Reversible Steady-Flow Work

7-87C The work associated with steady-flow devices is proportional to the specific volume of the gas. Cooling a gas during compression will reduce its specific volume, and thus the power consumed by the compressor.

7-88C Cooling the steam as it expands in a turbine will reduce its specific volume, and thus the work output of the turbine. Therefore, this is not a good proposal.

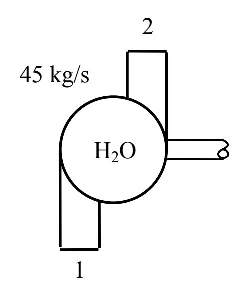
7-89C We would not support this proposal since the steady-flow work input to the pump is proportional to the specific volume of the liquid, and cooling will not affect the specific volume of a liquid significantly.

7-90 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,



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

Substituting,

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

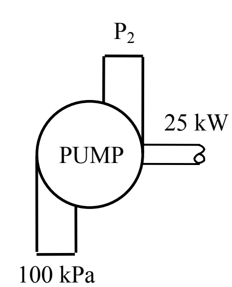
7-91 Liquid water is to be pumped by a 25-kW pump at a specified rate. The highest pressure the water can be pumped to is to be determined.

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 $v_1 = 0.001\text{ m}^3/\text{kg}$.

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 = $ **5100 kPa**

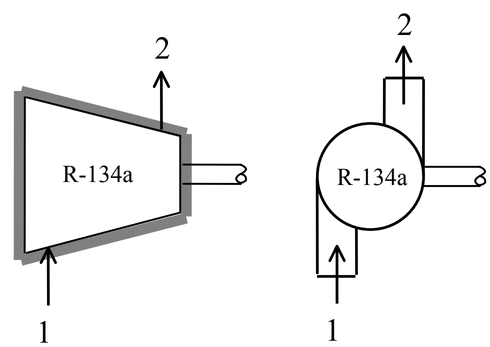
7-92E Saturated refrigerant-134a vapor is to be compressed reversibly to a specified pressure. The power input to the compressor is to be determined, and it is also to be compared to the work input for the liquid case.

Assumptions **1** Liquid refrigerant is an incompressible substance. **2** Kinetic and potential energy changes are negligible. **3** The process is reversible. **4** The compressor is adiabatic.

Analysis The compression process is reversible and adiabatic, and thus isentropic, $s_1 = s_2$. Then the properties of the refrigerant are (Tables A-11E through A-13E)

$$\left.\begin{array}{l} P_1 = 15 \text{ psia} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = 100.99 \text{ Btu/lbm} \\ s_1 = 0.22715 \text{ Btu/lbm} \cdot \text{R} \end{array}$$

$$\left.\begin{array}{l} P_1 = 80 \text{ psia} \\ s_2 = s_1 \end{array}\right\} h_2 = 115.80 \text{ Btu/lbm}$$



The work input to this isentropic compressor is determined from the steady-flow energy balance to be

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

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

$$\dot{W}_{in} + \dot{m}h_1 = \dot{m}h_2$$

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

Thus, $w_{in} = h_2 - h_1 = 115.80 - 100.99 =$ **14.8 Btu/lbm**

If the refrigerant were first condensed at constant pressure before it was compressed, we would use a pump to compress the liquid. In this case, the pump work input could be determined from the steady-flow work relation to be

$$w_{in} = \int_1^2 \boldsymbol{v}\, dP + \Delta ke^{\nearrow 0} + \Delta pe^{\nearrow 0} = \boldsymbol{v}_1 (P_2 - P_1)$$

where $\boldsymbol{v}_3 = \boldsymbol{v}_{f @ 15 \text{ psia}} = 0.01165 \text{ ft}^3\text{/lbm}$. Substituting,

$$w_{in} = (0.01165 \text{ ft}^3\text{/lbm})(80 - 15) \text{ psia} \left(\frac{1 \text{ Btu}}{5.4039 \text{ psia} \cdot \text{ft}^3} \right) = \mathbf{0.14\ Btu/lbm}$$

7-93 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.

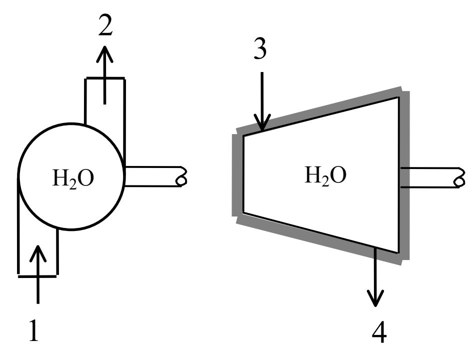
Properties The specific volume of saturated liquid water at 20 kPa is $v_1 = v_{f\,@\,20\text{ kPa}} = 0.001017\ \text{m}^3/\text{kg}$ (Table A-5).

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

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

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

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

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

Substituting,

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

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

$$w_{\text{pump,in}} = \int_1^2 v\,dP + \Delta ke^{\nearrow 0} + \Delta pe^{\nearrow 0} = v_1(P_2 - P_1)$$

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

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

Thus,

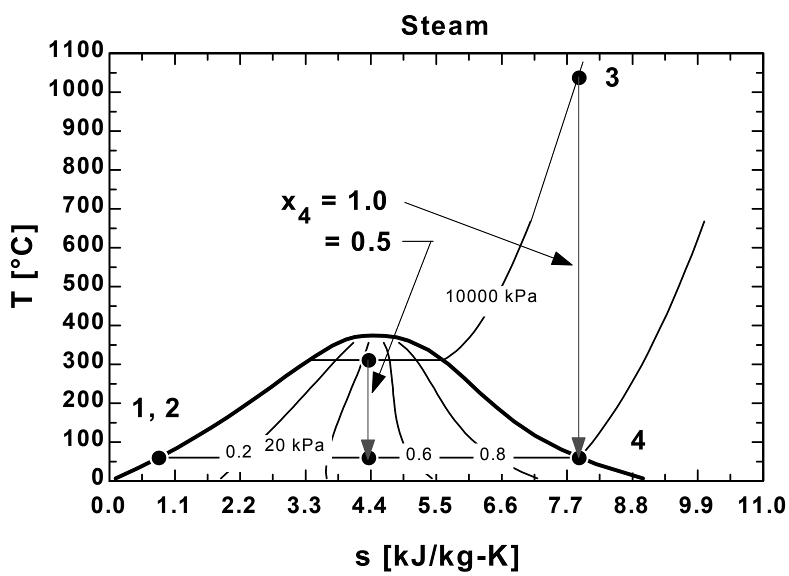
$$\frac{w_{\text{turb,out}}}{w_{\text{pump,in}}} = \frac{2098.3}{10.15} = \mathbf{206.7}$$

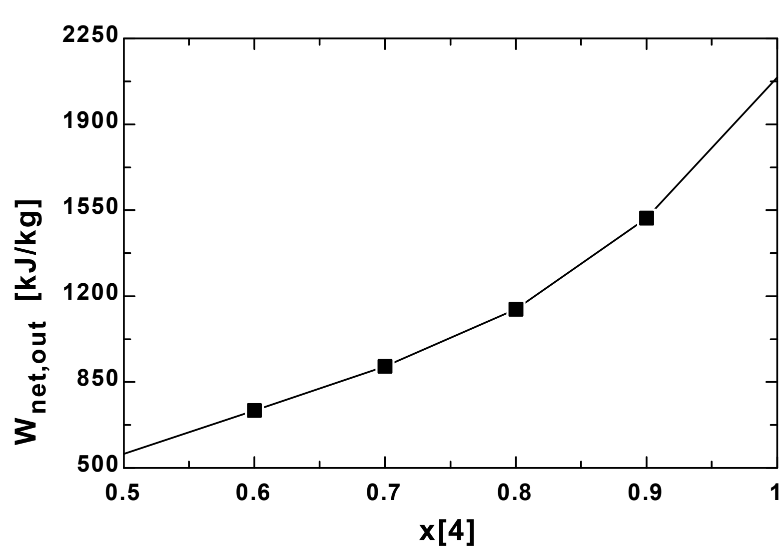
7-94 EES Problem 7-93 is reconsidered. The effect of the quality of the steam at the turbine exit on the net work output is to be investigated as the quality is varied from 0.5 to 1.0, and the net work output us to be plotted as a function of this quality.

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

```
"System: control volume for the pump and turbine"
"Property relation: Steam functions"
"Process: For Pump and Turbine: Steady state, steady flow, adiabatic, reversible or
isentropic"
"Since we don't know the mass, we write the conservation of energy per unit mass."
"Conservation of mass: m_dot[1]= m_dot[2]"

"Knowns:"
WorkFluid$ = 'Steam_IAPWS'
P[1] = 20  [kPa]
x[1] = 0
P[2] = 10000 [kPa]
x[4] = 1.0

"Pump Analysis:"
T[1]=temperature(WorkFluid$,P=P[1],x=0)
v[1]=volume(workFluid$,P=P[1],x=0)
h[1]=enthalpy(WorkFluid$,P=P[1],x=0)
s[1]=entropy(WorkFluid$,P=P[1],x=0)
s[2] = s[1]
h[2]=enthalpy(WorkFluid$,P=P[2],s=s[2])
T[2]=temperature(WorkFluid$,P=P[2],s=s[2])

"The Volume function has the same form for an ideal gas as for a real fluid."
v[2]=volume(WorkFluid$,T=T[2],p=P[2])
"Conservation of Energy - SSSF energy balance for pump"
"      -- neglect the change in potential energy, no  heat transfer:"
h[1]+W_pump = h[2]
"Also the work of pump can be obtained from the incompressible fluid, steady-flow result:"
W_pump_incomp = v[1]*(P[2] - P[1])

"Conservation of Energy - SSSF energy balance for turbine -- neglecting the change in potential
energy, no  heat transfer:"
P[4] = P[1]
P[3] = P[2]
h[4]=enthalpy(WorkFluid$,P=P[4],x=x[4])
s[4]=entropy(WorkFluid$,P=P[4],x=x[4])
T[4]=temperature(WorkFluid$,P=P[4],x=x[4])
s[3] = s[4]
h[3]=enthalpy(WorkFluid$,P=P[3],s=s[3])
T[3]=temperature(WorkFluid$,P=P[3],s=s[3])
h[3] = h[4] + W_turb
W_net_out  = W_turb - W_pump
```

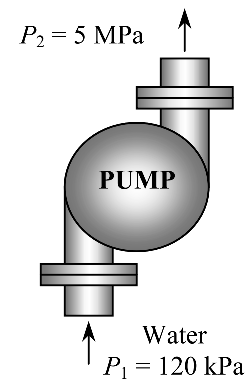
$W_{net,out}$ [kJ/kg]	W_{pump} [kJ/kg]	$W_{pump,incomp}$ [kJ/kg]	W_{turb} [kJ/kg]	x_4
557.1	10.13	10.15	567.3	0.5
734.7	10.13	10.15	744.8	0.6
913.6	10.13	10.15	923.7	0.7
1146	10.13	10.15	1156	0.8
1516	10.13	10.15	1527	0.9
2088	10.13	10.15	2098	1





7-95 Liquid water is pumped by a 70-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,

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

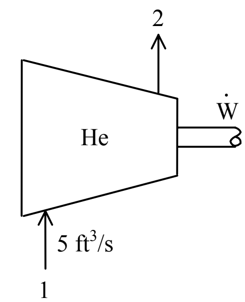
It yields

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

7-96E 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 = 2.6805 psia.ft^3/lbm.R = 0.4961 Btu/lbm.R. The specific heat ratio of helium is k = 1.667 (Table A-2E).



Analysis The mass flow rate of helium is

$$\dot{m} = \frac{P_1 \dot{V}_1}{RT_1} = \frac{(14 \text{ psia})(5 \text{ ft}^3/\text{s})}{(2.6805 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(530 \text{ R})} = 0.0493 \text{ lbm/s}$$

(a) Isentropic compression with k = 1.667:

$$\dot{W}_{\text{comp,in}} = \dot{m}\frac{kRT_1}{k-1}\left\{\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right\}$$
$$= (0.0493 \text{ lbm/s})\frac{(1.667)(0.4961 \text{ Btu/lbm}\cdot\text{R})(530 \text{ R})}{1.667-1}\left\{\left(\frac{120 \text{ psia}}{14 \text{ psia}}\right)^{0.667/1.667} - 1\right\}$$
$$= 44.11 \text{ Btu/s}$$
$$= \mathbf{62.4 \text{ hp}} \quad \text{since } 1 \text{ hp} = 0.7068 \text{ Btu/s}$$

(b) Polytropic compression with n = 1.2:

$$\dot{W}_{\text{comp,in}} = \dot{m}\frac{nRT_1}{n-1}\left\{\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1\right\}$$
$$= (0.0493 \text{ lbm/s})\frac{(1.2)(0.4961 \text{ Btu/lbm}\cdot\text{R})(530 \text{ R})}{1.2-1}\left\{\left(\frac{120 \text{ psia}}{14 \text{ psia}}\right)^{0.2/1.2} - 1\right\}$$
$$= 33.47 \text{ Btu/s}$$
$$= \mathbf{47.3 \text{ hp}} \quad \text{since } 1 \text{ hp} = 0.7068 \text{ Btu/s}$$

(c) Isothermal compression:

$$\dot{W}_{\text{comp,in}} = \dot{m}RT \ln\frac{P_2}{P_1} = (0.0493 \text{ lbm/s})(0.4961 \text{ Btu/lbm}\cdot\text{R})(530 \text{ R})\ln\frac{120 \text{ psia}}{14 \text{ psia}} = 27.83 \text{ Btu/s} = \mathbf{39.4 \text{ hp}}$$

(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

$$P_x = \sqrt{P_1 P_2} = \sqrt{(14 \text{ psia})(120 \text{ psia})} = 41.0 \text{ psia}$$

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

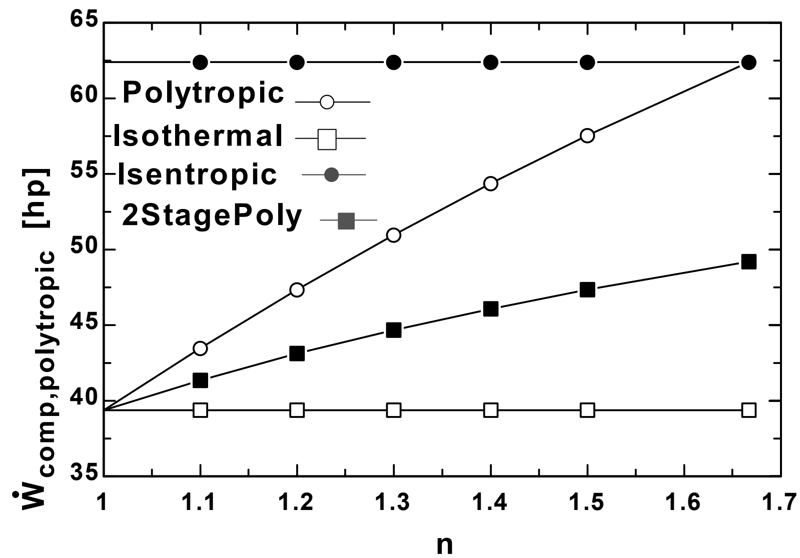
$$\dot{W}_{\text{comp,in}} = 2\dot{m}w_{\text{comp,I}} = 2\dot{m}\frac{nRT_1}{n-1}\left\{\left(\frac{P_x}{P_1}\right)^{(n-1)/n} - 1\right\}$$
$$= 2(0.0493 \text{ lbm/s})\frac{(1.2)(0.4961 \text{ Btu/lbm}\cdot\text{R})(530 \text{ R})}{1.2-1}\left\{\left(\frac{41 \text{ psia}}{14 \text{ psia}}\right)^{0.2/1.2} - 1\right\}$$
$$= 30.52 \text{ Btu/s}$$
$$= \mathbf{43.2 \text{ hp}} \quad \text{since } 1 \text{ hp} = 0.7068 \text{ Btu/s}$$

7-97E EES Problem 7-96E is reconsidered. The work of compression and entropy change of the helium is to be evaluated and plotted as functions of the polytropic exponent as it varies from 1 to 1.667.

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

```
Procedure FuncPoly(m_dot,k, R,
T1,P2,P1,n:W_dot_comp_polytropic,W_dot_comp_2stagePoly,Q_dot_Out_polytropic,Q_dot_Out
_2stagePoly)

If n =1 then
T2=T1
W_dot_comp_polytropic= m_dot*R*(T1+460)*ln(P2/P1)*convert(Btu/s,hp) "[hp]"
W_dot_comp_2stagePoly = W_dot_comp_polytropic "[hp]"
Q_dot_Out_polytropic=W_dot_comp_polytropic*convert(hp,Btu/s) "[Btu/s]"
Q_dot_Out_2stagePoly = Q_dot_Out_polytropic*convert(hp,Btu/s) "[Btu/s]"

Else
C_P = k*R/(k-1) "[Btu/lbm-R]"
T2=(T1+460)*((P2/P1)^((n+1)/n)-460)"[F]"
W_dot_comp_polytropic = m_dot*n*R*(T1+460)/(n-1)*((P2/P1)^((n-1)/n) -
1)*convert(Btu/s,hp)"[hp]"
Q_dot_Out_polytropic=W_dot_comp_polytropic*convert(hp,Btu/s)+m_dot*C_P*(T1-T2)"[Btu/s]"

Px=(P1*P2)^0.5
T2x=(T1+460)*((Px/P1)^((n+1)/n)-460)"[F]"
W_dot_comp_2stagePoly = 2*m_dot*n*R*(T1+460)/(n-1)*((Px/P1)^((n-1)/n) -
1)*convert(Btu/s,hp)"[hp]"
Q_dot_Out_2stagePoly=W_dot_comp_2stagePoly*convert(hp,Btu/s)+2*m_dot*C_P*(T1-
T2x)"[Btu/s]"

endif
END

R=0.4961[Btu/lbm-R]
k=1.667
n=1.2
P1=14 [psia]
T1=70 [F]
P2=120 [psia]
V_dot = 5 [ft^3/s]
P1*V_dot=m_dot*R*(T1+460)*convert(Btu,psia-ft^3)

W_dot_comp_isentropic = m_dot*k*R*(T1+460)/(k-1)*((P2/P1)^((k-1)/k) -
1)*convert(Btu/s,hp)"[hp]"
Q_dot_Out_isentropic = 0"[Btu/s]"
Call FuncPoly(m_dot,k, R,
T1,P2,P1,n:W_dot_comp_polytropic,W_dot_comp_2stagePoly,Q_dot_Out_polytropic,Q_dot_Out
_2stagePoly)

W_dot_comp_isothermal= m_dot*R*(T1+460)*ln(P2/P1)*convert(Btu/s,hp)"[hp]"
Q_dot_Out_isothermal = W_dot_comp_isothermal*convert(hp,Btu/s)"[Btu/s]"
```

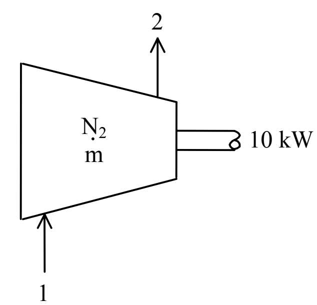
n	$W_{comp2StagePoly}$ [hp]	$W_{compisentropic}$ [hp]	$W_{compisothermal}$ [hp]	$W_{comppolytropic}$ [hp]
1	39.37	62.4	39.37	39.37
1.1	41.36	62.4	39.37	43.48
1.2	43.12	62.4	39.37	47.35
1.3	44.68	62.4	39.37	50.97
1.4	46.09	62.4	39.37	54.36
1.5	47.35	62.4	39.37	57.54
1.667	49.19	62.4	39.37	62.4



7-98 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

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

7-99 Water mist is to be sprayed into the air stream in the compressor to cool the air as the water evaporates and to reduce the compression power. The reduction in the exit temperature of the compressed air and the compressor power saved are to be determined.

Assumptions **1** Air is an ideal gas with variable specific heats. **2** The process is reversible. **3** Kinetic and potential energy changes are negligible. **3** Air is compressed isentropically. **4** Water vaporizes completely before leaving the compressor. **4** Air properties can be used for the air-vapor mixture.

Properties The gas constant of air is R = 0.287 kJ/kg.K (Table A-1). The specific heat ratio of air is k = 1.4. The inlet enthalpies of water and air are (Tables A-4 and A-17)

$$h_{w1} = h_{f@20°C} = 83.29 \text{ kJ/kg}, \; h_{fg@20°C} = 2453.9 \text{ kJ/kg and } h_{a1} = h_{@300\text{ K}} = 300.19 \text{ kJ/kg}$$

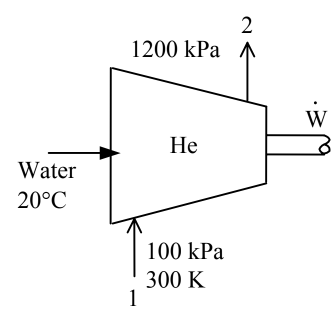
Analysis In the case of isentropic operation (thus no cooling or water spray), the exit temperature and the power input to the compressor are

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{(k-1)/k} \rightarrow T_2 = (300\text{ K})\left(\frac{1200\text{ kPa}}{100\text{ kPa}}\right)^{(1.4-1)/1.4} = 610.2\text{ K}$$

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

$$= (2.1\text{ kg/s})\frac{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(300\text{ K})}{1.4-1}\left\{(1200\text{ kPa}/100\text{ kPa})^{0.4/1.4} - 1\right\} = 654.3\text{ kW}$$

When water is sprayed, we first need to check the accuracy of the assumption that the water vaporizes completely in the compressor. In the limiting case, the compression will be isothermal at the compressor inlet temperature, and the water will be a saturated vapor. To avoid the complexity of dealing with two fluid streams and a gas mixture, we disregard water in the air stream (other than the mass flow rate), and assume air is cooled by an amount equal to the enthalpy change of water.



The rate of heat absorption of water as it evaporates at the inlet temperature completely is

$$\dot{Q}_{\text{cooling,max}} = \dot{m}_w h_{fg\,@\,20°C} = (0.2\text{ kg/s})(2453.9\text{ kJ/kg}) = 490.8\text{ kW}$$

The minimum power input to the compressor is

$$\dot{W}_{\text{comp,in,min}} = \dot{m}RT\ln\frac{P_2}{P_1} = (2.1\text{ kg/s})(0.287\text{ kJ/kg}\cdot\text{K})(300\text{ K})\ln\left(\frac{1200\text{ kPa}}{100\text{ kPa}}\right) = 449.3\text{ kW}$$

This corresponds to maximum cooling from the air since, at constant temperature, Δh = 0 and thus $\dot{Q}_{\text{out}} = \dot{W}_{\text{in}} = 449.3\text{ kW}$, which is close to 490.8 kW. Therefore, the assumption that all the water vaporizes is approximately valid. Then the reduction in required power input due to water spray becomes

$$\Delta\dot{W}_{\text{comp,in}} = \dot{W}_{\text{comp, isentropic}} - \dot{W}_{\text{comp, isothermal}} = 654.3 - 449.3 = \mathbf{205\ kW}$$

Discussion (can be ignored): At constant temperature, Δh = 0 and thus $\dot{Q}_{out} = \dot{W}_{in} = 449.3\text{ kW}$ corresponds to maximum cooling from the air, which is less than 490.8 kW. Therefore, the assumption that all the water vaporizes is only roughly valid. As an alternative, we can assume the compression process to be polytropic and the water to be a saturated vapor at the compressor exit temperature, and disregard the remaining liquid. But in this case there is not a unique solution, and we will have to select either the amount of water or the exit temperature or the polytropic exponent to obtain a solution. Of course we can also tabulate the results for different cases, and then make a selection.

Sample Analysis: We take the compressor exit temperature to be T_2 = 200°C = 473 K. Then,

$$h_{w2} = h_{g@200°C} = 2792.0 \text{ kJ/kg and } h_{a2} = h_{@473\text{ K}} = 475.3 \text{ kJ/kg}$$

Then,

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{(n-1)/n} \quad \rightarrow \frac{473\text{ K}}{300\text{ K}} = \left(\frac{1200\text{ kPa}}{100\text{ kPa}}\right)^{(n-1)/n} \quad \rightarrow n = 1.224$$

$$\dot{W}_{comp,in} = \dot{m}\frac{nRT_1}{n-1}\left\{(P_2/P_1)^{(n-1)/n} - 1\right\} = \dot{m}\frac{nR}{n-1}(T_2 - T_1)$$

$$= (2.1\text{ kg/s})\frac{(1.224)(0.287\text{ kJ/kg}\cdot\text{K})}{1.224-1}(473-300)\text{K} = 570\text{ kW}$$

Energy balance:

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

$$= 569.7\text{ kW} - (2.1\text{ kg/s})(475.3 - 300.19) = 202.0\text{ kW}$$

Noting that this heat is absorbed by water, the rate at which water evaporates in the compressor becomes

$$\dot{Q}_{\text{out,air}} = \dot{Q}_{\text{in,water}} = \dot{m}_w(h_{w2} - h_{w1}) \longrightarrow \dot{m}_w = \frac{\dot{Q}_{\text{in,water}}}{h_{w2} - h_{w1}} = \frac{202.0\text{ kJ/s}}{(2792.0 - 83.29)\text{ kJ/kg}} = 0.0746\text{ kg/s}$$

Then the reductions in the exit temperature and compressor power input become

$$\Delta T_2 = T_{2,\text{isentropic}} - T_{2,\text{water cooled}} = 610.2 - 473 = \mathbf{137.2°\,C}$$

$$\Delta\dot{W}_{comp,in} = \dot{W}_{comp,\text{isentropic}} - \dot{W}_{comp,\text{water cooled}} = 654.3 - 570 = \mathbf{84.3\ kW}$$

Note that selecting a different compressor exit temperature T_2 will result in different values.

7-100 A water-injected compressor is used in a gas turbine power plant. It is claimed that the power output of a gas turbine will increase when water is injected into the compressor because of the increase in the mass flow rate of the gas (air + water vapor) through the turbine. This, however, is **not necessarily right** since the compressed air in this case enters the combustor at a low temperature, and thus it absorbs much more heat. In fact, the cooling effect will most likely dominate and cause the cyclic efficiency to drop.

Isentropic Efficiencies of Steady-Flow Devices

7-101C The ideal process for all three devices is the reversible adiabatic (i.e., isentropic) process. The adiabatic efficiencies of these devices are defined as

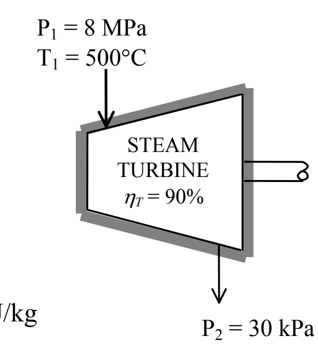
$$\eta_T = \frac{\text{actual work output}}{\text{insentropic work output}}, \ \eta_C = \frac{\text{insentropic work input}}{\text{actual work input}}, \text{ and } \eta_N = \frac{\text{actual exit kineticenergy}}{\text{insentropic exit kinetic energy}}$$

7-102C No, because the isentropic process is not the model or ideal process for compressors that are cooled intentionally.

7-103C Yes. Because the entropy of the fluid must increase during an actual adiabatic process as a result of irreversibilities. Therefore, the actual exit state has to be on the right-hand side of the isentropic exit state

7-104 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

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

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

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

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

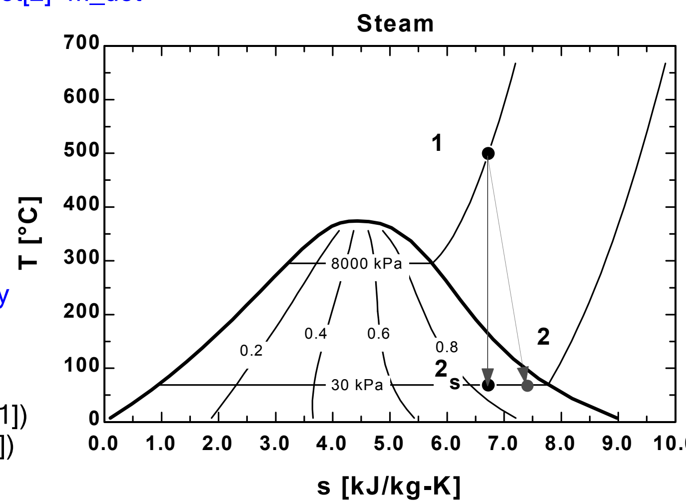
Substituting,

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

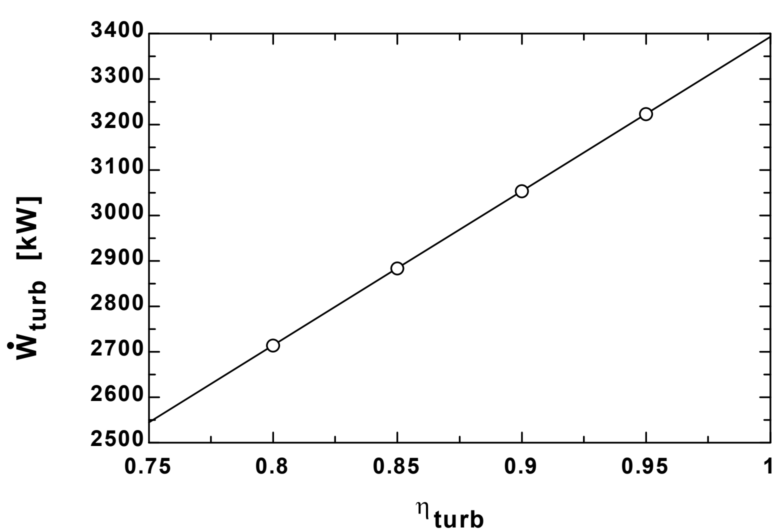
7-105 EES Problem 7-104 is reconsidered. The effect of varying the turbine isentropic efficiency from 0.75 to 1.0 on both the work done and the exit temperature of the steam are to be investigated, and the results are to be plotted.

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

```
"System: control volume for turbine"
"Property relation: Steam functions"
"Process: Turbine: Steady state, steady flow, adiabatic, reversible or isentropic"
"Since we don't know the mass, we write the conservation of energy per unit mass."
"Conservation of mass: m_dot[1]= m_dot[2]=m_dot"

"Knowns:"
WorkFluid$ = 'Steam_iapws'
m_dot = 3 [kg/s]
P[1] = 8000  [kPa]
T[1] = 500 [C]
P[2] = 30  [kPa]
"eta_turb = 0.9"

"Conservation of Energy - SSSF energy
balance for turbine -- neglecting the
change in potential energy, no  heat
transfer:"
h[1]=enthalpy(WorkFluid$,P=P[1],T=T[1])
s[1]=entropy(WorkFluid$,P=P[1],T=T[1])
T_s[1] = T[1]
s[2] =s[1]
s_s[2] = s[1]
h_s[2]=enthalpy(WorkFluid$,P=P[2],s=s_s[2])
T_s[2]=temperature(WorkFluid$,P=P[2],s=s_s[2])
eta_turb = w_turb/w_turb_s
h[1] = h[2] + w_turb
h[1] = h_s[2] + w_turb_s
T[2]=temperature(WorkFluid$,P=P[2],h=h[2])
W_dot_turb  = m_dot*w_turb
```



η_{turb}	W_{turb} [kW]
0.75	2545
0.8	2715
0.85	2885
0.9	3054
0.95	3224
1	3394



7-106 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.

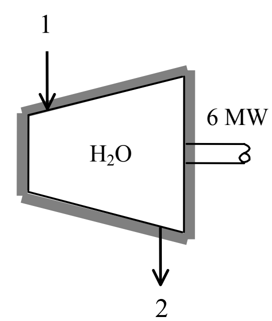
Analysis (*a*) From the steam tables (Tables A-4 and A-6),

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

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

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

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



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

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

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

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

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

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

(*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

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

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

$$= 8174\text{ kW}$$

Then the isentropic efficiency of the turbine becomes

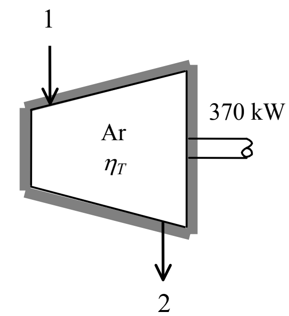
$$\eta_T = \frac{\dot{W}_a}{\dot{W}_s} = \frac{6000\text{ kW}}{8174\text{ kW}} = 0.734 = \mathbf{73.4\%}$$

7-107 Argon 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



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

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

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

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

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

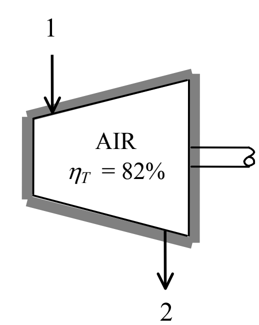
Then the isentropic efficiency of the turbine is determined from

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

7-108E Combustion gases enter an adiabatic gas turbine with an isentropic efficiency of 82% at a specified state, and leave at a specified pressure. The work output 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** Combustion gases can be treated as air that is an ideal gas with variable specific heats.

Analysis From the air table and isentropic relations,



$$T_1 = 2000\text{ R} \longrightarrow \begin{array}{l} h_1 = 504.71\text{ Btu/lbm} \\ P_{r_1} = 174.0 \end{array}$$

$$P_{r_2} = \left(\frac{P_2}{P_1}\right)P_{r_1} = \left(\frac{60\text{ psia}}{120\text{ psia}}\right)(174.0) = 87.0 \longrightarrow h_{2s} = 417.3\text{ Btu/lbm}$$

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 as

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

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

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

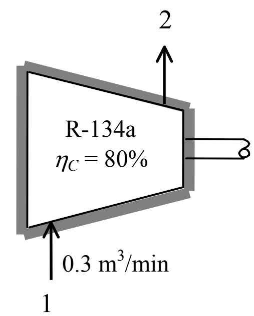
Noting that $w_a = \eta_T w_s$, the work output of the turbine per unit mass is determined from

$$w_a = (0.82)(504.71 - 417.3)\text{Btu/lbm} = \mathbf{71.7\ Btu/lbm}$$

7-109 [*Also solved by EES on enclosed CD*] 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),



$$\left.\begin{array}{l} P_1 = 120 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g@120 \text{ kPa}} = 236.97 \text{ kJ/kg} \\ s_1 = s_{g@120 \text{ kPa}} = 0.94779 \text{ kJ/kg}\cdot\text{K} \\ \boldsymbol{v}_1 = \boldsymbol{v}_{g@120 \text{ kPa}} = 0.16212 \text{ m}^3/\text{kg} \end{array}$$

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

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,

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

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

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

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

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

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

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

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

Substituting, the power input to the compressor becomes,

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

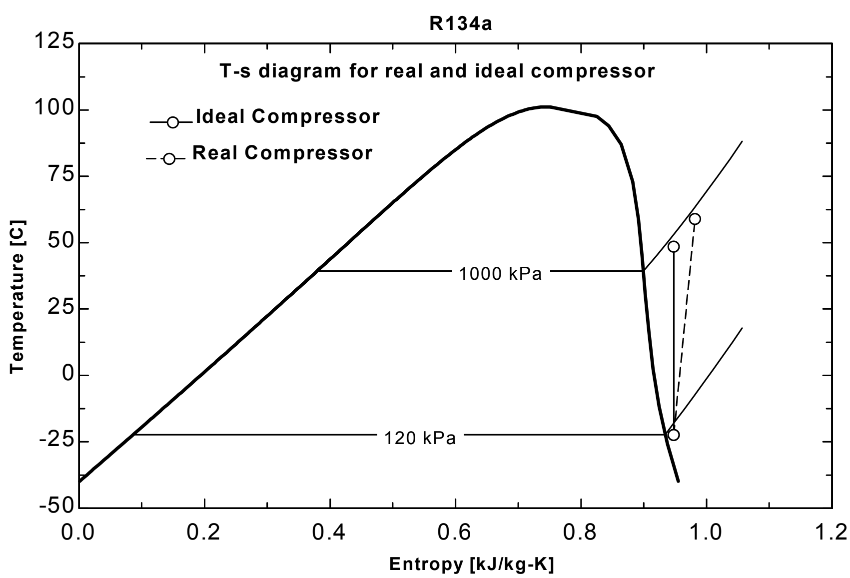
7-110 EES Problem 7-109 is reconsidered. The problem is to be solved by considering the kinetic energy and by assuming an inlet-to-exit area ratio of 1.5 for the compressor when the compressor exit pipe inside diameter is 2 cm.

Analysis The problem is solved using EES, and the solution is given below.

```
"Input Data from diagram window"
{P[1] = 120 "kPa"
P[2] = 1000 "kPa"
Vol_dot_1 = 0.3 "m^3/min"
Eta_c = 0.80 "Compressor adiabatic efficiency"
A_ratio = 1.5
d_2 = 2/100 "m"}

"System: Control volume containing the compressor, see the diagram window.
Property Relation:  Use the real fluid properties for R134a.
Process: Steady-state, steady-flow, adiabatic process."
Fluid$='R134a'
"Property Data for state 1"
T[1]=temperature(Fluid$,P=P[1],x=1)"Real fluid equ. at the sat. vapor state"
h[1]=enthalpy(Fluid$, P=P[1], x=1)"Real fluid equ. at the sat. vapor state"
s[1]=entropy(Fluid$, P=P[1], x=1)"Real fluid equ. at the sat. vapor state"
v[1]=volume(Fluid$, P=P[1], x=1)"Real fluid equ. at the sat. vapor state"

"Property Data for state 2"
s_s[1]=s[1]; T_s[1]=T[1] "needed for plot"
s_s[2]=s[1] "for the ideal, isentropic process across the compressor"
h_s[2]=ENTHALPY(Fluid$, P=P[2], s=s_s[2])"Enthalpy 2 at the isentropic state 2s and
pressure P[2]"
T_s[2]=Temperature(Fluid$, P=P[2], s=s_s[2])"Temperature of ideal state - needed only for
plot."
"Steady-state, steady-flow conservation of mass"
 m_dot_1 = m_dot_2
m_dot_1 = Vol_dot_1/(v[1]*60)
Vol_dot_1/v[1]=Vol_dot_2/v[2]
Vel[2]=Vol_dot_2/(A[2]*60)
A[2] = pi*(d_2)^2/4
A_ratio*Vel[1]/v[1] = Vel[2]/v[2] "Mass flow rate: = A*Vel/v, A_ratio = A[1]/A[2]"
A_ratio=A[1]/A[2]

"Steady-state, steady-flow conservation of energy, adiabatic compressor, see diagram
window"
m_dot_1*(h[1]+(Vel[1])^2/(2*1000)) + W_dot_c= m_dot_2*(h[2]+(Vel[2])^2/(2*1000))

"Definition of the compressor adiabatic efficiency, Eta_c=W_isen/W_act"
Eta_c = (h_s[2]-h[1])/(h[2]-h[1])

"Knowing h[2], the other properties at state 2 can be found."
v[2]=volume(Fluid$, P=P[2], h=h[2])"v[2] is found at the actual state 2, knowing P and h."
T[2]=temperature(Fluid$, P=P[2],h=h[2])"Real fluid equ. for T at the known outlet h and P."
s[2]=entropy(Fluid$, P=P[2], h=h[2])   "Real fluid equ. at the known outlet h and P."
T_exit=T[2]
"Neglecting the kinetic energies, the work is:"
m_dot_1*h[1] + W_dot_c_noke= m_dot_2*h[2]
```

SOLUTION

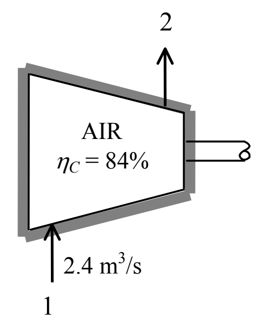
A[1]=0.0004712 [m^2]
A[2]=0.0003142 [m^2]
A_ratio=1.5
d_2=0.02 [m]
Eta_c=0.8
Fluid$='R134a'
h[1]=237 [kJ/kg]
h[2]=292.3 [kJ/kg]
h_s[2]=281.2 [kJ/kg]
m_dot_1=0.03084 [kg/s]
m_dot_2=0.03084 [kg/s]
P[1]=120.0 [kPa]
P[2]=1000.0 [kPa]
s[1]=0.9478 [kJ/kg-K]
s[2]=0.9816 [kJ/kg-K]
s_s[1]=0.9478 [kJ/kg-K]
s_s[2]=0.9478 [kJ/kg-K]
T[1]=-22.32 [C]
T[2]=58.94 [C]
T_exit=58.94 [C]
T_s[1]=-22.32 [C]
T_s[2]=48.58 [C]
Vol_dot_1=0.3 [m^3 /min]
Vol_dot_2=0.04244 [m^3 /min]
v[1]=0.1621 [m^3/kg]
v[2]=0.02294 [m^3/kg]
Vel[1]=10.61 [m/s]
Vel[2]=2.252 [m/s]
W_dot_c=1.704 [kW]
W_dot_c_noke=1.706 [kW]



7-111 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),

$$T_1 = 290 \text{ K} \longrightarrow h_1 = 290.16 \text{ kJ/kg}, \ P_{r1} = 1.2311$$

$$T_2 = 530 \text{ K} \longrightarrow h_{2a} = 533.98 \text{ kJ/kg}$$

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 ,

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

(*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

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

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

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

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

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

Then the power input to the compressor is determined to be

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

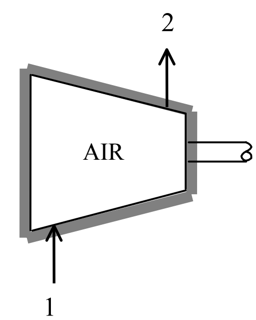
7-112 Air is compressed by an adiabatic compressor from a specified state to another specified state. The isentropic efficiency of the compressor and the exit temperature of air for the isentropic case are to be determined.

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),

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

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



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

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

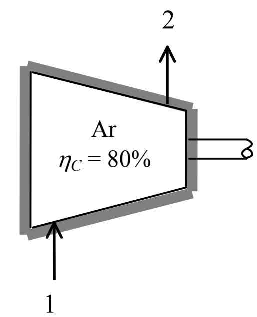
(b) If the process were isentropic, the exit temperature would be

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

7-113E 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 = 0.1253 Btu/lbm.R (Table A-2E).



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

$$T_{2s} = T_1\left(\frac{P_{2s}}{P_1}\right)^{(k-1)/k} = (550\text{ R})\left(\frac{200\text{ psia}}{20\text{ psia}}\right)^{0.667/1.667} = 1381.9\text{ R}$$

The actual kinetic energy change during this process is

$$\Delta ke_a = \frac{V_2^2 - V_1^2}{2} = \frac{(240\text{ ft/s})^2 - (60\text{ ft/s})^2}{2}\left(\frac{1\text{ Btu/lbm}}{25{,}037\text{ ft}^2/\text{s}^2}\right) = 1.08\text{ Btu/lbm}$$

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 same in efficiency calculations. From the isentropic efficiency relation, including the effect of kinetic energy,

$$\eta_C = \frac{w_s}{w_a} = \frac{(h_{2s} - h_1) + \Delta ke}{(h_{2a} - h_1) + \Delta ke} = \frac{c_p(T_{2s} - T_1) + \Delta ke_s}{c_p(T_{2a} - T_1) + \Delta ke_a} \longrightarrow 0.8 = \frac{0.1253(1381.9 - 550) + 1.08}{0.1253(T_{2a} - 550) + 1.08}$$

It yields $\quad T_{2a} = \mathbf{1592\ R}$

(*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

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

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

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

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

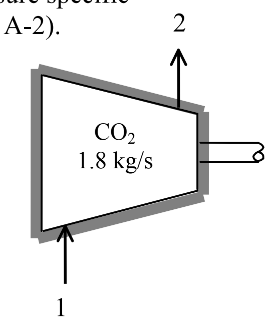
Substituting, the work input to the compressor is determined to be

$$w_{a,in} = (0.1253\text{ Btu/lbm}\cdot\text{R})(1592 - 550)\text{R} + 1.08\text{ Btu/lbm} = \mathbf{131.6\ Btu/lbm}$$

7-114 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,

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

7-115E 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.

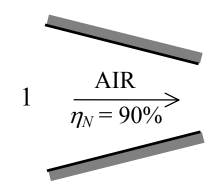
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Air is an ideal gas with variable specific heats.

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

$$T_1 = 1480\ \text{R} \longrightarrow h_1 = 363.89\ \text{Btu/lbm}, \quad P_{r_1} = 53.04$$

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

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



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

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

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

Substituting, the exit temperature of air is determined to be

$$h_2 = 363.89\ \text{kJ/kg} - \frac{(800\ \text{ft/s})^2 - 0}{2}\left(\frac{1\ \text{Btu/lbm}}{25{,}037\ \text{ft}^2/\text{s}^2}\right) = 351.11\ \text{Btu/lbm}$$

From the air table we read $\quad T_{2a}$ = **1431.3 R**

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

$$h_{2s} = h_1 + (h_{2a}-h_1)/\eta_N = 363.89 + (351.11-363.89)/(0.90) = 349.69\ \text{Btu/lbm} \longrightarrow P_{r_2} = 46.04$$

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

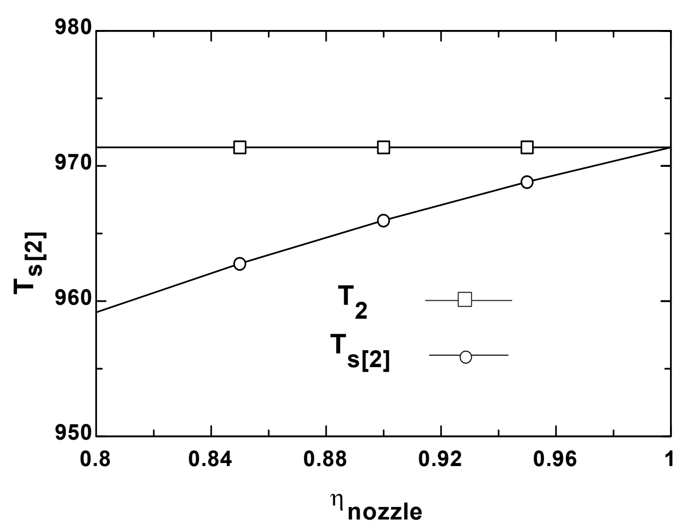
$$\frac{P_2}{P_1} = \frac{P_{r_2}}{P_{r_1}} \longrightarrow P_2 = \left(\frac{P_{r_2}}{P_{r_1}}\right)P_1 = \left(\frac{46.04}{53.04}\right)(60\ \text{psia}) = \mathbf{52.1\ psia}$$

7-116E EES Problem 7-115E is reconsidered. The effect of varying the nozzle isentropic efficiency from 0.8 to 1.0 on the exit temperature and pressure of the air is to be investigated, and the results are to be plotted.

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

```
"Knowns:"
WorkFluid$ = 'Air'
P[1] = 60  [psia]
T[1] = 1020 [F]
Vel[2] = 800 [ft/s]
Vel[1] = 0 [ft/s]
eta_nozzle = 0.9

"Conservation of Energy - SSSF energy balance for turbine -- neglecting the change in potential
energy, no  heat transfer:"
h[1]=enthalpy(WorkFluid$,T=T[1])
s[1]=entropy(WorkFluid$,P=P[1],T=T[1])
T_s[1] = T[1]
s[2] =s[1]
s_s[2] = s[1]
h_s[2]=enthalpy(WorkFluid$,T=T_s[2])
T_s[2]=temperature(WorkFluid$,P=P[2],s=s_s[2])
eta_nozzle = ke[2]/ke_s[2]
ke[1] = Vel[1]^2/2
ke[2]=Vel[2]^2/2
h[1]+ke[1]*convert(ft^2/s^2,Btu/lbm) = h[2] + ke[2]*convert(ft^2/s^2,Btu/lbm)
h[1] +ke[1]*convert(ft^2/s^2,Btu/lbm) = h_s[2] + ke_s[2]*convert(ft^2/s^2,Btu/lbm)
T[2]=temperature(WorkFluid$,h=h[2])
P_2_answer = P[2]
T_2_answer = T[2]
```

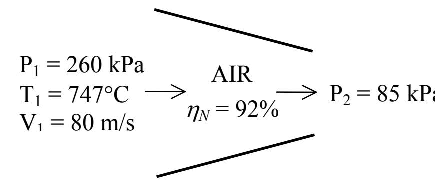
η_{nozzle}	P_2 [psia	T_2 [F	$T_{s,2}$ [F]
0.8	51.09	971.4	959.2
0.85	51.58	971.4	962.8
0.9	52.03	971.4	966
0.95	52.42	971.4	968.8
1	52.79	971.4	971.4



7-117 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

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

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

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

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

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 $T_{2a} = \mathbf{786.3 \text{ K}}$