7–58 An adiabatic pump is to be used to compress saturated liquid water at 10 kPa to a pressure to 15 MPa in a reversible manner. Determine the work input using (*a*) entropy data from the compressed liquid table, (*b*) inlet specific volume and pressure values, (*c*) average specific volume and pressure values. Also, determine the errors involved in parts (*b*) and (*c*).

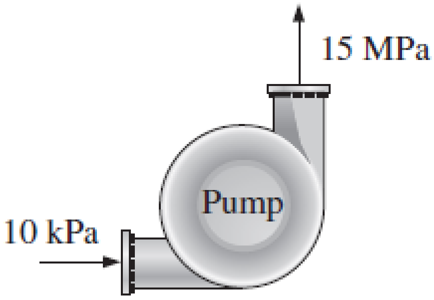


FIGURE P7–58

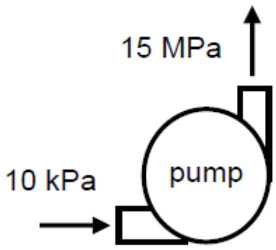
Solution An adiabatic pump is used to compress saturated liquid water in a reversible manner. The work input is to be determined by different approaches.

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

Analysis The properties of water at the inlet and exit of the pump are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 10 \text{ kPa} \\ x_1 = 0 \end{array}\right\} \begin{array}{l} h_1 = 191.81 \text{ kJ/kg} \\ s_1 = 0.6492 \text{ kJ/kg} \\ v_1 = 0.001010 \text{ m}^3/\text{kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 15 \text{ MPa} \\ s_2 = s_1 \end{array}\right\} \begin{array}{l} h_2 = 206.90 \text{ kJ/kg} \\ v_2 = 0.001004 \text{ m}^3/\text{kg} \end{array}$$



(*a*) Using the entropy data from the compressed liquid water table

$$w_P = h_2 - h_1 = 206.90 - 191.81 = \mathbf{15.10 \text{ kJ/kg}}$$

(*b*) Using inlet specific volume and pressure values

$$w_P = v_1(P_2 - P_1) = (0.001010 \text{ m}^3/\text{kg})(15{,}000 - 10)\text{kPa} = \mathbf{15.14 \text{ kJ/kg}}$$

Error = **0.3%**

(*b*) Using average specific volume and pressure values

$$w_P = v_{avg}(P_2 - P_1) = \left[1/2(0.001010 + 0.001004) \text{ m}^3/\text{kg}\right](15{,}000 - 10)\text{kPa} = \mathbf{15.10 \text{ kJ/kg}}$$

Error = **0%**

Discussion The results show that any of the method may be used to calculate reversible pump work.

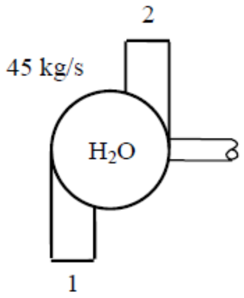
7–90 Water enters the pump of a steam power plant as saturated liquid at 20 kPa at a rate of 45 kg/s and exits at 6 MPa. Neglecting the changes in kinetic and potential energies and assuming the process to be reversible, determine the power input to the pump.

Solution Liquid water is pumped reversibly to a specified pressure at a specified rate. The power input to the pump is to be determined.

Assumptions **1** Liquid water is an incompressible substance. **2** Kinetic and potential energy changes are negligible. **3** The process is reversible.

Properties The specific volume of saturated liquid water at 20 kPa is $v_1 = v_{f\,@\,20\text{ kPa}} = 0.001017\ \text{m}^3/\text{kg}$ (Table A-5).

Analysis The power input to the pump can be determined directly from the steady-flow work relation for a liquid,



$$\dot{W}_{in} = \dot{m}\left(\int_1^2 v\,dP + \Delta ke^{\nearrow 0} + \Delta pe^{\nearrow 0}\right) = \dot{m}v_1(P_2 - P_1)$$

Substituting,

$$\dot{W}_{in} = (45\ \text{kg/s})(0.001017\ \text{m}^3/\text{kg})(6000 - 20)\text{kPa}\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) = \mathbf{274\ \ kW}$$

7–91 Liquid water enters a 25-kW pump at 100-kPa pressure at a rate of 5 kg/s. Determine the highest pressure the liquid water can have at the exit of the pump. Neglect the kinetic and potential energy changes of water, and take the specific volume of water to be 0.001 m^3/kg. *Answer:* 5100 kPa

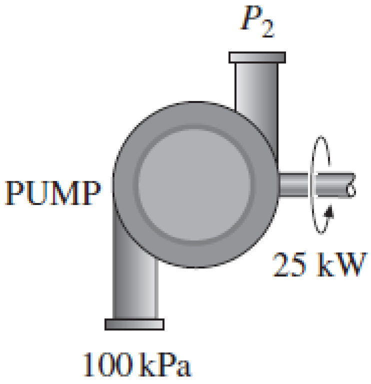


FIGURE P7–91

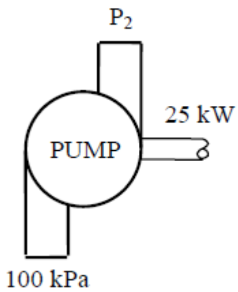
Solution Liquid water is to be pumped by a 25-kW pump at a specified rate. The highest pressure the water can be pumped to is to be determined.

Assumptions **1** Liquid water is an incompressible substance. **2** Kinetic and potential energy changes are negligible. **3** The process is assumed to be reversible since we will determine the limiting case.

Properties The specific volume of liquid water is given to be v_1 = 0.001 m^3/kg.

Analysis The highest pressure the liquid can have at the pump exit can be determined from the reversible steady-flow work relation for a liquid,

$$\dot{W}_{\text{in}} = \dot{m}\left(\int_1^2 v\,dP + \Delta ke^{\nearrow 0} + \Delta pe^{\nearrow 0}\right) = \dot{m}v_1(P_2 - P_1)$$



Thus,

$$25 \text{ kJ/s} = (5 \text{ kg/s})(0.001 \text{ m}^3/\text{kg})(P_2 - 100)\text{k Pa}\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right)$$

It yields $P_2 = \mathbf{5100\ kPa}$

7–93 Consider a steam power plant that operates between the pressure limits of 10 MPa and 20 kPa. Steam enters the pump as saturated liquid and leaves the turbine as saturated vapor. Determine the ratio of the work delivered by the turbine to the work consumed by the pump. Assume the entire cycle to be reversible and the heat losses from the pump and the turbine to be negligible.

Solution A steam power plant operates between the pressure limits of 10 MPa and 20 kPa. The ratio of the turbine work to the pump work is to be determined.

Assumptions **1** Liquid water is an incompressible substance. **2** Kinetic and potential energy changes are negligible. **3** The process is reversible. **4** The pump and the turbine are adiabatic.

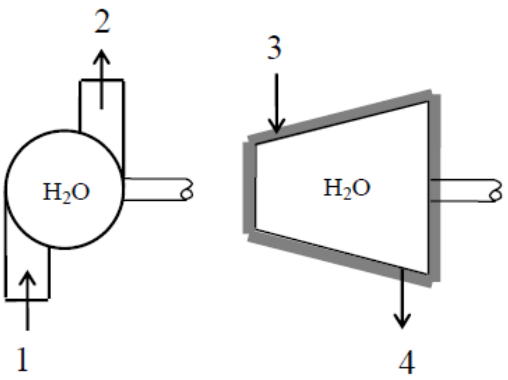
Properties The specific volume of saturated liquid water at 20 kPa is $v_1 = v_{f\,@\,20\text{ kPa}} = 0.001017 \text{ m}^3\text{/kg}$ (Table A-5).

Analysis Both the compression and expansion processes are reversible and adiabatic, and thus isentropic, $s_1 = s_2$ and $s_3 = s_4$. Then the properties of the steam are

$$\left.\begin{array}{l} P_4 = 20 \text{ kPa} \\ sat.vapor \end{array}\right\} \begin{array}{l} h_4 = h_{g\,@20\text{ kPa}} = 2608.9 \text{ kJ/kg} \\ s_4 = s_{g\,@20\text{ kPa}} = 7.9073 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_3 = 10 \text{ MPa} \\ s_3 = s_4 \end{array}\right\} h_3 = 4707.2 \text{ kJ/kg}$$

Also, $v_1 = v_{f\,@\,20\text{ kPa}} = 0.001017 \text{ m}^3\text{/kg}$.



The work output to this isentropic turbine is determined from the steady-flow energy balance to be

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\not{\,}0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

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

$$\dot{m}h_3 = \dot{m}h_4 + \dot{W}_{out}$$

$$\dot{W}_{out} = \dot{m}(h_3 - h_4)$$

Substituting,

$$w_{turb,out} = h_3 - h_4 = 4707.2 - 2608.9 = 2098.3 \text{ kJ/kg}$$

The pump work input is determined from the steady-flow work relation to be

$$w_{pump,in} = \int_1^2 v\,dP + \Delta ke^{\not{\,}0} + \Delta pe^{\not{\,}0} = v_1(P_2 - P_1)$$

$$= (0.001017 \text{ m}^3\text{/kg})(10{,}000 - 20)\text{kPa}\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right)$$

$$= 10.15 \text{ kJ/kg}$$

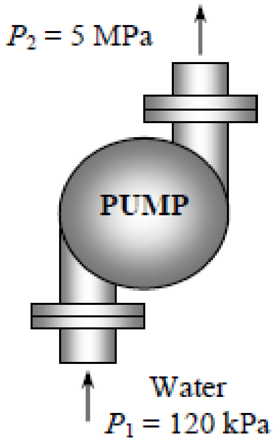
Thus,

$$\frac{w_{turb,out}}{w_{pump,in}} = \frac{2098.3}{10.15} = \mathbf{206.7}$$

7–95 Liquid water at 120 kPa enters a 7-kW pump where its pressure is raised to 5 MPa. If the elevation difference between the exit and the inlet levels is 10 m, determine the highest mass flow rate of liquid water this pump can handle. Neglect the kinetic energy change of water, and take the specific volume of water to be 0.001 m3/kg.

Solution Liquid water is pumped by a 7-kW pump to a specified pressure at a specified level. The highest possible mass flow rate of water is to be determined

Assumptions **1** Liquid water is an incompressible substance. **2** Kinetic energy changes are negligible, but potential energy changes may be significant. **3** The process is assumed to be reversible since we will determine the limiting case.



Properties The specific volume of liquid water is given to be v_1 = 0.001 m^3/kg.

Analysis The highest mass flow rate will be realized when the entire process is reversible. Thus it is determined from the reversible steady-flow work relation for a liquid,

$$\dot{W}_{\text{in}} = \dot{m}\left(\int_1^2 v\,dP + \Delta ke^{\not{\,0}} + \Delta pe\right) = \dot{m}\{v(P_2 - P_1) + g(z_2 - z_1)\}$$

Thus,

$$7\text{ kJ/s} = \dot{m}\left\{(0.001\text{ m}^3/\text{kg})(5000 - 120)\text{kPa}\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) + (9.8\text{ m/s}^2)(10\text{ m})\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right)\right\}$$

It yields

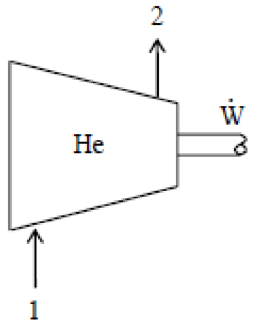
$$\dot{m} = \mathbf{1.41\ kg/s}$$

7–96 Helium gas is compressed from 100 kPa and 20°C to 850 kPa at a rate of 0.15 m^3/s. Determine the power input to the compressor, assuming the compression process to be (*a*) isentropic, (*b*) polytropic with n = 1.2, (*c*) isothermal, and (*d*) ideal two-stage polytropic with n = 1.2.

Solution Helium gas is compressed from a specified state to a specified pressure at a specified rate. The power input to the compressor is to be determined for the cases of isentropic, polytropic, isothermal, and two-stage compression.

Assumptions **1** Helium is an ideal gas with constant specific heats. **2** The process is reversible. **3** Kinetic and potential energy changes are negligible.

Properties The gas constant of helium is R = 2.0769 kPa.m^3/kg.K. The specific heat ratio of helium is k = 1.667 (Table A-2).



Analysis The mass flow rate of helium is

$$\dot{m} = \frac{P_1 \dot{V}_1}{RT_1} = \frac{(100 \text{ kPa})(0.15 \text{ m}^3/\text{s})}{(2.0769 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(293 \text{ K})} = 0.02465 \text{ kg/s}$$

(a) Isentropic compression with k = 1.667:

$$\dot{W}_{\text{comp,in}} = \dot{m}\frac{kRT_1}{k-1}\left\{\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right\}$$

$$= (0.02465 \text{ kg/s})\frac{(1.667)(2.0769 \text{ kJ/kg} \cdot \text{K})(293 \text{ K})}{1.667 - 1}\left\{\left(\frac{850 \text{ kPa}}{100 \text{ kPa}}\right)^{0.667/1.667} - 1\right\}$$

$$= \mathbf{50.8 \text{ kW}}$$

(b) Polytropic compression with n = 1.2:

$$\dot{W}_{\text{comp,in}} = \dot{m}\frac{nRT_1}{n-1}\left\{\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1\right\}$$

$$= (0.02465 \text{ kg/s})\frac{(1.2)(2.0769 \text{ kJ/kg} \cdot \text{K})(293 \text{ K})}{1.2 - 1}\left\{\left(\frac{850 \text{ kPa}}{100 \text{ kPa}}\right)^{0.2/1.2} - 1\right\}$$

$$= \mathbf{38.6 \text{ kW}}$$

(c) Isothermal compression:

$$\dot{W}_{\text{comp,in}} = \dot{m}RT\ln\frac{P_2}{P_1} = (0.02465 \text{ kg/s})(2.0769 \text{ kJ/kg} \cdot \text{K})(293 \text{ K})\ln\frac{850 \text{ kPa}}{100 \text{ kPa}} = \mathbf{32.1 \text{ kW}}$$

(d) Ideal two-stage compression with intercooling (n = 1.2): In this case, the pressure ratio across each stage is the same, and its value is determined from

$$P_x = \sqrt{P_1 P_2} = \sqrt{(100 \text{ kPa})(850 \text{ kPa})} = 291.5 \text{ kPa}$$

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

$$\dot{W}_{\text{comp,in}} = 2\dot{m}\frac{nRT_1}{n-1}\left\{\left(\frac{P_x}{P_1}\right)^{(n-1)/n} - 1\right\}$$

$$= 2(0.02465 \text{ kg/s})\frac{(1.2)(2.0769 \text{ kJ/kg} \cdot \text{K})(293 \text{ K})}{1.2 - 1}\left\{\left(\frac{291.5 \text{ kPa}}{100 \text{ kPa}}\right)^{0.2/1.2} - 1\right\}$$

$$= \mathbf{35.1 \text{ kW}}$$

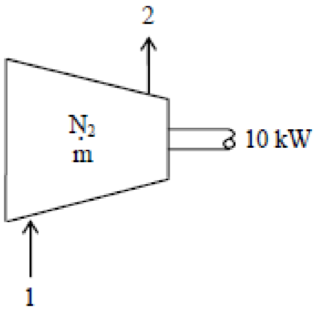
7–98 Nitrogen gas is compressed from 80 kPa and 27°C to 480 kPa by a 10-kW compressor. Determine the mass flow rate of nitrogen through the compressor, assuming the compression process to be (*a*) isentropic, (*b*) polytropic with n = 1.3, (*c*) isothermal, and (*d*) ideal two-stage polytropic with n = 1.3. *Answers:* (*a*) 0.048 kg/s, (*b*) 0.051 kg/s, (*c*) 0.063 kg/s, (*d*) 0.056 kg/s

Solution Nitrogen gas is compressed by a 10-kW compressor from a specified state to a specified pressure. The mass flow rate of nitrogen through the compressor is to be determined for the cases of isentropic, polytropic, isothermal, and two-stage compression.

Assumptions **1** Nitrogen is an ideal gas with constant specific heats. **2** The process is reversible. **3** Kinetic and potential energy changes are negligible.

Properties The gas constant of nitrogen is R = 0.297 kJ/kg.K (Table A-1). The specific heat ratio of nitrogen is k = 1.4 (Table A-2).

Analysis (*a*) Isentropic compression:



$$\dot{W}_{\text{comp,in}} = \dot{m}\frac{kRT_1}{k-1}\left\{(P_2/P_1)^{(k-1)/k} - 1\right\}$$

or,

$$10 \text{ kJ/s} = \dot{m}\frac{(1.4)(0.297 \text{ kJ/kg}\cdot\text{K})(300 \text{ K})}{1.4-1}\left\{(480 \text{ kPa}/80 \text{ kPa})^{0.4/1.4} - 1\right\}$$

It yields

$$\dot{m} = \mathbf{0.048 \text{ kg/s}}$$

(*b*) Polytropic compression with n = 1.3:

$$\dot{W}_{\text{comp,in}} = \dot{m}\frac{nRT_1}{n-1}\left\{(P_2/P_1)^{(n-1)/n} - 1\right\}$$

or,

$$10 \text{ kJ/s} = \dot{m}\frac{(1.3)(0.297 \text{ kJ/kg}\cdot\text{K})(300 \text{ K})}{1.3-1}\left\{(480 \text{ kPa}/80 \text{ kPa})^{0.3/1.3} - 1\right\}$$

It yields

$$\dot{m} = \mathbf{0.051 \text{ kg/s}}$$

(*c*) Isothermal compression:

$$\dot{W}_{\text{comp,in}} = \dot{m}RT\ln\frac{P_1}{P_2} \longrightarrow 10 \text{ kJ/s} = \dot{m}(0.297 \text{ kJ/kg}\cdot\text{K})(300 \text{ K})\ln\left(\frac{480 \text{ kPa}}{80 \text{ kPa}}\right)$$

It yields

$$\dot{m} = \mathbf{0.063 \text{ kg/s}}$$

(*d*) Ideal two-stage compression with intercooling (n = 1.3): In this case, the pressure ratio across each stage is the same, and its value is determined to be

$$P_x = \sqrt{P_1P_2} = \sqrt{(80 \text{ kPa})(480 \text{ kPa})} = 196 \text{ kPa}$$

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

$$\dot{W}_{\text{comp,in}} = 2\dot{m}w_{\text{comp,I}} = 2\dot{m}\frac{nRT_1}{n-1}\left\{(P_x/P_1)^{(n-1)/n} - 1\right\}$$

or,

$$10 \text{ kJ/s} = 2\dot{m}\frac{(1.3)(0.297 \text{ kJ/kg}\cdot\text{K})(300 \text{ K})}{1.3-1}\left\{(196 \text{ kPa}/80 \text{ kPa})^{0.3/1.3} - 1\right\}$$

It yields

$$\dot{m} = \mathbf{0.056 \text{ kg/s}}$$

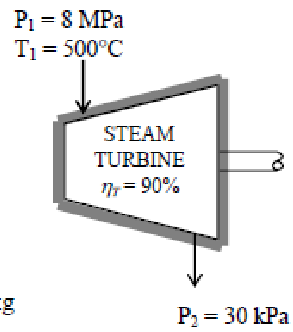
7–101C Describe the ideal process for an (*a*) adiabatic turbine, (*b*) adiabatic compressor, and (*c*) adiabatic nozzle, and define the isentropic efficiency for each device.

Solution The ideal process for all three devices is the reversible adiabatic (i.e., isentropic) process. The adiabatic efficiencies of these devices are defined as

$$\eta_T = \frac{\text{actual work output}}{\text{insentropic work output}},\ \eta_C = \frac{\text{insentropic work input}}{\text{actual work input}},\ \text{and } \eta_N = \frac{\text{actual exit kinetic energy}}{\text{insentropic exit kinetic energy}}$$

7–104 Steam enters an adiabatic turbine at 8 MPa and 500°C with a mass flow rate of 3 kg/s and leaves at 30 kPa. The isentropic efficiency of the turbine is 0.90. Neglecting the kinetic energy change of the steam, determine (*a*) the temperature at the turbine exit and (*b*) the power output of the turbine. *Answers: (a)* 69.1°C, *(b)* 3054 kW

Solution Steam enters an adiabatic turbine with an isentropic efficiency of 0.90 at a specified state with a specified mass flow rate, and leaves at a specified pressure. The turbine exit temperature and power output of the turbine 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.

Analysis (*a*) From the steam tables (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 8 \text{ MPa} \\ T_1 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3399.5 \text{ kJ/kg} \\ s_1 = 6.7266 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_{2s} = 30 \text{ kPa} \\ s_{2s} = s_1 \end{array}\right\} \begin{array}{l} x_{2s} = \dfrac{s_{2s} - s_f}{s_{fg}} = \dfrac{6.7266 - 0.9441}{6.8234} = 0.8475 \\ h_{2s} = h_f + x_{2s} h_{fg} = 289.27 + (0.8475)(2335.3) = 2268.3 \text{ kJ/kg} \end{array}$$

From the isentropic efficiency relation,

$$\eta_T = \frac{h_1 - h_{2a}}{h_1 - h_{2s}} \longrightarrow h_{2a} = h_1 - \eta_T (h_1 - h_{2s}) = 3399.5 - (0.9)(3399.5 - 2268.3) = 2381.4 \text{ kJ/kg}$$

Thus,

$$\left.\begin{array}{l} P_{2a} = 30 \text{ kPa} \\ h_{2a} = 2381.4 \text{ kJ/kg} \end{array}\right\} T_{2a} = T_{\text{sat@30 kPa}} = \mathbf{69.09°C}$$

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

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

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

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

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

Substituting,

$$\dot{W}_{\text{a,out}} = (3 \text{kg/s})(3399.5 - 2381.4) \text{ kJ/kg} = \mathbf{3054 \text{ kW}}$$

7–106 Steam enters an adiabatic turbine at 7 MPa, 600°C, and 80 m/s and leaves at 50 kPa, 150°C, and 140 m/s. If the power output of the turbine is 6 MW, determine (*a*) the mass flow rate of the steam flowing through the turbine and (*b*) the isentropic efficiency of the turbine. *Answers:* (*a*) 6.95 kg/s, (*b*) 73.4 percent

Solution Steam enters an adiabatic turbine at a specified state, and leaves at a specified state. The mass flow rate of the steam and the isentropic efficiency are to be determined.

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

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

$$\left.\begin{array}{l} P_1 = 7 \text{ MPa} \\ T_1 = 600°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3650.6 \text{ kJ/kg} \\ s_1 = 7.0910 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 50 \text{ kPa} \\ T_2 = 150°\text{C} \end{array}\right\} h_{2a} = 2780.2 \text{ kJ/kg}$$

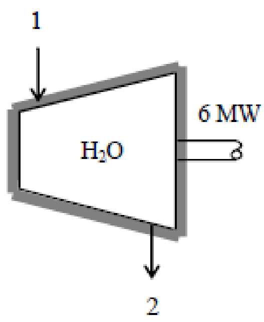
There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

$$\dot{W}_{a,out} = -\dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$



Substituting, the mass flow rate of the steam is determined to be

$$6000 \text{ kJ/s} = -\dot{m}\left(2780.2 - 3650.6 + \frac{(140 \text{ m/s})^2 - (80 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)\right)$$

$$\dot{m} = \mathbf{6.95} \textbf{ kg/s}$$

(*b*) The isentropic exit enthalpy of the steam and the power output of the isentropic turbine are

$$\left.\begin{array}{l} P_{2s} = 50 \text{ kPa} \\ s_{2s} = s_1 \end{array}\right\} \begin{array}{l} x_{2s} = \dfrac{s_{2s} - s_f}{s_{fg}} = \dfrac{7.0910 - 1.0912}{6.5019} = 0.9228 \\ h_{2s} = h_f + x_{2s}h_{fg} = 340.54 + (0.9228)(2304.7) = 2467.3 \text{ kJ/kg} \end{array}$$

and

$$\dot{W}_{s,out} = -\dot{m}\left(h_{2s} - h_1 + \left\{\left(V_2^2 - V_1^2\right)/2\right\}\right)$$

$$\dot{W}_{s,out} = -(6.95 \text{ kg/s})\left(2467.3 - 3650.6 + \frac{(140 \text{ m/s})^2 - (80 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)\right)$$

$$= 8174 \text{ kW}$$

Then the isentropic efficiency of the turbine becomes

$$\eta_T = \frac{\dot{W}_a}{\dot{W}_s} = \frac{6000 \text{ kW}}{8174 \text{ kW}} = 0.734 = \mathbf{73.4\%}$$

7–107 Argon gas enters an adiabatic turbine at 800°C and 1.5 MPa at a rate of 80 kg/min and exhausts at 200 kPa. If the power output of the turbine is 370 kW, determine the isentropic efficiency of the turbine.

Solution Argon enters an adiabatic turbine at a specified state with a specified mass flow rate, and leaves at a specified pressure. The isentropic efficiency of 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 device is adiabatic and thus heat transfer is negligible. **4** Argon is an ideal gas with constant specific heats.

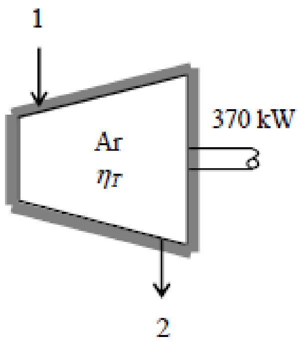
Properties The specific heat ratio of argon is k = 1.667. The constant pressure specific heat of argon is c_p = 0.5203 kJ/kg.K (Table A-2).

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

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

$$\dot{m}h_1 = \dot{W}_{s,\text{out}} + \dot{m}h_{2s} \quad (\text{since } \dot{Q} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{s,\text{out}} = \dot{m}(h_1 - h_{2s})$$



From the isentropic relations,

$$T_{2s} = T_1\left(\frac{P_{2s}}{P_1}\right)^{(k-1)/k} = (1073\text{ K})\left(\frac{200\text{ kPa}}{1500\text{ kPa}}\right)^{0.667/1.667} = 479\text{ K}$$

Then the power output of the isentropic turbine becomes

$$\dot{W}_{s,\text{out}} = \dot{m}c_p(T_1 - T_{2s}) = (80/60\text{ kg/min})(0.5203\text{ kJ/kg}\cdot\text{K})(1073 - 479) = 412.1\text{ kW}$$

Then the isentropic efficiency of the turbine is determined from

$$\eta_T = \frac{\dot{W}_a}{\dot{W}_s} = \frac{370\text{ kW}}{412.1\text{ kW}} = 0.898 = \mathbf{89.8\%}$$

7–109 Refrigerant-134a enters an adiabatic compressor as saturated vapor at 120 kPa at a rate of 0.3 m^3/min and exits at 1-MPa pressure. If the isentropic efficiency of the compressor is 80 percent, determine (*a*) the temperature of the refrigerant at the exit of the compressor and (*b*) the power input, in kW. Also, show the process on a *T*-*s* diagram with respect to saturation lines.

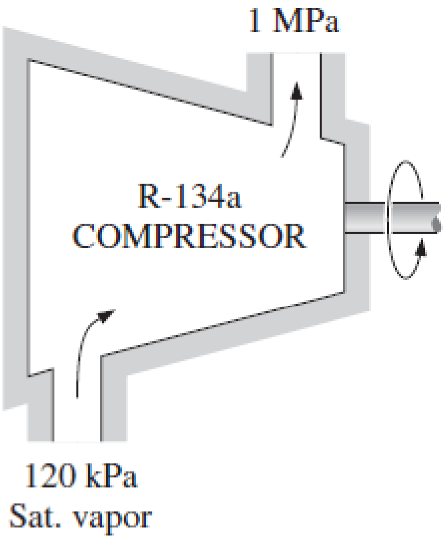
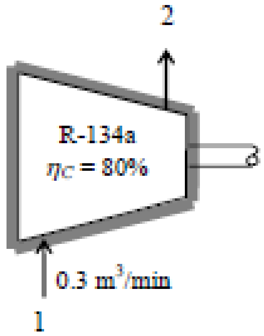


FIGURE P7–109

Solution Refrigerant-134a enters an adiabatic compressor with an isentropic efficiency of 0.80 at a specified state with a specified volume flow rate, and leaves at a specified pressure. The compressor exit temperature and power input to the compressor 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.

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



$$\left.\begin{array}{l} P_1 = 120\text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g@120\text{ kPa}} = 236.97\text{ kJ/kg} \\ s_1 = s_{g@120\text{ kPa}} = 0.94779\text{ kJ/kg}\cdot\text{K} \\ v_1 = v_{g@120\text{ kPa}} = 0.16212\text{ m}^3\text{/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1\text{ MPa} \\ s_{2s} = s_1 \end{array}\right\} h_{2s} = 281.21\text{ kJ/kg}$$

From the isentropic efficiency relation,

$$\eta_c = \frac{h_{2s} - h_1}{h_{2a} - h_1} \longrightarrow h_{2a} = h_1 + (h_{2s} - h_1)/\eta_c = 236.97 + (281.21 - 236.97)/0.80 = 292.26\text{ kJ/kg}$$

Thus,

$$\left.\begin{array}{l} P_{2a} = 1\text{ MPa} \\ h_{2a} = 292.26\text{ kJ/kg} \end{array}\right\} T_{2a} = 58.9^\circ\text{C}$$

(b) The mass flow rate of the refrigerant is determined from

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{0.3/60\text{ m}^3\text{/s}}{0.16212\text{ m}^3\text{/kg}} = 0.0308\text{ kg/s}$$

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 since mass crosses the boundary. The energy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\;\not{0}\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

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

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

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

Substituting, the power input to the compressor becomes,

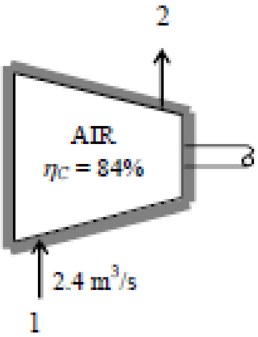
$$\dot{W}_{\text{a,in}} = (0.0308\text{ kg/s})(292.26 - 236.97)\text{kJ/kg} = \mathbf{1.70\ kW}$$

7–111 Air enters an adiabatic compressor at 100 kPa and 17°C at a rate of 2.4 m^3/s, and it exits at 257°C. The compressor has an isentropic efficiency of 84 percent. Neglecting the changes in kinetic and potential energies, determine (*a*) the exit pressure of air and (*b*) the power required to drive the compressor.

Solution Air enters an adiabatic compressor with an isentropic efficiency of 84% at a specified state, and leaves at a specified temperature. The exit pressure of air and the power input to the compressor are to be determined.

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

Properties The gas constant of air is $R = 0.287$ kPa.m^3/kg.K (Table A-1)



Analysis (*a*) From the air table (Table A-17),

$$T_1 = 290 \text{ K} \longrightarrow h_1 = 290.16 \text{ kJ/kg}, \quad P_{r1} = 1.2311$$
$$T_2 = 530 \text{ K} \longrightarrow h_{2a} = 533.98 \text{ kJ/kg}$$

From the isentropic efficiency relation $\eta_c = \dfrac{h_{2s} - h_1}{h_{2a} - h_1}$,

$$h_{2s} = h_1 + \eta_c (h_{2a} - h_1)$$
$$= 290.16 + (0.84)(533.98 - 290.16) = 495.0 \text{ kJ/kg} \longrightarrow P_{r_2} = 7.951$$

Then from the isentropic relation ,

$$\frac{P_2}{P_1} = \frac{P_{r_2}}{P_{r_1}} \longrightarrow P_2 = \left(\frac{P_{r_2}}{P_{r_1}}\right) P_1 = \left(\frac{7.951}{1.2311}\right)(100 \text{ kPa}) = \mathbf{646 \text{ kPa}}$$

(*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 since mass crosses the boundary. The energy balance for this steady-flow system can be expressed as

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

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

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

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

where $\quad \dot{m} = \dfrac{P_1 \dot{V}_1}{RT_1} = \dfrac{(100 \text{ kPa})(2.4 \text{ m}^3/\text{s})}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(290 \text{ K})} = 2.884 \text{ kg/s}$

Then the power input to the compressor is determined to be

$$\dot{W}_{a,in} = (2.884 \text{ kg/s})(533.98 - 290.16) \text{ kJ/kg} = \mathbf{703 \text{ kW}}$$

7–112 Air is compressed by an adiabatic compressor from 95 kPa and 27°C to 600 kPa and 277°C. Assuming variable specific heats and neglecting the changes in kinetic and potential energies, determine (*a*) the isentropic efficiency of the compressor and (*b*) the exit temperature of air if the process were reversible. *Answers:* (*a*) 81.9 percent, (*b*) 505.5 K

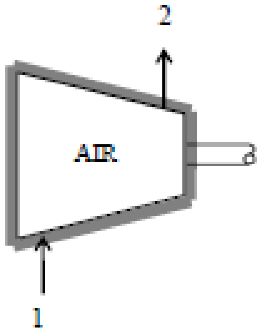
Solution Air is compressed by an adiabatic compressor from a specified state to another specified state. The isentropic efficiency of the compressor and the exit temperature of air for the isentropic case are to be determined.

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

Analysis (a) From the air table (Table A-17)

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

$$T_2 = 550\text{ K} \longrightarrow h_{2a} = 554.74\text{ kJ/kg}$$



From the isentropic relation,

$$P_{r_2} = \left(\frac{P_2}{P_1}\right)P_{r_1} = \left(\frac{600\text{ kPa}}{95\text{ kPa}}\right)(1.386) = 8.754 \longrightarrow h_{2s} = 508.72\text{ kJ/kg}$$

Then the isentropic efficiency becomes

$$\eta_C = \frac{h_{2s} - h_1}{h_{2a} - h_1} = \frac{508.72 - 300.19}{554.74 - 300.19} = 0.819 = \mathbf{81.9\%}$$

(b) If the process were isentropic, the exit temperature would be

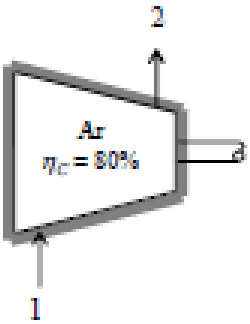
$$h_{2s} = 508.72\text{ kJ/kg} \longrightarrow T_{2s} = \mathbf{505.5\text{ K}}$$

7–113 Argon gas enters an adiabatic compressor at 150 kPa and 30°C with a velocity of 20 m/s, and it exits at 1400 kPa and 75 m/s. If the isentropic efficiency of the compressor is 80 percent, determine (*a*) the exit temperature of the argon and (*b*) the work input to the compressor.

Solution Argon enters an adiabatic compressor with an isentropic efficiency of 80% at a specified state, and leaves at a specified pressure. The exit temperature of argon and the work input to the compressor are to be determined.

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

Properties The specific heat ratio of argon is k = 1.667. The constant pressure specific heat of argon is c_p = 0.5203 kJ/kg.K (Table A-2).



Analysis (*a*) The isentropic exit temperature T_{2s} is determined from

$$T_{2s} = T_1\left(\frac{P_{2s}}{P_1}\right)^{(k-1)/k} = (303\text{ K})\left(\frac{1400\text{ kPa}}{150\text{ kPa}}\right)^{0.667/1.667} = 740.6\text{ K}$$

The actual kinetic energy change during this process is

$$\Delta ke_a = \frac{V_2^2 - V_1^2}{2} = \frac{(75\text{ m/s})^2 - (20\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 2.61\text{ kJ/kg}$$

The effect of kinetic energy on isentropic efficiency is very small. Therefore, we can take the kinetic energy changes for the actual and isentropic cases to be the same in efficiency calculations. From the isentropic efficiency relation, including the effect of kinetic energy,

$$\eta_c = \frac{w_s}{w_a} = \frac{(h_{2s}-h_1)+\Delta ke}{(h_{2a}-h_1)+\Delta ke} = \frac{c_p(T_{2s}-T_1)+\Delta ke_s}{c_p(T_{2a}-T_1)+\Delta ke_a} \longrightarrow 0.8 = \frac{0.5203(740.6-303)+2.61}{0.5203(T_{2a}-303)+2.61}$$

It yields T_{2a} = **851.2 K**

(*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 since mass crosses the boundary. The energy balance for this steady-flow system can be expressed as

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

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

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

$$\dot{W}_{a,in} = \dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right) \longrightarrow w_{a,in} = h_2 - h_1 + \Delta ke$$

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

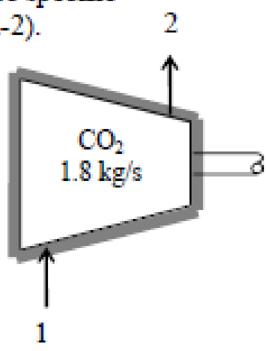
$$w_{a,in} = (0.5203\text{ kJ/kg}\cdot\text{K})(851.2-303)\text{K} + 2.61\text{ kJ/kg} = \mathbf{288\ kJ/kg}$$

7–114 Carbon dioxide enters an adiabatic compressor at 100 kPa and 300 K at a rate of 1.8 kg/s and exits at 600 kPa and 450 K. Neglecting the kinetic energy changes, determine the isentropic efficiency of the compressor.

Solution CO_2 gas is compressed by an adiabatic compressor from a specified state to another specified state. The isentropic efficiency of the compressor 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 device is adiabatic and thus heat transfer is negligible. **4** CO_2 is an ideal gas with constant specific heats.

Properties At the average temperature of (300 + 450)/2 = 375 K, the constant pressure specific heat and the specific heat ratio of CO_2 are $k = 1.260$ and $c_p = 0.917$ kJ/kg.K (Table A-2).



Analysis The isentropic exit temperature T_{2s} is

$$T_{2s} = T_1\left(\frac{P_{2s}}{P_1}\right)^{(k-1)/k} = (300\text{ K})\left(\frac{600\text{ kPa}}{100\text{ kPa}}\right)^{0.260/1.260} = 434.2\text{ K}$$

From the isentropic efficiency relation,

$$\eta_c = \frac{w_s}{w_a} = \frac{h_{2s}-h_1}{h_{2a}-h_1} = \frac{c_p(T_{2s}-T_1)}{c_p(T_{2a}-T_1)} = \frac{T_{2s}-T_1}{T_{2a}-T_1} = \frac{434.2-300}{450-300} = 0.895 = \mathbf{89.5\%}$$

7–115 Air enters an adiabatic nozzle at 400 kPa and 547°C with low velocity and exits at 240 m/s. If the isentropic efficiency of the nozzle is 90 percent, determine the exit temperature and pressure of the air.

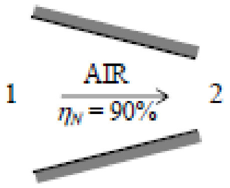
Solution Air is accelerated in a 90% efficient adiabatic nozzle from low velocity to a specified velocity. The exit temperature and pressure of the air are to be determined.

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

Analysis From the air table (Table A-17)

$$T_1 = 820\text{ K} \longrightarrow h_1 = 843.98\text{ kJ/kg},\ P_{r_1} = 52.59$$

There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the 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 as



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

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

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

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

Substituting, the exit temperature of air is determined to be

$$h_2 = 843.98\text{ kJ/kg} - \frac{(240\text{ m/s})^2 - 0}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 815.18\text{ kJ/kg}$$

From the air table we read $\quad T_{2a} = \mathbf{793.8\ K}$

From the isentropic efficiency relation $\eta_N = \dfrac{h_{2a} - h_1}{h_{2s} - h_1}$,

$$h_{2s} = h_1 + (h_{2a} - h_1)/\eta_N = 843.98 + (815.18 - 843.98)/(0.90) = 811.98\text{ kJ/kg} \longrightarrow P_{r_2} = 45.75$$

Then the exit pressure is determined from the isentropic relation to be

$$\frac{P_2}{P_1} = \frac{P_{r_2}}{P_{r_1}} \longrightarrow P_2 = \left(\frac{P_{r_2}}{P_{r_1}}\right)P_1 = \left(\frac{45.75}{52.59}\right)(400\text{ kPa}) = \mathbf{348\ kPa}$$

7–117 Hot combustion gases enter the nozzle of a turbojet engine at 260 kPa, 747°C, and 80 m/s, and they exit at a pressure of 85 kPa. Assuming an isentropic efficiency of 92 percent and treating the combustion gases as air, determine (*a*) the exit velocity and (*b*) the exit temperature. *Answers:* (*a*) 728.2 m/s, (*b*) 786.3 K

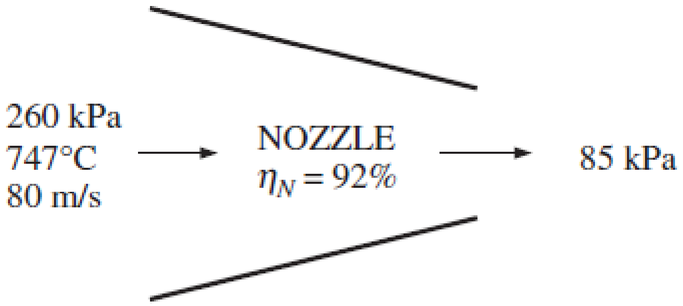
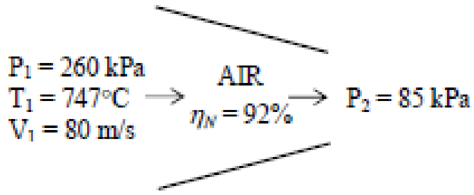


FIGURE P7–117

Solution Hot combustion gases are accelerated in a 92% efficient adiabatic nozzle from low velocity to a specified velocity. The exit velocity and the exit temperature are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Combustion gases can be treated as air that is an ideal gas with variable specific heats.

Analysis From the air table (Table A-17)



$$T_1 = 1020 \text{ K} \longrightarrow h_1 = 1068.89 \text{ kJ/kg}, \ P_{r_1} = 123.4$$

From the isentropic relation ,

$$P_{r_2} = \left(\frac{P_2}{P_1}\right) P_{r_1} = \left(\frac{85 \text{ kPa}}{260 \text{ kPa}}\right)(123.4) = 40.34 \longrightarrow h_{2s} = 783.92 \text{ kJ/kg}$$

There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system for the isentropic process can be expressed as

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

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

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

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

Then the isentropic exit velocity becomes

$$V_{2s} = \sqrt{V_1^2 + 2(h_1 - h_{2s})} = \sqrt{(80 \text{ m/s})^2 + 2(1068.89 - 783.92)\text{kJ/kg}\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)} = 759.2 \text{ m/s}$$

Therefore,

$$V_{2a} = \sqrt{\eta_N} V_{2s} = \sqrt{0.92}(759.2 \text{ m/s}) = \mathbf{728.2 \text{ m/s}}$$

The exit temperature of air is determined from the steady-flow energy equation,

$$h_{2a} = 1068.89 \text{ kJ/kg} - \frac{(728.2 \text{ m/s})^2 - (80 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) = 806.95 \text{ kJ/kg}$$

From the air table we read $\qquad T_{2a} = \mathbf{786.3 \text{ K}}$