy and money that will be saved per year if the energy consumed by compressors is reduced by 5 percent is to be determined.

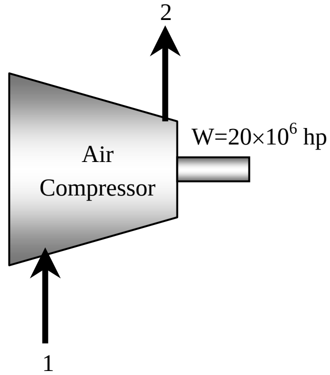
Assumptions **1** The compressors operate at full load during one-third of the time on average, and are shut down the rest of the time. **2** The average motor efficiency is 85 percent.

Analysis The electrical energy consumed by compressors per year is

Energy consumed = (Power rating)(Load factor)(Annual Operating Hours)/Motor efficiency

= $(20\times10^6$ hp)(0.746 kW/hp)(1/3)(365×24 hours/year)/0.85

= 5.125×10^{10} kWh/year



Then the energy and cost savings corresponding to a 5% reduction in energy use for compressed air become

Energy Savings = (Energy consumed)(Fraction saved)

= $(5.125\times10^{10}$ kWh)(0.05)

= **2.563×10^9 kWh/year**

Cost Savings = (Energy savings)(Unit cost of energy)

= $(2.563\times10^9$ kWh/year)($0.07/kWh)

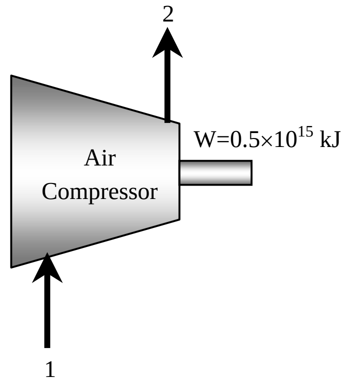
= **$\$0.179\times10^9$ /year**

Therefore, reducing the energy usage of compressors by 5% will save $179 million a year.

7-151 The total energy used to compress air in the US is estimated to be 0.5×10^{15} kJ per year. About 20% of the compressed air is estimated to be lost by air leaks. The amount and cost of electricity wasted per year due to air leaks is to be determined.

Assumptions About 20% of the compressed air is lost by air leaks.

Analysis The electrical energy and money wasted by air leaks are



Energy wasted = (Energy consumed)(Fraction wasted)

= $(0.5\times10^{15}$ kJ)(1 kWh/3600 kJ)(0.20)

= **27.78×10^9 kWh/year**

Money wasted = (Energy wasted)(Unit cost of energy)

= $(27.78\times10^9$ kWh/year)($0.07/kWh)

$$= \mathbf{\$1.945\times10^9\ /year}$$

Therefore, air leaks are costing almost $2 billion a year in electricity costs. The environment also suffers from this because of the pollution associated with the generation of this much electricity.

7-152 The compressed air requirements of a plant is being met by a 125 hp compressor that compresses air from 101.3 kPa to 900 kPa. The amount of energy and money saved by reducing the pressure setting of compressed air to 750 kPa is to be determined.

Assumptions **1** Air is an ideal gas with constant specific heats. **2** Kinetic and potential energy changes are negligible. **3** The load factor of the compressor is given to be 0.75. **4** The pressures given are absolute pressure rather than gage pressure.

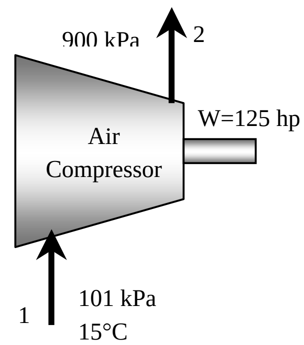
Properties The specific heat ratio of air is k = 1.4 (Table A-2).

Analysis The electrical energy consumed by this compressor per year is

Energy consumed = (Power rating)(Load factor)(Annual Operating Hours)/Motor efficiency

= (125 hp)(0.746 kW/hp)(0.75)(3500 hours/year)/0.88

= 278,160 kWh/year



The fraction of energy saved as a result of reducing the pressure setting of the compressor is

$$\text{Power Reduction Factor} = 1 - \frac{(P_{2,\text{reduced}}/P_1)^{(k-1)/k} - 1}{(P_2/P_1)^{(k-1)/k} - 1}$$

$$= 1 - \frac{(750/101.3)^{(1.4-1)/1.4} - 1}{(900/101.3)^{(1.4-1)/1.4} - 1}$$

$$= 0.1093$$

That is, reducing the pressure setting will result in about 11 percent savings from the energy consumed by the compressor and the associated cost. Therefore, the energy and cost savings in this case become

Energy Savings = (Energy consumed)(Power reduction factor)

= (278,160 kWh/year)(0.1093)

= **30,410 kWh/year**

Cost Savings = (Energy savings)(Unit cost of energy)

= (30,410 kWh/year)($0.085/kWh)

= **$2585/year**

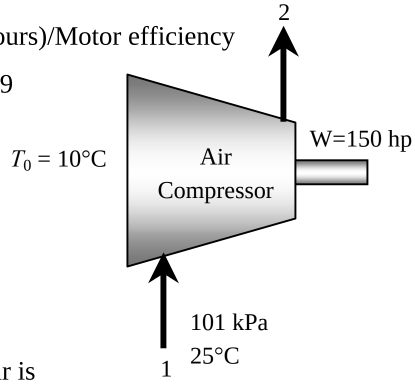
Therefore, reducing the pressure setting by 150 kPa will result in annual savings of 30.410 kWh that is worth $2585 in this case.

Discussion Some applications require very low pressure compressed air. In such cases the need can be met by a blower instead of a compressor. Considerable energy can be saved in this manner, since a blower requires a small fraction of the power needed by a compressor for a specified mass flow rate.

7-153 A 150 hp compressor in an industrial facility is housed inside the production area where the average temperature during operating hours is 25°C. The amounts of energy and money saved as a result of drawing cooler outside air to the compressor instead of using the inside air are to be determined.

Assumptions **1** Air is an ideal gas with constant specific heats. **2** Kinetic and potential energy changes are negligible.

Analysis The electrical energy consumed by this compressor per year is



Energy consumed = (Power rating)(Load factor)(Annual Operating Hours)/Motor efficiency

= (150 hp)(0.746 kW/hp)(0.85)(4500 hours/year)/0.9

= 475,384 kWh/year

Also,

Cost of Energy = (Energy consumed)(Unit cost of energy)

= (475,384 kWh/year)($0.07/kWh)

= $33,277/year

The fraction of energy saved as a result of drawing in cooler outside air is

$$\text{Power Reduction Factor} = 1 - \frac{T_{\text{outside}}}{T_{\text{inside}}} = 1 - \frac{10+273}{25+273} = 0.0503$$

That is, drawing in air which is 15°C cooler will result in 5.03 percent savings from the energy consumed by the compressor and the associated cost. Therefore, the energy and cost savings in this case become

Energy Savings = (Energy consumed)(Power reduction factor)

= (475,384 kWh/year)(0.0503)

= 23,929 kWh/year

Cost Savings = (Energy savings)(Unit cost of energy)

= (23,929 kWh/year)($0.07/kWh)

= $1675/year

Therefore, drawing air in from the outside will result in annual savings of 23,929 kWh, which is worth $1675 in this case.

Discussion The price of a typical 150 hp compressor is much lower than $50,000. Therefore, it is interesting to note that the cost of energy a compressor uses a year may be more than the cost of the compressor itself.

The implementation of this measure requires the installation of an ordinary sheet metal or PVC duct from the compressor intake to the outside. The installation cost associated with this measure is relatively low, and the pressure drop in the duct in most cases is negligible. About half of the manufacturing facilities we have visited, especially the newer ones, have the duct from the compressor intake to the outside in place, and they are already taking advantage of the savings associated with this measure.

7-154 The compressed air requirements of the facility during 60 percent of the time can be met by a 25 hp reciprocating compressor instead of the existing 100 hp compressor. The amounts of energy and money saved as a result of switching to the 25 hp compressor during 60 percent of the time are to be determined.

Analysis Noting that 1 hp = 0.746 kW, the electrical energy consumed by each compressor per year is determined from

$$\text{(Energy consumed)}_{\text{Large}} = \text{(Power)(Hours)}[(\text{LFxTF}/\eta_{\text{motor}})_{\text{Unloaded}} + (\text{LFxTF}/\eta_{\text{motor}})_{\text{Loaded}}]$$

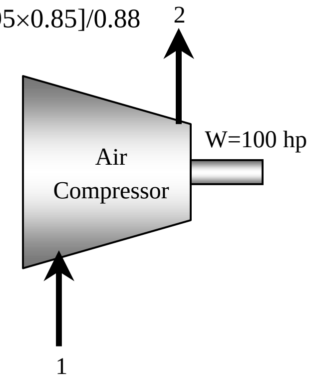
$$= (100 \text{ hp})(0.746 \text{ kW/hp})(3800 \text{ hours/year})[0.35\times0.6/0.82+0.90\times0.4/0.9]$$

$$= 185{,}990 \text{ kWh/year}$$

$$\text{(Energy consumed)}_{\text{Small}} = \text{(Power)(Hours)}[(\text{LFxTF}/\eta_{\text{motor}})_{\text{Unloaded}} + (\text{LFxTF}/\eta_{\text{motor}})_{\text{Loaded}}]$$

$$= (25 \text{ hp})(0.746 \text{ kW/hp})(3800 \text{ hours/year})[0.0\times0.15+0.95\times0.85]/0.88$$

$$= 65{,}031 \text{ kWh/year}$$



Therefore, the energy and cost savings in this case become

$$\text{Energy Savings} = \text{(Energy consumed)}_{\text{Large}} - \text{(Energy consumed)}_{\text{Small}}$$

$$= 185{,}990 - 65{,}031 \text{ kWh/year}$$

$$= \mathbf{120{,}959 \text{ kWh/year}}$$

$$\text{Cost Savings} = \text{(Energy savings)(Unit cost of energy)}$$

$$= (120{,}959 \text{ kWh/year})(\$0.075/\text{kWh})$$

$$= \mathbf{\$9{,}072/year}$$

Discussion Note that utilizing a small compressor during the times of reduced compressed air requirements and shutting down the large compressor will result in annual savings of 120,959 kWh, which is worth $9,072 in this case.

7-155 A facility stops production for one hour every day, including weekends, for lunch break, but the 125 hp compressor is kept operating. If the compressor consumes 35 percent of the rated power when idling, the amounts of energy and money saved per year as a result of turning the compressor off during lunch break are to be determined.

Analysis It seems like the compressor in this facility is kept on unnecessarily for one hour a day and thus 365 hours a year, and the idle factor is 0.35. Then the energy and cost savings associated with turning the compressor off during lunch break are determined to be

$$\text{Energy Savings} = \text{(Power Rating)(Turned Off Hours)(Idle Factor)}/\eta_{\text{motor}}$$

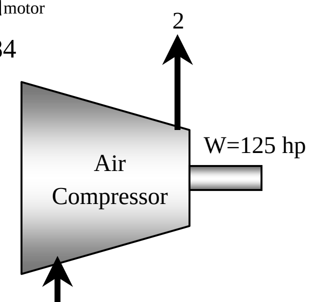
$$= (125 \text{ hp})(0.746 \text{ kW/hp})(365 \text{ hours/year})(0.35)/0.84$$

$$= \mathbf{14{,}182 \text{ kWh/year}}$$

$$\text{Cost Savings} = \text{(Energy savings)(Unit cost of energy)}$$

$$= (14{,}182 \text{ kWh/year})(\$0.09/\text{kWh})$$

$$= \mathbf{\$1{,}276/year}$$



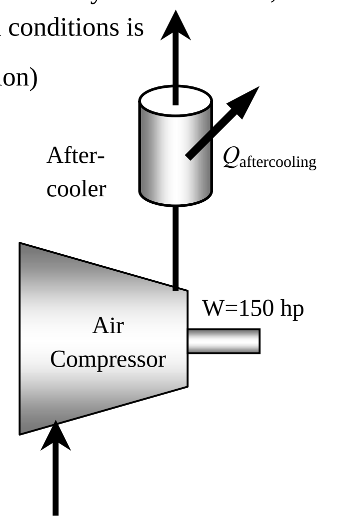
Discussion Note that the simple practice of turning the compressor off during lunch break will save this facility $1,276 a year in energy costs. There are also side benefits such as extending the life of the motor and the compressor, and reducing the maintenance costs.

7-156 It is determined that 40 percent of the energy input to the compressor is removed from the compressed air as heat in the aftercooler with a refrigeration unit whose COP is 3.5. The amounts of the energy and money saved per year as a result of cooling the compressed air before it enters the refrigerated dryer are to be determined.

Assumptions The compressor operates at full load when operating.

Analysis Noting that 40 percent of the energy input to the compressor is removed by the aftercooler, the rate of heat removal from the compressed air in the aftercooler under full load conditions is

$$\begin{aligned}\dot{Q}_{\text{aftercooling}} &= \text{(Rated Power of Compressor)(Load Factor)(Aftercooling Fraction)}\\ &= (150\text{ hp})(0.746\text{ kW/hp})(1.0)(0.4) = 44.76\text{ kW}\end{aligned}$$



The compressor is said to operate at full load for 1600 hours a year, and the COP of the refrigeration unit is 3.5. Then the energy and cost savings associated with this measure become

$$\begin{aligned}\text{Energy Savings} &= (\dot{Q}_{\text{aftercooling}})\text{(Annual Operating Hours)/COP}\\ &= (44.76\text{ kW})(1600\text{ hours/year})/3.5\\ &= \mathbf{20{,}462\ kWh/year}\end{aligned}$$

$$\begin{aligned}\text{Cost Savings} &= \text{(Energy savings)(Unit cost of energy saved)}\\ &= (20{,}462\text{ kWh/year})(\$0.06/\text{kWh})\\ &= \mathbf{\$1227/year}\end{aligned}$$

Discussion Note that the aftercooler will save this facility 20,462 kWh of electrical energy worth $1227 per year. The actual savings will be less than indicated above since we have not considered the power consumed by the fans and/or pumps of the aftercooler. However, if the heat removed by the aftercooler is utilized for some useful purpose such as space heating or process heating, then the actual savings will be much more.

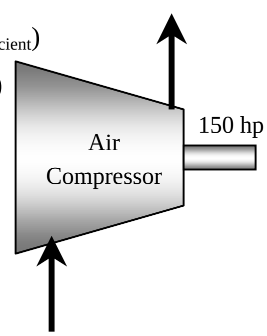
7-157 The motor of a 150 hp compressor is burned out and is to be replaced by either a 93% efficient standard motor or a 96.2% efficient high efficiency motor. It is to be determined if the savings from the high efficiency motor justify the price differential.

Assumptions **1** The compressor operates at full load when operating. **2** The life of the motors is 10 years. **3** There are no rebates involved. **4** The price of electricity remains constant.

Analysis The energy and cost savings associated with the installation of the high efficiency motor in this case are determined to be

$$\begin{aligned}\text{Energy Savings} &= \text{(Power Rating)(Operating Hours)(Load Factor)}(1/\eta_{\text{standard}} - 1/\eta_{\text{efficient}})\\ &= (150\text{ hp})(0.746\text{ kW/hp})(4{,}368\text{ hours/year})(1.0)(1/0.930 - 1/0.962)\\ &= \mathbf{17{,}483\ kWh/year}\end{aligned}$$

$$\begin{aligned}\text{Cost Savings} &= \text{(Energy savings)(Unit cost of energy)}\\ &= (17{,}483\text{ kWh/year})(\$0.075/\text{kWh})\\ &= \mathbf{\$1311/year}\end{aligned}$$



The additional cost of the energy efficient motor is

Cost Differential = \$10,942 - \$9,031 = \$1,911

Discussion The money saved by the high efficiency motor will pay for this cost difference in \$1,911/\$1311 = 1.5 years, and will continue saving the facility money for the rest of the 10 years of its lifetime. Therefore, the use of the high efficiency motor is recommended in this case even in the absence of any incentives from the local utility company.

7-158 The compressor of a facility is being cooled by air in a heat-exchanger. This air is to be used to heat the facility in winter. The amount of money that will be saved by diverting the compressor waste heat into the facility during the heating season is to be determined.

Assumptions The compressor operates at full load when operating.

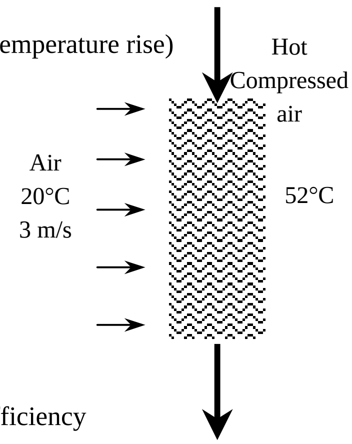
Analysis Assuming operation at sea level and taking the density of air to be 1.2 kg/m^3, the mass flow rate of air through the liquid-to-air heat exchanger is determined to be

Mass flow rate of air = (Density of air)(Average velocity)(Flow area)

= (1.2 kg/m^3)(3 m/s)(1.0 m^2)

= 3.6 kg/s = 12,960 kg/h

Noting that the temperature rise of air is 32°C, the rate at which heat can be recovered (or the rate at which heat is transferred to air) is



Rate of Heat Recovery = (Mass flow rate of air)(Specific heat of air)(Temperature rise)

= (12,960 kg/h)(1.0 kJ/kg.°C)(32°C)

= 414,720 kJ/h

The number of operating hours of this compressor during the heating season is

Operating hours = (20 hours/day)(5 days/week)(26 weeks/year)

= 2600 hours/year

Then the annual energy and cost savings become

Energy Savings = (Rate of Heat Recovery)(Annual Operating Hours)/Efficiency

= (414,720 kJ/h)(2600 hours/year)/0.8

= 1,347,840,000 kJ/year

= 12,776 therms/year

Cost Savings = (Energy savings)(Unit cost of energy saved)

= (12,776 therms/year)($1.0/therm)

= **$12,776/year**

Therefore, utilizing the waste heat from the compressor will save $12,776 per year from the heating costs.

Discussion The implementation of this measure requires the installation of an ordinary sheet metal duct from the outlet of the heat exchanger into the building. The installation cost associated with this measure is relatively low. A few of the manufacturing facilities we have visited already have this conservation system in place. A damper is used to direct the air into the building in winter and to the ambient in summer.

Combined compressor/heat-recovery systems are available in the market for both air-cooled (greater than 50 hp) and water cooled (greater than 125 hp) systems.

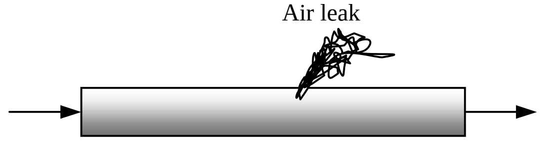
7-159 The compressed air lines in a facility are maintained at a gage pressure of 850 kPa at a location where the atmospheric pressure is 85.6 kPa. There is a 5-mm diameter hole on the compressed air line. The energy and money saved per year by sealing the hole on the compressed air line.

Assumptions **1** Air is an ideal gas with constant specific heats. **2** Kinetic and potential energy changes are negligible.

Properties The gas constant of air is R = 0.287 kJ/kg.K. The specific heat ratio of air is k = 1.4 (Table A-2).

Analysis Disregarding any pressure losses and noting that the absolute pressure is the sum of the gage pressure and the atmospheric pressure, the work needed to compress a unit mass of air at 15°C from the atmospheric pressure of 85.6 kPa to 850+85.6 = 935.6 kPa is determined to be

$$
\begin{aligned}
w_{\text{comp, in}} &= \frac{kRT_1}{\eta_{\text{comp}}(k-1)}\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k}-1\right] \\
&= \frac{(1.4)(0.287\ \text{kJ/kg.K})(288\ \text{K})}{(0.8)(1.4-1)}\left[\left(\frac{935.6\ \text{kPa}}{85.6\ \text{kPa}}\right)^{(1.4-1)/1.4}-1\right] \\
&= 354.5\ \text{kJ/kg}
\end{aligned}
$$



The cross-sectional area of the 5-mm diameter hole is

$$A = \pi D^2/4 = \pi(5\times10^{-3}\ \text{m})^2/4 = 19.63\times10^{-6}\ \text{m}^2$$

Noting that the line conditions are T_0 = 298 K and P_0 = 935.6 kPa, the mass flow rate of the air leaking through the hole is determined to be

$$
\begin{aligned}
\dot{m}_{\text{air}} &= C_{\text{loss}}\left(\frac{2}{k+1}\right)^{1/(k-1)}\frac{P_0}{RT_0}A\sqrt{kR\left(\frac{2}{k+1}\right)T_0} \\
&= (0.65)\left(\frac{2}{1.4+1}\right)^{1/(1.4-1)}\frac{935.6\ \text{kPa}}{(0.287\ \text{kPa.m}^3/\text{kg.K})(298\ \text{K})}(19.63\times10^{-6}\ \text{m}^2) \\
&\quad\times\sqrt{(1.4)(0.287\ \text{kJ/kg.K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)\left(\frac{2}{1.4+1}\right)(298\ \text{K})} \\
&= 0.02795\ \text{kg/s}
\end{aligned}
$$

Then the power wasted by the leaking compressed air becomes

$$\text{Power wasted} = \dot{m}_{\text{air}}w_{\text{comp,in}} = (0.02795\ \text{kg/s})(354.5\ \text{kJ/kg}) = 9.91\ \text{kW}$$

Noting that the compressor operates 4200 hours a year and the motor efficiency is 0.93, the annual energy and cost savings resulting from repairing this leak are determined to be

Energy Savings = (Power wasted)(Annual operating hours)/Motor efficiency
= (9.91 kW)(4200 hours/year)/0.93
= 44,755 kWh/year

Cost Savings = (Energy savings)(Unit cost of energy)
= (44,755 kWh/year)($0.07/kWh)
= $3133/year

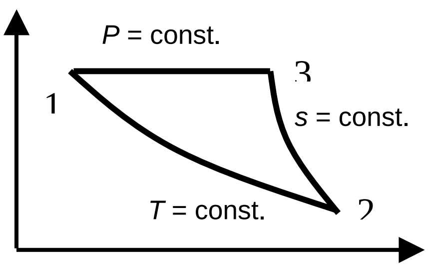
Therefore, the facility will save 44,755 kWh of electricity that is worth $3133 a year when this air leak is sealed.

Review Problems

7-160 A piston-cylinder device contains steam that undergoes a reversible thermodynamic cycle composed of three processes. The work and heat transfer for each process and for the entire cycle are to be determined.

Assumptions **1** All processes are reversible. **2** Kinetic and potential energy changes are negligible.

Analysis The properties of the steam at various states are (Tables A-4 through A-6)



$$\left.\begin{array}{l} P_1 = 400 \text{ kPa} \\ T_1 = 350°\text{C} \end{array}\right\} \begin{array}{l} u_1 = 2884.5 \text{ kJ/kg} \\ v_1 = 0.71396 \text{ m}^3\text{/kg} \\ s_1 = 7.7399 \text{ kJ/kg.K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 150 \text{ kPa} \\ T_2 = 350°\text{C} \end{array}\right\} \begin{array}{l} u_2 = 2888.0 \text{ kJ/kg} \\ s_2 = 8.1983 \text{ kJ/kg.K} \end{array}$$

$$\left.\begin{array}{l} P_3 = 400 \text{ kPa} \\ s_3 = s_2 = 8.1983 \text{ kJ/kg.K} \end{array}\right\} \begin{array}{l} u_3 = 3132.9 \text{ kJ/kg} \\ v_3 = 0.89148 \text{ m}^3\text{/kg} \end{array}$$

The mass of the steam in the cylinder and the volume at state 3 are

$$m = \frac{V_1}{v_1} = \frac{0.3 \text{ m}^3}{0.71396 \text{ m}^3\text{/kg}} = 0.4202 \text{ kg}$$

$$V_3 = mV_3 = (0.4202 \text{ kg})(0.89148 \text{ m}^3\text{/kg}) = 0.3746 \text{ m}^3$$

Process 1-2: Isothermal expansion ($T_2 = T_1$)

$$\Delta S_{1-2} = m(s_2 - s_1) = (0.4202 \text{ kg})(8.1983 - 7.7399)\text{kJ/kg.K} = 0.1926 \text{ kJ/kg.K}$$

$$Q_{\text{in},1-2} = T_1 \Delta S_{1-2} = (350 + 273 \text{ K})(0.1926 \text{ kJ/K}) = 120 \text{ kJ}$$

$$W_{\text{out},1-2} = Q_{\text{in},1-2} - m(u_2 - u_1) = 120 \text{ kJ} - (0.4202 \text{ kg})(2888.0 - 2884.5)\text{kJ/kg} = 118.5 \text{ kJ}$$

Process 2-3: Isentropic (reversible-adiabatic) compression ($s_3 = s_2$)

$$W_{\text{in},2-3} = m(u_3 - u_2) = (0.4202 \text{ kg})(3132.9 - 2888.0)\text{kJ/kg} = 102.9 \text{ kJ}$$

$$Q_{2\text{-}3} = 0 \text{ kJ}$$

Process 3-1: Constant pressure compression process ($P_1 = P_3$)

$$W_{\text{in},3-1} = P_3(V_3 - V_1) = (400 \text{ kPa})(0.3746 - 0.3) = 29.8 \text{ kJ}$$

$$Q_{\text{out},3-1} = W_{\text{in},3-1} - m(u_1 - u_3) = 29.8 \text{ kJ} - (0.4202 \text{ kg})(2884.5 - 3132.9) = 134.2 \text{ kJ}$$

The net work and net heat transfer are

$$W_{\text{net,in}} = W_{\text{in},3-1} + W_{\text{in},2-3} - W_{\text{out},1-2} = 29.8 + 102.9 - 118.5 = \mathbf{14.2 \text{ kJ}}$$

$$Q_{\text{net,in}} = Q_{\text{in},1-2} - Q_{\text{out},3-1} = 120 - 134.2 = -14.2 \text{ kJ} \longrightarrow Q_{\text{net,out}} = \mathbf{14.2 \text{ kJ}}$$

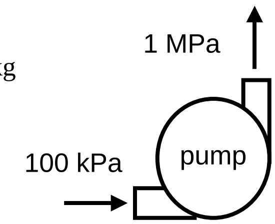
Discussion The results are not surprising since for a cycle, the net work and heat transfers must be equal to each other.

7-161 The work input and the entropy generation are to be determined for the compression of saturated liquid water in a pump and that of saturated vapor in a compressor.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer to or from the fluid is zero.

Analysis Pump Analysis: (Properties are obtained from EES)

$$\left.\begin{array}{l} P_1 = 100 \text{ kPa} \\ x_1 = 0 \text{ (sat. liq.)} \end{array}\right\} \begin{array}{l} h_1 = 417.51 \text{ kJ/kg} \\ s_1 = 1.3028 \text{ kJ/kg.K} \end{array} \qquad \left.\begin{array}{l} P_2 = 1 \text{ MPa} \\ s_2 = s_1 \end{array}\right\} h_{2s} = 418.45 \text{ kJ/kg}$$



$$h_2 = h_1 + \frac{h_{2s} - h_1}{\eta_P} = 417.51 + \frac{418.45 - 417.51}{0.85} = 418.61 \text{ kJ/kg}$$

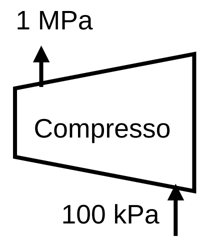
$$\left.\begin{array}{l} P_2 = 1 \text{ MPa} \\ h_2 = 418.61 \text{ kJ/kg} \end{array}\right\} s_2 = 1.3032 \text{ kJ/kg.K}$$

$$w_P = h_2 - h_1 = 418.61 - 417.51 = \mathbf{1.10\ kJ/kg}$$

$$s_{gen,P} = s_2 - s_1 = 1.3032 - 1.3028 = \mathbf{0.0004\ kJ/kg.K}$$

Compressor Analysis:

$$\left.\begin{array}{l} P_1 = 100 \text{ kPa} \\ x_1 = 1 \text{ (sat. vap.)} \end{array}\right\} \begin{array}{l} h_1 = 2675.0 \text{ kJ/kg} \\ s_1 = 7.3589 \text{ kJ/kg.K} \end{array} \qquad \left.\begin{array}{l} P_2 = 1 \text{ MPa} \\ s_2 = s_1 \end{array}\right\} h_{2s} = 3193.6 \text{ kJ/kg}$$



$$h_2 = h_1 + \frac{h_{2s} - h_1}{\eta_C} = 2675.0 + \frac{3193.6 - 2675.0}{0.85} = 3285.1 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 1 \text{ MPa} \\ h_2 = 3285.1 \text{ kJ/kg} \end{array}\right\} s_2 = 7.4974 \text{ kJ/kg.K}$$

$$w_C = h_2 - h_1 = 3285.1 - 2675.0 = \mathbf{610.1\ kJ/kg}$$

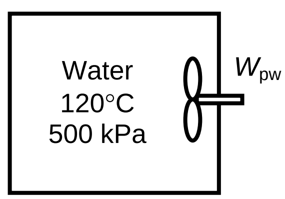
$$s_{gen,C} = s_2 - s_1 = 7.4974 - 7.3589 = \mathbf{0.1384\ kJ/kg.K}$$

7-162 A paddle wheel does work on the water contained in a rigid tank. For a zero entropy change of water, the final pressure in the tank, the amount of heat transfer between the tank and the surroundings, and the entropy generation during the process are to be determined.

Assumptions The tank is stationary and the kinetic and potential energy changes are negligible.

Analysis (*a*) Using saturated liquid properties for the compressed liquid at the initial state (Table A-4)

$$\left.\begin{array}{l} T_1 = 120°\text{C} \\ x_1 = 0 \text{ (sat. liq.)} \end{array}\right\} \begin{array}{l} u_1 = 503.60 \text{ kJ/kg} \\ s_1 = 1.5279 \text{ kJ/kg.K} \end{array}$$



The entropy change of water is zero, and thus at the final state we have

$$\left.\begin{array}{l} T_2 = 95°\text{C} \\ s_2 = s_1 = 1.5279 \text{ kJ/kg.K} \end{array}\right\} \begin{array}{l} P_2 = \mathbf{84.6\ kPa} \\ u_2 = 492.63 \text{ kJ/kg} \end{array}$$

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

$$Q_{out} = W_{Pw,in} - m(u_2 - u_1) = 22 \text{ kJ} - (1.5 \text{ kg})(492.63 - 503.60)\text{kJ/kg} = \mathbf{38.5\ kJ}$$

(*c*) Since the entropy change of water is zero, the entropy generation is only due to the entropy increase of the surroundings, which is determined from

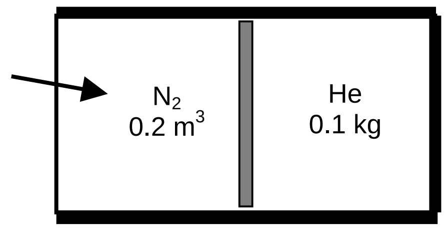
$$S_{\text{gen}} = \Delta S_{\text{surr}} = \frac{Q_{\text{out}}}{T_{\text{surr}}} = \frac{38.5\ \text{kJ}}{(15+273)\ \text{K}} = \mathbf{0.134\ kJ/K}$$

7-163 A horizontal cylinder is separated into two compartments by a piston, one side containing nitrogen and the other side containing helium. Heat is added to the nitrogen side. The final temperature of the helium, the final volume of the nitrogen, the heat transferred to the nitrogen, and the entropy generation during this process are to be determined.

Assumptions **1** Kinetic and potential energy changes are negligible. **2** Nitrogen and helium are ideal gases with constant specific heats at room temperature. **3** The piston is adiabatic and frictionless.

Properties The properties of nitrogen at room temperature are R = 0.2968 kPa.m^3/kg.K, c_p = 1.039 kJ/kg.K, c_v = 0.743 kJ/kg.K, k = 1.4. The properties for helium are R = 2.0769 kPa.m^3/kg.K, c_p = 5.1926 kJ/kg.K, c_v = 3.1156 kJ/kg.K, k = 1.667 (Table A-2).

Analysis (a) Helium undergoes an isentropic compression process, and thus the final helium temperature is determined from



$$T_{\mathrm{He},2} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (20+273)\mathrm{K}\left(\frac{120\ \mathrm{kPa}}{95\ \mathrm{kPa}}\right)^{(1.667-1)/1.667}$$
$$= \mathbf{321.7\ K}$$

(b) The initial and final volumes of the helium are

$$V_{\mathrm{He},1} = \frac{mRT_1}{P_1} = \frac{(0.1\ \mathrm{kg})(2.0769\ \mathrm{kPa\cdot m^3/kg\cdot K})(20+273\ \mathrm{K})}{95\ \mathrm{kPa}} = 0.6406\ \mathrm{m}^3$$

$$V_{\mathrm{He},2} = \frac{mRT_2}{P_2} = \frac{(0.1\ \mathrm{kg})(2.0769\ \mathrm{kPa\cdot m^3/kg\cdot K})(321.7\ \mathrm{K})}{120\ \mathrm{kPa}} = 0.5568\ \mathrm{m}^3$$

Then, the final volume of nitrogen becomes

$$V_{\mathrm{N2},2} = V_{\mathrm{N2},1} + V_{\mathrm{He},1} - V_{\mathrm{He},2} = 0.2 + 0.6406 - 0.5568 = \mathbf{0.2838\ m^3}$$

(c) The mass and final temperature of nitrogen are

$$m_{\mathrm{N2}} = \frac{P_1V_1}{RT_1} = \frac{(95\ \mathrm{kPa})(0.2\ \mathrm{m}^3)}{(0.2968\ \mathrm{kPa\cdot m^3/kg\cdot K})(20+273\ \mathrm{K})} = 0.2185\ \mathrm{kg}$$

$$T_{\mathrm{N2},2} = \frac{P_2V_2}{mR} = \frac{(120\ \mathrm{kPa})(0.2838\ \mathrm{m}^3)}{(0.2185\ \mathrm{kg})(0.2968\ \mathrm{kPa\cdot m^3/kg\cdot K})} = 525.1\ \mathrm{K}$$

The heat transferred to the nitrogen is determined from an energy balance

$$\begin{aligned} Q_{\mathrm{in}} &= \Delta U_{\mathrm{N2}} + \Delta U_{\mathrm{He}} \\ &= \left[mc_v(T_2-T_1)\right]_{\mathrm{N2}} + \left[mc_v(T_2-T_1)\right]_{\mathrm{He}} \\ &= (0.2185\ \mathrm{kg})(0.743\ \mathrm{kJ/kg.K})(525.1-293) + (0.1\ \mathrm{kg})(3.1156\ \mathrm{kJ/kg.K})(321.7-293) \\ &= \mathbf{46.6\ kJ} \end{aligned}$$

(d) Noting that helium undergoes an isentropic process, the entropy generation is determined to be

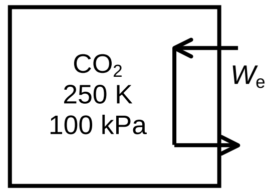
$$\begin{aligned} S_{\mathrm{gen}} &= \Delta S_{\mathrm{N2}} + \Delta S_{\mathrm{surr}} = m_{\mathrm{N2}}\left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}\right) + \frac{-Q_{\mathrm{in}}}{T_{\mathrm{R}}} \\ &= (0.2185\ \mathrm{kg})\left[(1.039\ \mathrm{kJ/kg.K})\ln\frac{525.1\ \mathrm{K}}{293\ \mathrm{K}} - (0.2968\ \mathrm{kJ/kg.K})\ln\frac{120\ \mathrm{kPa}}{95\ \mathrm{kPa}}\right] + \frac{-46.6\ \mathrm{kJ}}{(500+273)\ \mathrm{K}} \\ &= \mathbf{0.057\ kJ/K} \end{aligned}$$

7-164 An electric resistance heater is doing work on carbon dioxide contained an a rigid tank. The final temperature in the tank, the amount of heat transfer, and the entropy generation are to be determined.

Assumptions **1** Kinetic and potential energy changes are negligible. **2** Carbon dioxide is ideal gas with constant specific heats at room temperature.

Properties The properties of CO_2 at an anticipated average temperature of 350 K are R = 0.1889 kPa.m^3/kg.K, c_p = 0.895 kJ/kg.K, c_v = 0.706 kJ/kg.K (Table A-2b).

Analysis (*a*) The mass and the final temperature of CO_2 may be determined from ideal gas equation



$$m = \frac{P_1 \boldsymbol{V}}{RT_1} = \frac{(100\ \text{kPa})(0.8\ \text{m}^3)}{(0.1889\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(250\ \text{K})} = 1.694\ \text{kg}$$

$$T_2 = \frac{P_2 \boldsymbol{V}}{mR} = \frac{(175\ \text{kPa})(0.8\ \text{m}^3)}{(1.694\ \text{kg})(0.1889\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})} = \mathbf{437.5\ K}$$

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

$$\begin{aligned} Q_\text{out} &= \dot{E}_\text{e,in}\Delta t - mc_v(T_2 - T_1) \\ &= (0.5\ \text{kW})(40\times 60\ \text{s}) - (1.694\ \text{kg})(0.706\ \text{kJ/kg.K})(437.5 - 250)\text{K} = \mathbf{975.8\ \ kJ} \end{aligned}$$

(*c*) The entropy generation associated with this process may be obtained by calculating total entropy change, which is the sum of the entropy changes of CO_2 and the surroundings

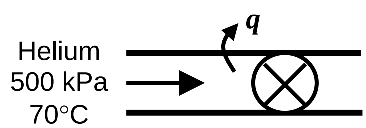
$$\begin{aligned} S_\text{gen} &= \Delta S_{CO2} + \Delta S_\text{surr} = m\left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}\right) + \frac{Q_\text{out}}{T_\text{surr}} \\ &= (1.694\ \text{kg})\left[(0.895\ \text{kJ/kg.K})\ln\frac{437.5\ \text{K}}{250\ \text{K}} - (0.1889\ \text{kJ/kg.K})\ln\frac{175\ \text{kPa}}{100\ \text{kPa}}\right] + \frac{975.8\ \text{kJ}}{300\ \text{K}} \\ &= \mathbf{3.92\ kJ/K} \end{aligned}$$

7-165 Heat is lost from the helium as it is throttled in a throttling valve. The exit pressure and temperature of helium and the entropy generation are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Helium is an ideal gas with constant specific heats.

Properties The properties of helium are R = 2.0769 kPa.m^3/kg.K, c_p = 5.1926 kJ/kg.K (Table A-2a).

Analysis (*a*) The final temperature of helium may be determined from an energy balance on the control volume



$$q_\text{out} = c_p(T_1 - T_2) \longrightarrow T_2 = T_1 - \frac{q_\text{out}}{c_p} = 70°\text{C} - \frac{2.5\ \text{kJ/kg}}{5.1926\ \text{kJ/kg.°C}} = 342.5\ \text{K} = \mathbf{69.5°C}$$

The final pressure may be determined from the relation for the entropy change of helium

$$\Delta s_\text{He} = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}$$

$$0.25\ \text{kJ/kg.K} = (5.1926\ \text{kJ/kg.K})\ln\frac{342.5\ \text{K}}{343\ \text{K}} - (2.0769\ \text{kJ/kg.K})\ln\frac{P_2}{500\ \text{kPa}}$$

$$P_2 = \mathbf{441.7\ kPa}$$

(*b*) The entropy generation associated with this process may be obtained by adding the entropy change of helium as it flows in the valve and the entropy change of the surroundings

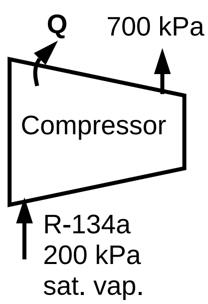
$$s_{\text{gen}} = \Delta s_{\text{He}} + \Delta s_{\text{surr}} = \Delta s_{\text{He}} + \frac{q_{\text{out}}}{T_{\text{surr}}} = 0.25\,\text{kJ/kg.K} + \frac{2.5\,\text{kJ/kg}}{(25+273)\,\text{K}} = \mathbf{0.258\,kJ/kg.K}$$

7-166 Refrigerant-134a is compressed in a compressor. The rate of heat loss from the compressor, the exit temperature of R-134a, and the rate of entropy generation 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 at the inlet of the compressor are (Table A-12)

$$\left.\begin{array}{l} P_1 = 200\text{ kPa} \\ x_1 = 1 \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.09987\text{ m}^3/\text{kg} \\ h_1 = 244.46\text{ kJ/kg} \\ s_1 = 0.93773\text{ kJ/kg.K} \end{array}$$



The mass flow rate of the refrigerant is

$$\dot{m} = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{0.03\text{ m}^3/\text{s}}{0.09987\text{ m}^3/\text{kg}} = 0.3004\text{ kg/s}$$

Given the entropy increase of the surroundings, the heat lost from the compressor is

$$\Delta\dot{S}_{\text{surr}} = \frac{\dot{Q}_{\text{out}}}{T_{\text{surr}}} \longrightarrow \dot{Q}_{\text{out}} = T_{\text{surr}}\Delta\dot{S}_{\text{surr}} = (20 + 273\text{ K})(0.008\text{ kW/K}) = \mathbf{2.344\ kW}$$

(*b*) An energy balance on the compressor gives

$$\dot{W}_{\text{in}} - \dot{Q}_{\text{out}} = \dot{m}(h_2 - h_1)$$

$$10\text{ kW - }2.344\text{ kW} = (0.3004\text{ kg/s})(h_2\text{ - }244.46)\text{ kJ/kg} \longrightarrow h_2 = 269.94\text{ kJ/kg}$$

The exit state is now fixed. Then,

$$\left.\begin{array}{l} P_2 = 700\text{ kPa} \\ h_2 = 269.94\text{ kJ/kg} \end{array}\right\} \begin{array}{l} T_2 = \mathbf{31.5^\circ C} \\ s_2 = 0.93620\text{ kJ/kg.K} \end{array}$$

(*c*) The entropy generation associated with this process may be obtained by adding the entropy change of R-134a as it flows in the compressor and the entropy change of the surroundings

$$\begin{aligned} \dot{S}_{\text{gen}} &= \Delta\dot{S}_{\text{R}} + \Delta\dot{S}_{\text{surr}} = \dot{m}(s_2 - s_1) + \Delta\dot{S}_{\text{surr}} \\ &= (0.3004\text{ kg/s})(0.93620\text{ - }0.93773)\text{ kJ/kg.K} + 0.008\text{ kW/K} \\ &= \mathbf{0.00754\ kJ/K} \end{aligned}$$

7-167 Air flows in an adiabatic nozzle. The isentropic efficiency, the exit velocity, and the entropy generation are to be determined.

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

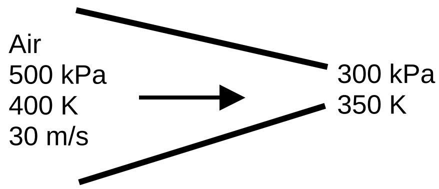
Assumptions **1** Steady operating conditions exist. **2** Potential energy changes are negligible.

Analysis (a) (b) Using variable specific heats, the properties can be determined from air table as follows

$$T_1 = 400\text{ K} \longrightarrow \begin{array}{l} h_1 = 400.98\text{ kJ/kg} \\ s_1^0 = 1.99194\text{ kJ/kg.K} \\ P_{r1} = 3.806 \end{array}$$

$$T_2 = 350\text{ K} \longrightarrow \begin{array}{l} h_2 = 350.49\text{ kJ/kg} \\ s_2^0 = 1.85708\text{ kJ/kg.K} \end{array}$$

$$P_{r2} = \frac{P_2}{P_1}P_{r1} = \frac{300\text{ kPa}}{500\text{ kPa}}(3.806) = 2.2836 \longrightarrow h_{2s} = 346.31\text{ kJ/kg}$$



Energy balances on the control volume for the actual and isentropic processes give

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2}$$

$$400.98\text{ kJ/kg} + \frac{(30\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 350.49\text{ kJ/kg} + \frac{V_2^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right)$$

$$V_2 = \mathbf{319.1\,m/s}$$

$$h_1 + \frac{V_1^2}{2} = h_{2s} + \frac{V_{2s}^2}{2}$$

$$400.98\text{ kJ/kg} + \frac{(30\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 346.31\text{ kJ/kg} + \frac{V_{2s}^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right)$$

$$V_{2s} = 331.8\text{ m/s}$$

The isentropic efficiency is determined from its definition,

$$\eta_N = \frac{V_2^2}{V_{2s}^2} = \frac{(319.1\text{ m/s})^2}{(331.8\text{ m/s})^2} = \mathbf{0.925}$$

(c) Since the nozzle is adiabatic, the entropy generation is equal to the entropy increase of the air as it flows in the nozzle

$$s_{gen} = \Delta s_{air} = s_2^0 - s_1^0 - R\ln\frac{P_2}{P_1}$$

$$= (1.85708 - 1.99194)\text{kJ/kg.K} - (0.287\text{ kJ/kg.K})\ln\frac{300\text{ kPa}}{500\text{ kPa}} = \mathbf{0.0118\,kJ/kg.K}$$

7-168 It is to be shown that the difference between the steady-flow and boundary works is the flow energy.

Analysis The total differential of flow energy $P\boldsymbol{v}$ can be expressed as

$$d(P\boldsymbol{v}) = Pd\boldsymbol{v} + \boldsymbol{v}dP = \delta w_b - \delta w_{flow} = \delta(w_b - w_{flow})$$

Therefore, the difference between the reversible steady-flow work and the reversible boundary work is the flow energy.

7-169 An insulated rigid tank is connected to a piston-cylinder device with zero clearance that is maintained at constant pressure. A valve is opened, and some steam in the tank is allowed to flow into the cylinder. The final temperatures in the tank and the cylinder are to be determined.

Assumptions **1** Both the tank and cylinder are well-insulated and thus heat transfer is negligible. **2** The water that remains in the tank underwent a reversible adiabatic process. **3** The thermal energy stored in the tank and cylinder themselves is negligible. **4** The system is stationary and thus kinetic and potential energy changes are negligible.

Analysis (a) The steam in tank A undergoes a reversible, adiabatic process, and thus $s_2 = s_1$. From the steam tables (Tables A-4 through A-6),

$$\left.\begin{array}{l} P_1 = 500\text{ kPa} \\ sat.vapor \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = v_{g@500\text{ kPa}} = 0.37483\text{ m}^3/\text{kg} \\ u_1 = u_{g@500\text{ kPa}} = 2560.7\text{ kJ/kg} \\ s_1 = s_{g@500\text{ kPa}} = 6.8207\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$T_{2,A} = T_{sat@150\text{ kPa}} = \mathbf{111.35°C}$$

$$\left.\begin{array}{l} P_2 = 150\text{ kPa} \\ s_2 = s_1 \\ (sat.mixture) \end{array}\right\} \begin{array}{l} x_{2,A} = \dfrac{s_{2,A} - s_f}{s_{fg}} = \dfrac{6.8207 - 1.4337}{5.7894} = 0.9305 \\ \boldsymbol{v}_{2,A} = \boldsymbol{v}_f + x_{2,A}\boldsymbol{v}_{fg} = 0.001053 + (0.9305)(1.1594 - 0.001053) = 1.0789\text{ m}^3/\text{kg} \\ u_{2,A} = u_f + x_{2,A}u_{fg} = 466.97 + (0.9305)(2052.3\text{ kJ/kg}) = 2376.6\text{ kJ/kg} \end{array}$$

The initial and the final masses in tank A are

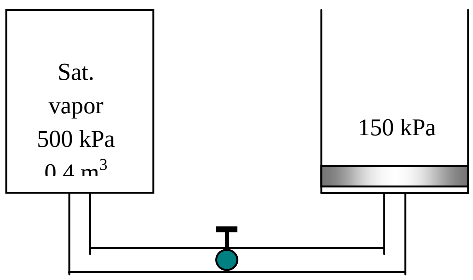
$$m_{1,A} = \frac{\boldsymbol{V}_A}{\boldsymbol{v}_{1,A}} = \frac{0.4\text{ m}^3}{0.37483\text{ m}^3/\text{kg}} = 1.067\text{ kg} \quad \text{and} \quad m_{2,A} = \frac{\boldsymbol{V}_A}{\boldsymbol{v}_{2,A}} = \frac{0.4\text{ m}^3}{1.0789\text{ m}^3/\text{kg}} = 0.371\text{ kg}$$

Thus,

$$m_{2,B} = m_{1,A} - m_{2,A} = 1.067 - 0.371 = 0.696\text{ kg}$$

(*b*) The boundary work done during this process is

$$W_{b,out} = \int_1^2 P d\boldsymbol{V} = P_B(\boldsymbol{V}_{2,B} - 0) = P_B m_{2,B}\boldsymbol{v}_{2,B}$$



Taking the contents of both the tank and the cylinder to be the system, the energy balance for this closed system can be expressed as

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

$$-W_{\text{b,out}} = \Delta U = (\Delta U)_A + (\Delta U)_B$$

$$W_{\text{b,out}} + (\Delta U)_A + (\Delta U)_B = 0$$

or,
$$P_B m_{2,B}\boldsymbol{v}_{2,B} + (m_2u_2 - m_1u_1)_A + (m_2u_2)_B = 0$$
$$m_{2,B}h_{2,B} + (m_2u_2 - m_1u_1)_A = 0$$

Thus,

$$h_{2,B} = \frac{(m_1u_1 - m_2u_2)_A}{m_{2,B}} = \frac{(1.067)(2560.7) - (0.371)(2376.6)}{0.696} = 2658.8\text{ kJ/kg}$$

At 150 kPa, h_f = 467.13 and h_g = 2693.1 kJ/kg. Thus at the final state, the cylinder will contain a saturated liquid-vapor mixture since $h_f < h_2 < h_g$. Therefore,

$T_{2,B} = T_{\text{sat@150 kPa}} = \mathbf{111.35°C}$

7-170 One ton of liquid water at 80°C is brought into a room. The final equilibrium temperature in the room and the entropy change during this process are to be determined.

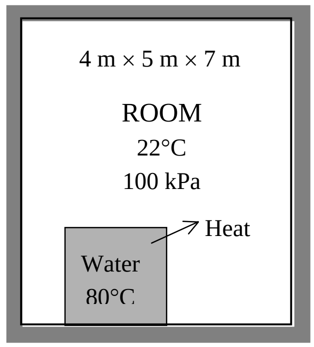
Assumptions **1** The room is well insulated and well sealed. **2** The thermal properties of water and air are constant at room temperature. **3** The system is stationary and thus the kinetic and potential energy changes are zero. **4** There are no work interactions involved.

Properties The gas constant of air is R = 0.287 kPa.m^3/kg.K (Table A-1). The specific heat of water at room temperature is c = 4.18 kJ/kg·°C (Table A-3). For air is c_v = 0.718 kJ/kg·°C at room temperature.

Analysis (*a*) The volume and the mass of the air in the room are

$$V = 4\text{x}5\text{x}7 = 140 \text{ m}^3$$

$$m_{air} = \frac{P_1 V_1}{RT_1} = \frac{(100 \text{ kPa})(140 \text{ m}^3)}{(0.2870 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(295 \text{ K})} = 165.4 \text{ kg}$$



Taking the contents of the room, including the water, as our system, the energy balance can be written as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}} \rightarrow 0 = \Delta U = (\Delta U)_{\text{water}} + (\Delta U)_{\text{air}}$$

or $$[mc(T_2 - T_1)]_{\text{water}} + [mc_v(T_2 - T_1)]_{\text{air}} = 0$$

Substituting,

$$(1000 \text{ kg})(4.18 \text{ kJ/kg}\cdot^\circ\text{C})(T_f - 80)^\circ\text{C} + (165.4 \text{ kg})(0.718 \text{ kJ/kg}\cdot^\circ\text{C})(T_f - 22)^\circ\text{C} = 0$$

It gives the final equilibrium temperature in the room to be

$$T_f = \mathbf{78.4°C}$$

(*b*) Considering that the system is well-insulated and no mass is entering and leaving, the total entropy change during this process is the sum of the entropy changes of water and the room air,

$$\Delta S_{\text{total}} = S_{\text{gen}} = \Delta S_{\text{air}} + \Delta S_{\text{water}}$$

where

$$\Delta S_{\text{air}} = mc_v \ln\frac{T_2}{T_1} + mR\ln\cancelto{0}{\frac{V_2}{V_1}} = (165.4 \text{ kg})(0.718 \text{ kJ/kg}\cdot\text{K})\ln\frac{351.4 \text{ K}}{295 \text{ K}} = 20.78 \text{ kJ/K}$$

$$\Delta S_{\text{water}} = mc\ln\frac{T_2}{T_1} = (1000 \text{ kg})(4.18 \text{ kJ/kg}\cdot\text{K})\ln\frac{351.4 \text{ K}}{353 \text{ K}} = -18.99 \text{ kJ/K}$$

Substituting, the total entropy change is determined to be

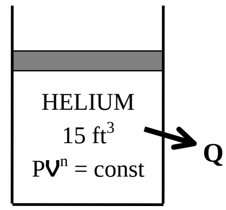
$$\Delta S_{\text{total}} = 20.78 - 18.99 = \mathbf{1.79 \text{ kJ/K}}$$

7-171E A cylinder initially filled with helium gas at a specified state is compressed polytropically to a specified temperature and pressure. The entropy changes of the helium and the surroundings are to be determined, and it is to be assessed if the process is reversible, irreversible, or impossible.

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 = 2.6805 psia.ft^3/lbm.R = 0.4961 Btu/lbm.R. The specific heats of helium are c_v = 0.753 and c_p = 1.25 Btu/lbm.R (Table A-2E).

Analysis (*a*) The mass of helium is



$$m = \frac{P_1 \mathbf{V}_1}{RT_1} = \frac{(25 \text{ psia})(15 \text{ ft}^3)}{(2.6805 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(530 \text{ R})} = 0.264 \text{ lbm}$$

Then the entropy change of helium becomes

$$\Delta S_{\text{sys}} = \Delta S_{\text{helium}} = m\left(c_{p,\text{avg}} \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}\right)$$

$$= (0.264 \text{ lbm})\left[(1.25 \text{ Btu/lbm} \cdot \text{R}) \ln\frac{760 \text{ R}}{530 \text{ R}} - (0.4961 \text{ Btu/lbm} \cdot \text{R}) \ln\frac{70 \text{ psia}}{25 \text{ psia}}\right] = \mathbf{-0.016 \text{ Btu/R}}$$

(*b*) The exponent n and the boundary work for this polytropic process are determined to be

$$\frac{P_1 \mathbf{V}_1}{T_1} = \frac{P_2 \mathbf{V}_2}{T_2} \longrightarrow \mathbf{V}_2 = \frac{T_2}{T_1}\frac{P_1}{P_2}\mathbf{V}_1 = \frac{(760 \text{ R})(25 \text{ psia})}{(530 \text{ R})(70 \text{ psia})}(15 \text{ ft}^3) = 7.682 \text{ ft}^3$$

$$P_2 \mathbf{V}_2^n = P_1 \mathbf{V}_1^n \longrightarrow \left(\frac{P_2}{P_1}\right) = \left(\frac{\mathbf{V}_1}{\mathbf{V}_2}\right)^n \longrightarrow \left(\frac{70}{25}\right) = \left(\frac{15}{7.682}\right)^n \longrightarrow n = 1.539$$

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

$$W_{\text{b,in}} = -\int_1^2 P\,d\mathbf{V} = -\frac{P_2 \mathbf{V}_2 - P_1 \mathbf{V}_1}{1-n} = -\frac{mR(T_2 - T_1)}{1-n}$$

$$= -\frac{(0.264 \text{ lbm})(0.4961 \text{ Btu/lbm} \cdot \text{R})(760 - 530)\text{R}}{1 - 1.539} = 55.9 \text{ Btu}$$

We take the helium in the cylinder as the system, which is a closed system. Taking the direction of heat transfer to be from the cylinder, the energy balance for this stationary closed system can be expressed as

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

$$-Q_{\text{out}} + W_{\text{b,in}} = \Delta U = m(u_2 - u_1)$$

$$-Q_{\text{out}} = m(u_2 - u_1) - W_{b,in}$$

$$Q_{\text{out}} = W_{\text{b,in}} - mc_v(T_2 - T_1)$$

Substituting, $\quad Q_{\text{out}} = 55.9 \text{ Btu} - (0.264 \text{ lbm})(0.753 \text{ Btu/lbm} \cdot \text{R})(760 - 530)\text{R} = 10.2 \text{ Btu}$

Noting that the surroundings undergo a reversible isothermal process, its entropy change becomes

$$\Delta S_{\text{surr}} = \frac{Q_{\text{surr,in}}}{T_{\text{surr}}} = \frac{10.2 \text{ Btu}}{530 \text{ R}} = \mathbf{0.019 \text{ Btu/R}}$$

(*c*) Noting that the system+surroundings combination can be treated as an isolated system,

$$\Delta S_{\text{total}} = \Delta S_{\text{sys}} + \Delta S_{\text{surr}} = -0.016 + 0.019 = 0.003 \text{ Btu/R} > 0$$

Therefore, the process is **irreversible**.

7-172 Air is compressed steadily by a compressor from a specified state to a specified pressure. The minimum power input required is to be determined for the cases of adiabatic and isothermal operation.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats. **4** The process is reversible since the work input to the compressor will be minimum when the compression process is reversible.

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

Analysis (a) For the adiabatic case, the process will be reversible and adiabatic (i.e., isentropic),

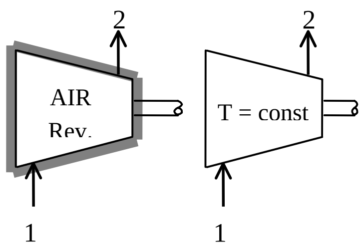
thus the isentropic relations are applicable.

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

and

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = \frac{700 \text{ kPa}}{100 \text{ kPa}}(1.2311) = 8.6177 \quad \rightarrow \quad \begin{array}{l} T_2 = 503.3 \text{ K} \\ h_2 = 506.45 \text{ kJ/kg} \end{array}$$

The energy balance for the compressor, which is a steady-flow system, can be expressed in the rate form as



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

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

$$\dot{W}_{in} + \dot{m}h_1 = \dot{m}h_2 \rightarrow \dot{W}_{in} = \dot{m}(h_2 - h_1)$$

Substituting, the power input to the compressor is determined to be

$$\dot{W}_{in} = (5/60 \text{ kg/s})(506.45 - 290.16) \text{kJ/kg} = \mathbf{18.0 \text{ kW}}$$

(b) In the case of the reversible isothermal process, the steady-flow energy balance becomes

$$\dot{E}_{in} = \dot{E}_{out} \rightarrow \dot{W}_{in} + \dot{m}h_1 - \dot{Q}_{out} = \dot{m}h_2 \rightarrow \dot{W}_{in} = \dot{Q}_{out} + \dot{m}(h_2 - h_1)^{\nearrow 0} = \dot{Q}_{out}$$

since $h = h(T)$ for ideal gases, and thus the enthalpy change in this case is zero. Also, for a reversible isothermal process,

$$\dot{Q}_{out} = \dot{m}T(s_1 - s_2) = -\dot{m}T(s_2 - s_1)$$

where

$$s_2 - s_1 = \left(s_2^\circ - s_1^\circ\right)^{\nearrow 0} - R\ln\frac{P_2}{P_1} = -R\ln\frac{P_2}{P_1} = -(0.287 \text{ kJ/kg}\cdot\text{K})\ln\frac{700 \text{ kPa}}{100 \text{ kPa}} = -0.5585 \text{ kJ/kg}\cdot\text{K}$$

Substituting, the power input for the reversible isothermal case becomes

$$\dot{W}_{in} = -(5/60 \text{ kg/s})(290 \text{ K})(-0.5585 \text{ kJ/kg}\cdot\text{K}) = \mathbf{13.5 \text{ kW}}$$

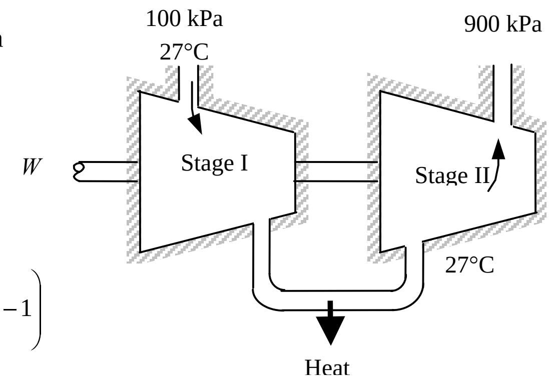
7-173 Air is compressed in a two-stage ideal compressor with intercooling. For a specified mass flow rate of air, the power input to the compressor is to be determined, and it is to be compared to the power input to a single-stage compressor.

Assumptions **1** The compressor operates steadily. **2** Kinetic and potential energies are negligible. **3** The compression process is reversible adiabatic, and thus isentropic. **4** Air is an ideal gas with constant specific heats at room temperature.

Properties The gas constant of air is R = 0.287 kPa.m^3/kg.K (Table A-1). The specific heat ratio of air is k = 1.4 (Table A-2).

Analysis The intermediate pressure between the two stages is

$$P_x = \sqrt{P_1 P_2} = \sqrt{(100\text{ kPa})(900\text{ kPa})} = 300\text{ kPa}$$



The compressor work across each stage is the same, thus total compressor work is twice the compression work for a single stage:

$$\begin{aligned} w_{\text{comp,in}} &= (2)(w_{\text{comp,in,I}}) = 2\frac{kRT_1}{k-1}\left((P_x/P_1)^{(k-1)/k} - 1\right) \\ &= 2\frac{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(300\text{ K})}{1.4-1}\left(\left(\frac{300\text{ kPa}}{100\text{ kPa}}\right)^{0.4/1.4} - 1\right) \\ &= 222.2\text{ kJ/kg} \end{aligned}$$

and

$$\dot{W}_{\text{in}} = \dot{m}w_{\text{comp,in}} = (0.02\text{ kg/s})(222.2\text{ kJ/kg}) = \mathbf{4.44\ kW}$$

The work input to a single-stage compressor operating between the same pressure limits would be

$$w_{\text{comp,in}} = \frac{kRT_1}{k-1}\left((P_2/P_1)^{(k-1)/k} - 1\right) = \frac{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(300\text{ K})}{1.4-1}\left(\left(\frac{900\text{ kPa}}{100\text{ kPa}}\right)^{0.4/1.4} - 1\right) = 263.2\text{ kJ/kg}$$

and

$$\dot{W}_{\text{in}} = \dot{m}w_{\text{comp,in}} = (0.02\text{ kg/s})(263.2\text{ kJ/kg}) = \mathbf{5.26\ \ kW}$$

Discussion Note that the power consumption of the compressor decreases significantly by using 2-stage compression with intercooling.

7-174 A three-stage compressor with two stages of intercooling is considered. The two intermediate pressures that will minimize the work input are to be determined in terms of the inlet and exit pressures.

Analysis The work input to this three-stage compressor with intermediate pressures P_x and P_y and two intercoolers can be expressed as

$$\begin{aligned}
w_{\text{comp}} &= w_{\text{comp,I}} + w_{\text{comp,II}} + w_{\text{comp,III}} \\
&= \frac{nRT_1}{n-1}\left(1-\left(P_x/P_1\right)^{(n-1)/n}\right) + \frac{nRT_1}{n-1}\left(1-\left(P_y/P_x\right)^{(n-1)/n}\right) + \frac{nRT_1}{n-1}\left(1-\left(P_x/P_1\right)^{(n-1)/n}\right) \\
&= \frac{nRT_1}{n-1}\left(1-\left(P_x/P_1\right)^{(n-1)/n} + 1-\left(P_y/P_x\right)^{(n-1)/n} + 1-\left(P_x/P_1\right)^{(n-1)/n}\right) \\
&= \frac{nRT_1}{n-1}\left(3-\left(P_x/P_1\right)^{(n-1)/n} - \left(P_y/P_x\right)^{(n-1)/n} - \left(P_x/P_1\right)^{(n-1)/n}\right)
\end{aligned}$$

The P_x and P_y values that will minimize the work input are obtained by taking the partial differential of w with respect to P_x and P_y, and setting them equal to zero:

$$\frac{\partial w}{\partial P_x} = 0 \longrightarrow -\frac{n-1}{n}\left(\frac{1}{P_1}\right)\left(\frac{P_x}{P_1}\right)^{\frac{n-1}{n}-1} + \frac{n-1}{n}\left(\frac{1}{P_y}\right)\left(\frac{P_x}{P_y}\right)^{-\frac{n-1}{n}-1} = 0$$

$$\frac{\partial w}{\partial P_y} = 0 \longrightarrow -\frac{n-1}{n}\left(\frac{1}{P_x}\right)\left(\frac{P_y}{P_x}\right)^{\frac{n-1}{n}-1} + \frac{n-1}{n}\left(\frac{1}{P_2}\right)\left(\frac{P_y}{P_2}\right)^{-\frac{n-1}{n}-1} = 0$$

Simplifying,

$$\frac{1}{P_1}\left(\frac{P_x}{P_1}\right)^{-\frac{1}{n}} = \frac{1}{P_y}\left(\frac{P_x}{P_y}\right)^{-\frac{2n-1}{n}} \longrightarrow \frac{1}{P_1^n}\left(\frac{P_1}{P_x}\right) = \frac{1}{P_y^n}\left(\frac{P_x}{P_y}\right)^{1-2n} \longrightarrow P_x^{2(1-n)} = \left(P_1 P_y\right)^{1-n}$$

$$\frac{1}{P_x}\left(\frac{P_y}{P_x}\right)^{-\frac{1}{n}} = \frac{1}{P_2}\left(\frac{P_y}{P_2}\right)^{-\frac{2n-1}{n}} \longrightarrow \frac{1}{P_x^n}\left(\frac{P_x}{P_y}\right) = \frac{1}{P_2^n}\left(\frac{P_y}{P_2}\right)^{1-2n} \longrightarrow P_y^{2(1-n)} = \left(P_x P_2\right)^{1-n}$$

which yield

$$\boldsymbol{P_x^2 = P_1\sqrt{P_x P_2} \longrightarrow P_x = \left(P_1^2 P_2\right)^{1/3}}$$

$$\boldsymbol{P_y^2 = P_2\sqrt{P_1 P_y} \longrightarrow P_y = \left(P_1 P_2^2\right)^{1/3}}$$

7-175 Steam expands in a two-stage adiabatic turbine from a specified state to specified pressure. Some steam is extracted at the end of the first stage. The power output of the turbine is to be determined for the cases of 100% and 88% isentropic efficiencies.

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 turbine is adiabatic and thus heat transfer is negligible.

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

$$\left.\begin{array}{l} P_1 = 6\text{ MPa} \\ T_1 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3423.1\text{ kJ/kg} \\ s_1 = 6.8826\text{ kJ/kg}\cdot\text{K} \end{array}$$

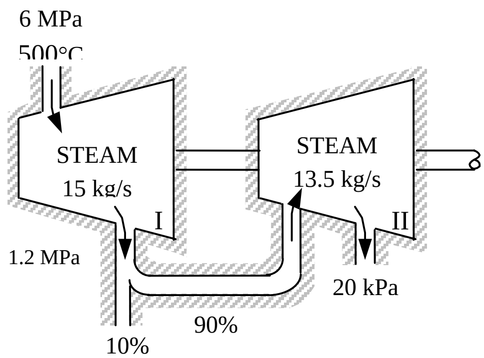
$$\left.\begin{array}{l} P_2 = 1.2\text{ MPa} \\ s_2 = s_1 \end{array}\right\} h_2 = 2962.8\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 20\text{ kPa} \\ s_3 = s_1 \end{array}\right\} \begin{array}{l} x_{3s} = \dfrac{s_{3s} - s_f}{s_{fg}} = \dfrac{6.8826 - 0.8320}{7.0752} = 0.8552 \\ h_{3s} = h_f + x_{3s}h_{fg} = 251.42 + (0.8552)(2357.5) = 2267.5\text{ kJ/kg} \end{array}$$

Analysis (*a*) The mass flow rate through the second stage is

$$\dot{m}_3 = 0.9\dot{m}_1 = (0.9)(15\text{ kg/s}) = 13.5\text{ kg/s}$$

We take the entire turbine, including the connection part between the two stages, as the system, which is a control volume since mass crosses the boundary. Noting that one fluid stream enters the turbine and two fluid streams leave, the energy balance for this steady-flow system can be expressed in the rate form as



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

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

$$\dot{m}_1 h_1 = (\dot{m}_1 - \dot{m}_3)h_2 + \dot{W}_{out} + \dot{m}_3 h_3$$

$$\dot{W}_{out} = \dot{m}_1 h_1 - (\dot{m}_1 - \dot{m}_3)h_2 - \dot{m}_3 h_3$$

$$= \dot{m}_1(h_1 - h_2) + \dot{m}_3(h_2 - h_3)$$

Substituting, the power output of the turbine is

$$\dot{W}_{out} = (15\text{ kg/s})(3423.1 - 2962.8)\text{kJ/kg} + (13.5\text{ kg})(2962.8 - 2267.5)\text{kJ/kg} = \mathbf{16{,}291\ \ kW}$$

(*b*) If the turbine has an adiabatic efficiency of 88%, then the power output becomes

$$\dot{W}_a = \eta_T \dot{W}_s = (0.88)(16{,}291\text{ kW}) = \mathbf{14{,}336\ \ kW}$$

7-176 Steam expands in an 84% efficient two-stage adiabatic turbine from a specified state to a specified pressure. Steam is reheated between the stages. For a given power output, the mass flow rate of steam through the turbine is 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 turbine is adiabatic and thus heat transfer is negligible.

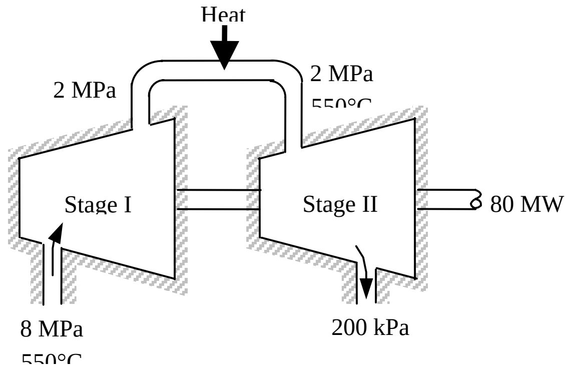
Properties From the steam tables (Tables A-4 through 6)

$$\left.\begin{array}{l} P_1 = 8 \text{ MPa} \\ T_1 = 550°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3521.8 \text{ kJ/kg} \\ s_1 = 6.8800 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} P_{2s} = 2 \text{ MPa} \\ s_{2s} = s_1 \end{array}\right\} h_{2s} = 3089.7 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 2 \text{ MPa} \\ T_3 = 550°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3579.0 \text{ kJ/kg} \\ s_3 = 7.5725 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} P_{4s} = 200 \text{ kPa} \\ s_{4s} = s_3 \end{array}\right\} h_{4s} = 2901.7 \text{ kJ/kg}$$



Analysis The power output of the actual turbine is given to be 80 MW. Then the power output for the isentropic operation becomes

$$\dot{W}_{\text{s,out}} = \dot{W}_{\text{a,out}} / \eta_T = (80{,}000 \text{ kW}) / 0.84 = 95{,}240 \text{ kW}$$

We take the entire turbine, excluding the reheat section, as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system in isentropic operation can be expressed in the rate form as

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

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

$$\dot{m}h_1 + \dot{m}h_3 = \dot{m}h_{2s} + \dot{m}h_{4s} + \dot{W}_{\text{s,out}}$$

$$\dot{W}_{\text{s,out}} = \dot{m}[(h_1 - h_{2s}) + (h_3 - h_{4s})]$$

Substituting,

$$95{,}240 \text{ kJ/s} = \dot{m}\left[(3521.8 - 3089.7)\text{kJ/kg} + (3579.0 - 2901.7)\text{kJ/kg}\right]$$

which gives

$$\dot{m} = \mathbf{85.8 \text{ kg/s}}$$

7-177 Refrigerant-134a is compressed by a 0.7-kW adiabatic compressor from a specified state to another specified state. The isentropic efficiency, the volume flow rate at the inlet, and the maximum flow rate at the compressor inlet are to be determined.

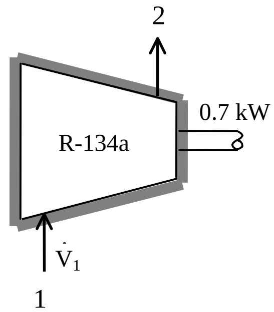
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

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

$$\left.\begin{array}{l} P_1 = 140 \text{ kPa} \\ T_1 = -10°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.14605 \text{ m}^3/\text{kg} \\ h_1 = 246.36 \text{ kJ/kg} \\ s_1 = 0.9724 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 700 \text{ kPa} \\ T_2 = 50°\text{C} \end{array}\right\} h_2 = 288.53 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 700 \text{ kPa} \\ s_{2s} = s_1 \end{array}\right\} h_{2s} = 281.16 \text{ kJ/kg}$$



Analysis (a) The isentropic efficiency is determined from its definition,

$$\eta_C = \frac{h_{2s} - h_1}{h_{2a} - h_1} = \frac{281.16 - 246.36}{288.53 - 246.36} = 0.825 = \mathbf{82.5\%}$$

(*b*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual compressor as the system, which is a control volume. The energy balance for this steady-flow system can be expressed as

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

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

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

$$\dot{W}_{\text{a,in}} = \dot{m}(h_2 - h_1)$$

Then the mass and volume flow rates of the refrigerant are determined to be

$$\dot{m} = \frac{\dot{W}_{\text{a,in}}}{h_{2a} - h_1} = \frac{0.7 \text{ kJ/s}}{(288.53 - 246.36)\text{kJ/kg}} = 0.0166 \text{ kg/s}$$

$$\dot{\boldsymbol{V}}_1 = \dot{m}\boldsymbol{v}_1 = (0.0166 \text{ kg/s})(0.14605 \text{ m}^3/\text{kg}) = 0.00242 \text{ m}^3/\text{s} = \mathbf{145\ L/min}$$

(*c*) The volume flow rate will be a maximum when the process is isentropic, and it is determined similarly from the steady-flow energy equation applied to the isentropic process. It gives

$$\dot{m}_{\text{max}} = \frac{\dot{W}_{\text{s,in}}}{h_{2s} - h_1} = \frac{0.7 \text{ kJ/s}}{(281.16 - 246.36)\text{kJ/kg}} = 0.0201 \text{ kg/s}$$

$$\dot{V}_{1,\text{max}} = \dot{m}_{\text{max}}\boldsymbol{v}_1 = (0.0201 \text{ kg/s})(0.14605 \text{ m}^3/\text{kg}) = 0.00294 \text{ m}^3/\text{s} = \mathbf{176\ \ L/min}$$

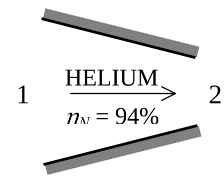
Discussion Note that the raising the isentropic efficiency of the compressor to 100% would increase the volumetric flow rate by more than 20%.

7-178E Helium is accelerated by a 94% efficient nozzle from a low velocity to 1000 ft/s. The pressure and temperature at the nozzle inlet are to be determined.

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

Properties The specific heat ratio of helium is k = 1.667. The constant pressure specific heat of helium is 1.25 Btu/lbm.R (Table A-2E).

Analysis We take nozzle 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



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

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

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta\text{pe} \cong 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} \longrightarrow 0 = c_{p,\text{avg}}(T_2 - T_1) + \frac{V_2^2 - V_1^2}{2}$$

Solving for T_1 and substituting,

$$T_1 = T_{2a} + \frac{V_{2s}^2 - V_1^{2\ \nearrow 0}}{2C_p} = 180°\text{F} + \frac{(1000\text{ ft/s})^2}{2(1.25\text{ Btu/lbm}\cdot\text{R})}\left(\frac{1\text{ Btu/lbm}}{25{,}037\text{ ft}^2/\text{s}^2}\right) = \mathbf{196.0°F} = \mathbf{656\ R}$$

From the isentropic efficiency relation,

$$\eta_N = \frac{h_{2a} - h_1}{h_{2s} - h_1} = \frac{c_P(T_{2a} - T_1)}{c_P(T_{2s} - T_1)}$$

or,

$$T_{2s} = T_1 + (T_{2a} - T_1)/\eta_N = 656 + (640 - 656)/(0.94) = 639\text{ R}$$

From the isentropic relation, $\dfrac{T_{2s}}{T_1} = \left(\dfrac{P_2}{P_1}\right)^{(k-1)/k}$

$$P_1 = P_2\left(\frac{T_1}{T_{2s}}\right)^{k/(k-1)} = (14\text{ psia})\left(\frac{656\text{ R}}{639\text{ R}}\right)^{1.667/0.667} = \mathbf{14.9\ \ psia}$$

7-179 [*Also solved by EES on enclosed CD***]** An adiabatic compressor is powered by a direct-coupled steam turbine, which also drives a generator. The net power delivered to the generator and the rate of 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 devices are adiabatic and thus heat transfer is negligible. **4** 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). From the steam tables (Tables A-4 through 6) and air table (Table A-17),

$$T_1 = 295 \text{ K} \longrightarrow h_1 = 295.17 \text{ kJ/kg},\ s_1^\circ = 1.68515 \text{ kJ/kg}\cdot\text{K}$$

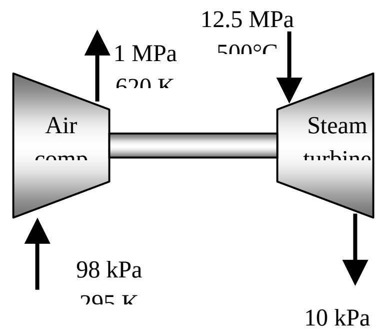
$$T_2 = 620 \text{ K} \longrightarrow h_1 = 628.07 \text{ kJ/kg},\ s_2^\circ = 2.44356 \text{ kJ/kg}\cdot\text{K}$$

$$\left.\begin{array}{l} P_3 = 12.5 \text{ MPa} \\ T_3 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3343.6 \text{ kJ/kg} \\ s_3 = 6.4651 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 10 \text{ kPa} \\ x_4 = 0.92 \end{array}\right\} \begin{array}{l} h_4 = h_f + x_4 h_{fg} = 191.81 + (0.92)(2392.1) = 2392.5 \text{ kJ/kg} \\ s_4 = s_f + x_4 s_{fg} = 0.6492 + (0.92)(7.4996) = 7.5489 \text{ kJ/kg}\cdot\text{K} \end{array}$$

Analysis There is only one inlet and one exit for either device, and thus $\dot{m}_{\text{in}} = \dot{m}_{\text{out}} = \dot{m}$. We take either the turbine or the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for either steady-flow system can be expressed in the rate form as

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



For the turbine and the compressor it becomes

Compressor: $\dot{W}_{\text{comp, in}} + \dot{m}_{\text{air}} h_1 = \dot{m}_{\text{air}} h_2 \rightarrow \dot{W}_{\text{comp, in}} = \dot{m}_{\text{air}}(h_2 - h_1)$

Turbine: $\dot{m}_{\text{steam}} h_3 = \dot{W}_{\text{turb, out}} + \dot{m}_{\text{steam}} h_4 \rightarrow \dot{W}_{\text{turb, out}} = \dot{m}_{\text{steam}}(h_3 - h_4)$

Substituting,

$$\dot{W}_{\text{comp,in}} = (10 \text{ kg/s})(628.07 - 295.17) \text{kJ/kg} = 3329 \text{ kW}$$

$$\dot{W}_{\text{turb,out}} = (25 \text{ kg/s})(3343.6 - 2392.5) \text{kJ/kg} = 23{,}777 \text{ kW}$$

Therefore, $\dot{W}_{\text{net,out}} = \dot{W}_{\text{turb,out}} - \dot{W}_{\text{comp,in}} = 23{,}777 - 3329 = \mathbf{20{,}448\ kW}$

Noting that the system is adiabatic, the total rate of entropy change (or generation) during this process is the sum of the entropy changes of both fluids,

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{air}}(s_2 - s_1) + \dot{m}_{\text{steam}}(s_4 - s_3)$$

where

$$\dot{m}_{\text{air}}(s_2 - s_1) = \dot{m}\left(s_2^\circ - s_1^\circ - R\ln\frac{P_2}{P_1}\right)$$

$$= (10 \text{ kg/s})\left(2.44356 - 1.68515 - 0.287\ln\frac{1000 \text{ kPa}}{98 \text{ kPa}}\right)\text{kJ/kg}\cdot\text{K} = 0.92 \text{ kW/K}$$

$$\dot{m}_{\text{steam}}(s_4 - s_3) = (25 \text{ kg/s})(7.5489 - 6.4651)\text{kJ/kg}\cdot\text{K} = 27.1 \text{ kW/K}$$

Substituting, the total rate of entropy generation is determined to be

$$\dot{S}_{\text{gen,total}} = \dot{S}_{\text{gen,comp}} + \dot{S}_{\text{gen,turb}} = 0.92 + 27.1 = \mathbf{28.02\ \ kW/K}$$

7-180 EES Problem 7-179 is reconsidered. The isentropic efficiencies for the compressor and turbine are to be determined, and then the effect of varying the compressor efficiency over the range 0.6 to 0.8 and the turbine efficiency over the range 0.7 to 0.95 on the net work for the cycle and the entropy generated for the process is to be investigated. The net work is to be plotted as a function of the compressor efficiency for turbine efficiencies of 0.7, 0.8, and 0.9.

Analysis The problem is solved using EES, and the results are tabulated and plotted below.

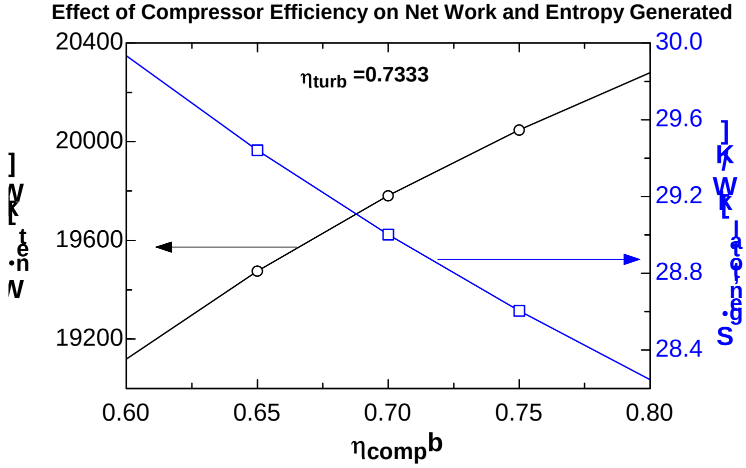
```
"Input Data"
m_dot_air = 10 [kg/s] "air compressor (air) data"
T_air[1]=(295-273) "[C]" "We will input  temperature in C"
P_air[1]=98 [kPa]
T_air[2]=(700-273) "[C]"
P_air[2]=1000 [kPa]
m_dot_st=25 [kg/s] "steam turbine (st) data"
T_st[1]=500 [C]
P_st[1]=12500 [kPa]
P_st[2]=10 [kPa]
x_st[2]=0.92 "quality"
"Compressor Analysis:"
"Conservation of mass for the compressor m_dot_air_in = m_dot_air_out =m_dot_air"
"Conservation of energy for the compressor is:"
E_dot_comp_in - E_dot_comp_out = DELTAE_dot_comp
DELTAE_dot_comp = 0                    "Steady flow requirement"
E_dot_comp_in=m_dot_air*(enthalpy(air,T=T_air[1])) + W_dot_comp_in
E_dot_comp_out=m_dot_air*(enthalpy(air,T=T_air[2]))
"Compressor adiabatic efficiency:"
Eta_comp=W_dot_comp_in_isen/W_dot_comp_in
W_dot_comp_in_isen=m_dot_air*(enthalpy(air,T=T_air_isen[2])-enthalpy(air,T=T_air[1]))
s_air[1]=entropy(air,T=T_air[1],P=P_air[1])
s_air[2]=entropy(air,T=T_air[2],P=P_air[2])
s_air_isen[2]=entropy(air, T=T_air_isen[2],P=P_air[2])
s_air_isen[2]=s_air[1]
"Turbine Analysis:"
"Conservation of mass for the turbine m_dot_st_in = m_dot_st_out =m_dot_st"
"Conservation of energy for the turbine is:"
E_dot_turb_in - E_dot_turb_out = DELTAE_dot_turb
DELTAE_dot_turb = 0                    "Steady flow requirement"
E_dot_turb_in=m_dot_st*h_st[1]
h_st[1]=enthalpy(steam,T=T_st[1], P=P_st[1])
E_dot_turb_out=m_dot_st*h_st[2]+W_dot_turb_out
h_st[2]=enthalpy(steam,P=P_st[2], x=x_st[2])
"Turbine adiabatic efficiency:"
Eta_turb=W_dot_turb_out/W_dot_turb_out_isen
W_dot_turb_out_isen=m_dot_st*(h_st[1]-h_st_isen[2])
s_st[1]=entropy(steam,T=T_st[1],P=P_st[1])
h_st_isen[2]=enthalpy(steam, P=P_st[2],s=s_st[1])
"Note: When Eta_turb is specified as an independent variable in
the Parametric Table, the iteration process may put the steam state 2 in the
superheat region, where the quality is undefined. Thus, s_st[2], T_st[2] are
calculated at P_st[2], h_st[2] and not P_st[2] and x_st[2]"
s_st[2]=entropy(steam,P=P_st[2],h=h_st[2])
T_st[2]=temperature(steam,P=P_st[2], h=h_st[2])
s_st_isen[2]=s_st[1]
"Net work done by the process:"
W_dot_net=W_dot_turb_out-W_dot_comp_in
```

"Entropy generation:"
"Since both the compressor and turbine are adiabatic, and thus there is no heat transfer to the surroundings, the entropy generation for the two steady flow devices becomes:"
S_dot_gen_comp=m_dot_air*(s_air[2]-s_air[1])
S_dot_gen_turb=m_dot_st*(s_st[2]-s_st[1])
S_dot_gen_total=S_dot_gen_comp+S_dot_gen_turb
"To generate the data for Plot Window 1, Comment out the line ' T_air[2]=(700-273) C' and select values for Eta_comp in the Parmetric Table, then press F3 to solve the table. EES then solves for the unknown value of T_air[2] for each Eta_comp."
"To generate the data for Plot Window 2, Comment out the two lines ' x_st[2]=0.92 quality ' and ' h_st[2]=enthalpy(steam,P=P_st[2], x=x_st[2]) ' and select values for Eta_turb in the Parmetric Table, then press F3 to solve the table. EES then solves for the h_st[2] for each Eta_turb."

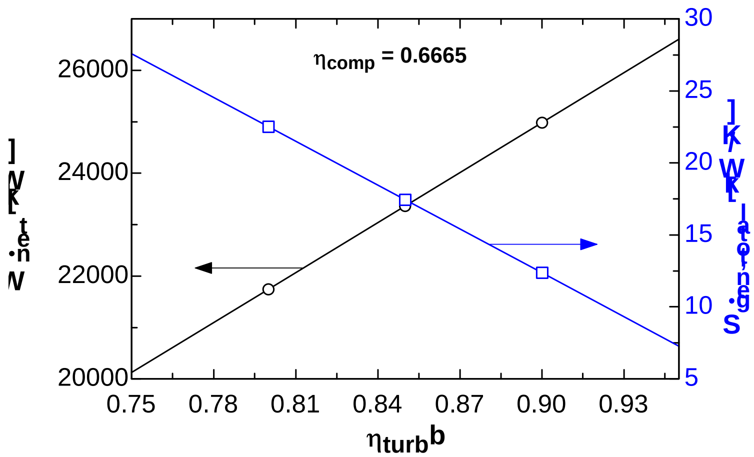
W_{net} [kW]	$S_{gentotal}$ [kW/K]	η_{turb}	η_{comp}
20124	27.59	0.75	0.6665
21745	22.51	0.8	0.6665
23365	17.44	0.85	0.6665
24985	12.36	0.9	0.6665
26606	7.281	0.95	0.6665

W_{net} [kW]	$S_{gentotal}$ [kW/K]	η_{turb}	η_{comp}
19105	30	0.7327	0.6
19462	29.51	0.7327	0.65
19768	29.07	0.7327	0.7
20033	28.67	0.7327	0.75
20265	28.32	0.7327	0.8

Effect of Compressor Efficiency on Net Work and Entropy Generated

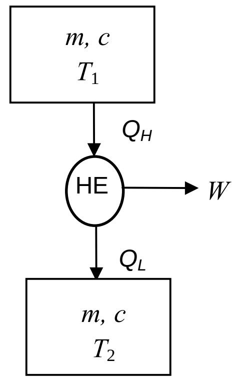


Effect of Turbine Efficiency on Net Work and Entropy Generated



7-181 Two identical bodies at different temperatures are connected to each other through a heat engine. It is to be shown that the final common temperature of the two bodies will be $T_f = \sqrt{T_1 T_2}$ when the work output of the heat engine is maximum.

Analysis For maximum power production, the entropy generation must be zero. Taking the source, the sink, and the heat engine as our system, which is adiabatic, and noting that the entropy change for cyclic devices is zero, the entropy generation for this system can be expressed as



$$S_{\text{gen}} = (\Delta S)_{\text{source}} + (\Delta S)_{\text{engine}}^{\nearrow 0} + (\Delta S)_{\text{sink}} = 0$$

$$mc\ln\frac{T_f}{T_1} + 0 + mc\ln\frac{T_f}{T_2} = 0$$

$$\ln\frac{T_f}{T_1} + \ln\frac{T_f}{T_2} = 0 \longrightarrow \ln\frac{T_f}{T_1}\frac{T_f}{T_2} = 0 \longrightarrow T_f^2 = T_1 T_2$$

and thus

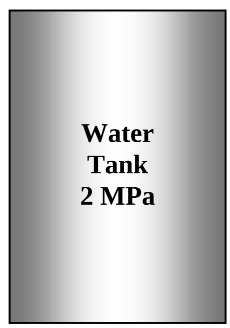
$$T_f = \sqrt{T_1 T_2}$$

for maximum power production.

7-182 The pressure in a hot water tank rises to 2 MPa, and the tank explodes. The explosion energy of the water is to be determined, and expressed in terms of its TNT equivalence.

Assumptions **1** The expansion process during explosion is isentropic. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer with the surroundings during explosion is negligible.

Properties The explosion energy of TNT is 3250 kJ/kg. From the steam tables (Tables A-4 through 6)



$$\left.\begin{array}{l} P_1 = 2 \text{ MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = v_{f@2\text{ MPa}} = 0.001177 \text{ m}^3/\text{kg} \\ u_1 = u_{f@2\text{ MPa}} = 906.12 \text{ kJ/kg} \\ s_1 = s_{f@2\text{ MPa}} = 2.4467 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 100 \text{ kPa} \\ s_2 = s_1 \end{array}\right\} \begin{array}{ll} u_f = 417.40, & u_{fg} = 2088.2 \text{ kJ/kg} \\ s_f = 1.3028, & s_{fg} = 6.0562 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$x_2 = \frac{s_2 - s_f}{s_{fg}} = \frac{2.4467 - 1.3028}{6.0562} = 0.1889$$

$$u_2 = u_f + x_2 u_{fg} = 417.40 + (0.1889)(2088.2) = 811.83 \text{ kJ/kg}$$

Analysis We idealize the water tank as a closed system that undergoes a reversible adiabatic process with negligible changes in kinetic and potential energies. The work done during this idealized process represents the explosive energy of the tank, and is determined from the closed system energy balance to be

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

$$-W_{\text{b,out}} = \Delta U = m(u_2 - u_1)$$

$$E_{\text{exp}} = W_{\text{b,out}} = m(u_1 - u_2)$$

where $$m = \frac{\boldsymbol{V}}{\boldsymbol{v}_1} = \frac{0.080 \text{ m}^3}{0.001177 \text{ m}^3/\text{kg}} = 67.99 \text{ kg}$$

Substituting, $$E_{\text{exp}} = (67.99 \text{ kg})(906.12 - 811.83)\text{kJ/kg} = 6410 \text{ kJ}$$

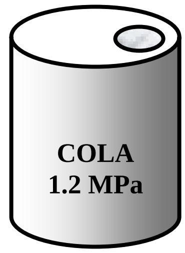
which is equivalent to $m_{\text{TNT}} = \dfrac{6410\ \text{kJ}}{3250\ \text{kJ/kg}} = \mathbf{1.972\ \ kg\ TNT}$

7-183 A 0.35-L canned drink explodes at a pressure of 1.2 MPa. The explosive energy of the drink is to be determined, and expressed in terms of its TNT equivalence.

Assumptions **1** The expansion process during explosion is isentropic. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer with the surroundings during explosion is negligible. **4** The drink can be treated as pure water.

Properties The explosion energy of TNT is 3250 kJ/kg. From the steam tables (Tables A-4 through 6)

$$\left.\begin{array}{l} P_1 = 1.2 \text{ MPa} \\ \text{Comp. liquid} \end{array}\right\} \begin{array}{l} v_1 = v_{f@1.2 \text{ MPa}} = 0.001138 \text{ m}^3/\text{kg} \\ u_1 = u_{f@1.2 \text{ MPa}} = 796.96 \text{ kJ/kg} \\ s_1 = s_{f@1.2 \text{ MPa}} = 2.2159 \text{ kJ/kg}\cdot\text{K} \end{array}$$



$$\left.\begin{array}{l} P_2 = 100 \text{ kPa} \\ s_2 = s_1 \end{array}\right\} \begin{array}{ll} u_f = 417.40, & u_{fg} = 2088.2 \text{ kJ/kg} \\ s_f = 1.3028, & s_{fg} = 6.0562 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$x_2 = \frac{s_2 - s_f}{s_{fg}} = \frac{2.2159 - 1.3028}{6.0562} = 0.1508$$

$$u_2 = u_f + x_2 u_{fg} = 417.40 + (0.1508)(2088.2) = 732.26 \text{ kJ/kg}$$

Analysis We idealize the canned drink as a closed system that undergoes a reversible adiabatic process with negligible changes in kinetic and potential energies. The work done during this idealized process represents the explosive energy of the can, and is determined from the closed system energy balance to be

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

$$-W_{b,out} = \Delta U = m(u_2 - u_1)$$

$$E_{exp} = W_{b,out} = m(u_1 - u_2)$$

where

$$m = \frac{V}{v_1} = \frac{0.00035 \text{ m}^3}{0.001138 \text{ m}^3/\text{kg}} = 0.3074 \text{ kg}$$

Substituting,

$$E_{exp} = (0.3074 \text{ kg})(796.96 - 732.26)\text{kJ/kg} = \mathbf{19.9 \text{ kJ}}$$

which is equivalent to

$$m_{TNT} = \frac{19.9 \text{ kJ}}{3250 \text{ kJ/kg}} = \mathbf{0.00612 \text{ kg TNT}}$$