Second-Law Analysis of Control Volumes

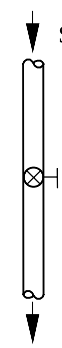
8-54 Steam is throttled from a specified state to a specified pressure. The wasted work potential during this throttling process is to be determined.

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 temperature of the surroundings is given to be 25°C. **4** Heat transfer is negligible.

Properties The properties of steam before and after the throttling process are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 8 \text{ MPa} \\ T_1 = 450°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3273.3 \text{ kJ/kg} \\ s_1 = 6.5579 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 6 \text{ MPa} \\ h_2 = h_1 \end{array}\right\} s_2 = 6.6806 \text{ kJ/kg}\cdot\text{K}$$



Analysis The wasted work potential is equivalent to the exergy destruction (or irreversibility). It can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ where the entropy generation is determined from an entropy balance on the device, which is an adiabatic steady-flow system,

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

$$\dot{m}s_1 - \dot{m}s_2 + \dot{S}_{\text{gen}} = 0 \rightarrow \dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1) \quad \text{or} \quad s_{\text{gen}} = s_2 - s_1$$

Substituting,

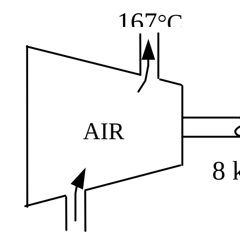
$$x_{\text{destroyed}} = T_0 s_{\text{gen}} = T_0 (s_2 - s_1) = (298 \text{ K})(6.6806 - 6.5579)\text{kJ/kg}\cdot\text{K} = \mathbf{36.6 \ kJ/kg}$$

Discussion Note that 36.6 kJ/kg of work potential is wasted during this throttling process.

8-55 [*Also solved by EES on enclosed CD***]** Air is compressed steadily by an 8-kW compressor from a specified state to another specified state. The increase in the exergy of air and the rate of exergy destruction are to be determined.

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

Properties The gas constant of air is R = 0.287 kJ/kg.K (Table A-1). From the air table (Table A-17)



$$T_1 = 290\text{ K} \longrightarrow \begin{array}{l} h_1 = 290.16\text{ kJ/kg} \\ s_1^{o} = 1.66802\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$T_2 = 440\text{ K} \longrightarrow \begin{array}{l} h_2 = 441.61\text{ kJ/kg} \\ s_2^{o} = 2.0887\text{ kJ/kg}\cdot\text{K} \end{array}$$

Analysis The increase in exergy is the difference between the exit and inlet flow exergies,

$$\begin{aligned} \text{Increase in exergy} &= \psi_2 - \psi_1 \\ &= [(h_2 - h_1) + \Delta ke^{\nearrow 0} + \Delta pe^{\nearrow 0} - T_0(s_2 - s_1)] \\ &= (h_2 - h_1) - T_0(s_2 - s_1) \end{aligned}$$

where

$$\begin{aligned} s_2 - s_1 &= (s_2^{o} - s_1^{o}) - R\ln\frac{P_2}{P_1} \\ &= (2.0887 - 1.66802)\text{kJ/kg}\cdot\text{K} - (0.287\text{ kJ/kg}\cdot\text{K})\ln\frac{600\text{ kPa}}{100\text{ kPa}} \\ &= -0.09356\text{ kJ/kg}\cdot\text{K} \end{aligned}$$

Substituting,

$$\begin{aligned} \text{Increase in exergy} &= \psi_2 - \psi_1 \\ &= \left[(441.61 - 290.16)\text{kJ/kg} - (290\text{ K})(-0.09356\text{ kJ/kg}\cdot\text{K})\right] \\ &= \mathbf{178.6\ kJ/kg} \end{aligned}$$

Then the reversible power input becomes

$$\dot{W}_{\text{rev,in}} = \dot{m}(\psi_2 - \psi_1) = (2.1/60\text{ kg/s})(178.6\text{ kJ/kg}) = \mathbf{6.25\ kW}$$

(*b*) The rate of exergy destruction (or irreversibility) is determined from its definition,

$$\dot{X}_{\text{destroyed}} = \dot{W}_{\text{in}} - \dot{W}_{\text{rev,in}} = 8 - 6.25 = \mathbf{1.75\ kW}$$

Discussion Note that 1.75 kW of power input is wasted during this compression process.

8-56 EES Problem 8-55 is reconsidered. The problem is to be solved and the actual heat transfer, its direction, the minimum power input, and the compressor second-law efficiency are to be determined.

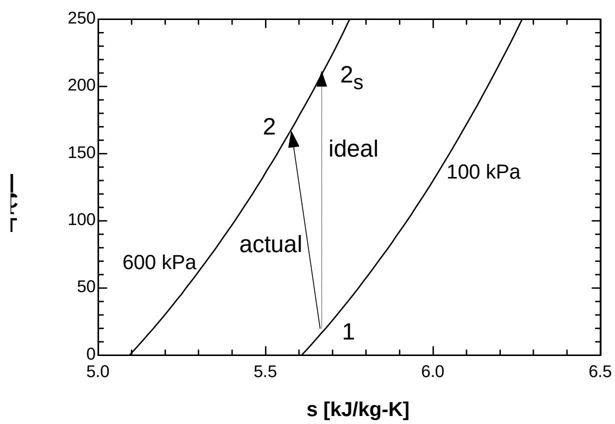
Analysis The problem is solved using EES, and the solution is given below.

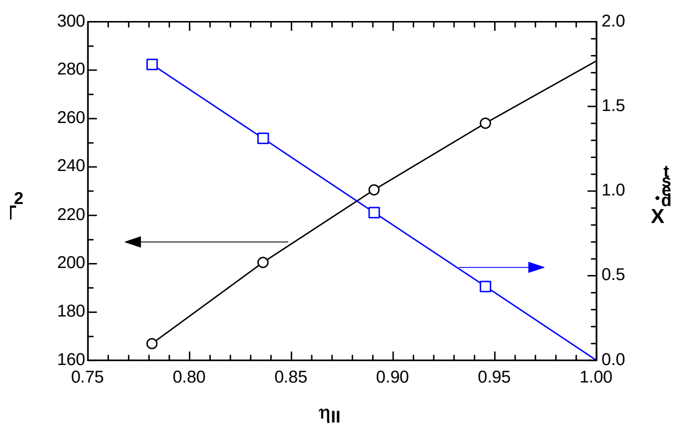
```
Function Direction$(Q)
If Q<0 then Direction$='out' else Direction$='in'
end
Function Violation$(eta)
If eta>1 then Violation$='You have violated the 2nd Law!!!!!' else Violation$=''
end

{"Input Data from the Diagram Window"
T_1=17 [C]
P_1=100 [kPa]
W_dot_c = 8 [kW]
P_2=600 [kPa]
S_dot_gen=0
Q_dot_net=0}
{"Special cases"
T_2=167 [C]
m_dot=2.1 [kg/min]}
T_o=T_1
P_o=P_1
m_dot_in=m_dot*Convert(kg/min, kg/s)
"Steady-flow conservation of mass"
m_dot_in = m_dot_out
"Conservation of energy for steady-flow is:"
E_dot_in - E_dot_out = DELTAE_dot
DELTAE_dot = 0
E_dot_in=Q_dot_net + m_dot_in*h_1 +W_dot_c
"If Q_dot_net < 0, heat is transferred from the compressor"
E_dot_out= m_dot_out*h_2
h_1 =enthalpy(air,T=T_1)
h_2 = enthalpy(air, T=T_2)
W_dot_net=-W_dot_c
W_dot_rev=-m_dot_in*(h_2 - h_1 -(T_1+273.15)*(s_2-s_1))
"Irreversibility, entropy generated, second law efficiency, and exergy destroyed:"
s_1=entropy(air, T=T_1,P=P_1)
s_2=entropy(air,T=T_2,P=P_2)
s_2s=entropy(air,T=T_2s,P=P_2)
s_2s=s_1"This yields the isentropic T_2s for an isentropic process bewteen T_1, P_1 and
P_2"I_dot=(T_o+273.15)*S_dot_gen"Irreversiblility for the Process, KW"
S_dot_gen=(-Q_dot_net/(T_o+273.15) +m_dot_in*(s_2-s_1)) "Entropy generated, kW"
Eta_II=W_dot_rev/W_dot_net"Definition of compressor second law efficiency, Eq. 7_6"
h_o=enthalpy(air,T=T_o)
s_o=entropy(air,T=T_o,P=P_o)
Psi_in=h_1-h_o-(T_o+273.15)*(s_1-s_o)  "availability function at state 1"
Psi_out=h_2-h_o-(T_o+273.15)*(s_2-s_o) "availability function at state 2"
X_dot_in=Psi_in*m_dot_in
X_dot_out=Psi_out*m_dot_out
DELTAX_dot=X_dot_in-X_dot_out
"General Exergy balance for a steady-flow system, Eq. 7-47"
(1-(T_o+273.15)/(T_o+273.15))*Q_dot_net-W_dot_net+m_dot_in*Psi_in - m_dot_out*Psi_out
=X_dot_dest
"For the Diagram Window"
```

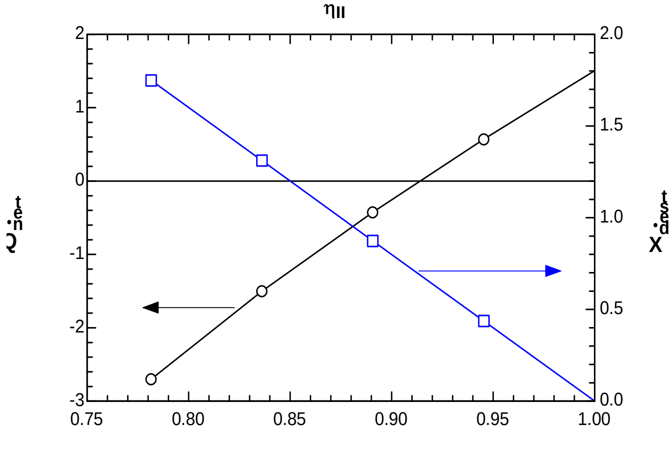
Text$=Direction$(Q_dot_net)
Text2$=Violation$(Eta_II)

η_{II}	I [kW]	X_{dest} [kW]	T_{2s} [C]	T_2 [C]	Q_{net} [kW]
0.7815	1.748	1.748	209.308	167	-2.7
0.8361	1.311	1.311	209.308	200.6	-1.501
0.8908	0.874	0.874	209.308	230.5	-0.4252
0.9454	0.437	0.437	209.308	258.1	0.5698
1	1.425E-13	5.407E-15	209.308	283.9	1.506

How can entropy decrease?







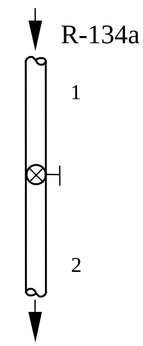
8-57 Refrigerant-124a is throttled from a specified state to a specified pressure. The reversible work and the exergy destroyed during this process are to be determined.

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

Properties The properties of R-134a before and after the throttling process are (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_1 = 1\text{ MPa} \\ T_1 = 100°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 335.06\text{ kJ/kg} \\ s_1 = 1.1031\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 0.8\text{ MPa} \\ h_2 = h_1 \end{array}\right\} s_2 = 1.1198\text{ kJ/kg}\cdot\text{K}$$



Analysis The exergy destruction (or irreversibility) can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ where the entropy generation is determined from an entropy balance on the system, which is an adiabatic steady-flow device,

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

$$\dot{m}s_1 - \dot{m}s_2 + \dot{S}_{\text{gen}} = 0 \;\rightarrow\; \dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1) \quad \text{or} \quad s_{\text{gen}} = s_2 - s_1$$

Substituting,

$$x_{\text{destroyed}} = T_0 s_{\text{gen}} = T_0(s_2 - s_1) = (303\text{ K})(1.1198 - 1.1031)\text{kJ/kg}\cdot\text{K} = \mathbf{5.04\ kJ/kg}$$

This process involves no actual work, and thus the reversible work and irreversibility are identical,

$$x_{\text{destroyed}} = w_{\text{rev,out}} - w_{\text{act,out}}^{\nearrow 0} \;\rightarrow\; w_{\text{rev,out}} = x_{\text{destroyed}} = \mathbf{5.04\ kJ/kg}$$

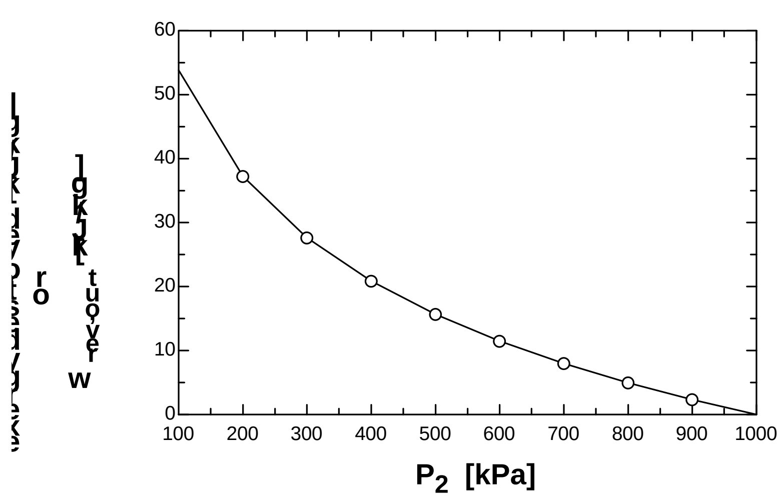
Discussion Note that 5.04 kJ/kg of work potential is wasted during this throttling process.

8-58 EES Problem 8-57 is reconsidered. The effect of exit pressure on the reversible work and exergy destruction is to be investigated.

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

```
T_1=100"[C]"
P_1=1000"[kPa]"
{P_2=800"[kPa]"}
T_o=298"[K]"
"Steady-flow conservation of mass"
"m_dot_in = m_dot_out"
"Conservation of energy for steady-flow per unit mass is:"
e_in - e_out = DELTAe
DELTAe = 0"[kJ/kg]"
E_in=h_1"[kJ/kg]"
E_out= h_2 "[kJ/kg]"
h_1 =enthalpy(R134a,T=T_1,P=P_1) "[kJ/kg]"
T_2 = temperature(R134a, P=P_2,h=h_2) "[C]"
"Irreversibility, entropy generated,  and exergy destroyed:"
s_1=entropy(R134a, T=T_1,P=P_1)"[kJ/kg-K]"
s_2=entropy(R134a,P=P_2,h=h_2)"[kJ/kg-K]"
I=T_o*s_gen"[kJ/kg]"     "Irreversiblility for the Process, KJ/kg"
s_gen=s_2-s_1"]kJ/kg-K]" "Entropy generated, kW"
x_destroyed = I"[kJ/kg]"
w_rev_out=x_destroyed"[kJ/kg]"
```

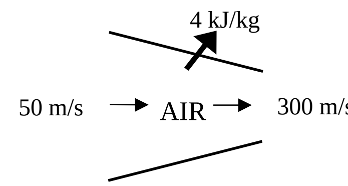
P_2 [kPa]	$w_{rev,out}$ [kJ/kg]	$x_{destroyed}$ [kJ/kg]
100	53.82	53.82
200	37.22	37.22
300	27.61	27.61
400	20.86	20.86
500	15.68	15.68
600	11.48	11.48
700	7.972	7.972
800	4.961	4.961
900	2.33	2.33
1000	-4.325E-10	-4.325E-10



8-59 Air is accelerated in a nozzle while losing some heat to the surroundings. The exit temperature of air and the exergy destroyed during the process are to be determined.

Assumptions **1** Air is an ideal gas with variable specific heats. **2** The nozzle operates steadily.

Properties The gas constant of air is R = 0.287 kJ/kg.K (Table A-1). The properties of air at the nozzle inlet are (Table A-17)



$$T_1 = 360\ \text{K} \longrightarrow \begin{array}{l} h_1 = 360.58\ \text{kJ/kg} \\ s_1^o = 1.88543\ \text{kJ/kg}\cdot\text{K} \end{array}$$

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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0 \longrightarrow \dot{E}_{in} = \dot{E}_{out} \longrightarrow \dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + \text{V}_2^2/2) + \dot{Q}_{out}$$

or $$0 = q_{out} + h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}$$

Therefore,

$$h_2 = h_1 - q_{out} - \frac{V_2^2 - V_1^2}{2} = 360.58 - 4 - \frac{(300\ \text{m/s})^2 - (50\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = 312.83\ \text{kJ/kg}$$

At this h_2 value we read, from Table A-17, $T_2 = \mathbf{312.5\ K = 39.5°C}$ and $s_2^0 = 1.74302\ \text{kJ/kg}\cdot\text{K}$

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

$$\underbrace{\dot{S}_{in} - \dot{S}_{out}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{gen}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{system}^{\nearrow 0}}_{\substack{\text{Rate of change}\\ \text{of entropy}}} = 0 \rightarrow \dot{m}s_1 - \dot{m}s_2 - \frac{\dot{Q}_{out}}{T_{b,surr}} + \dot{S}_{gen} = 0 \rightarrow \dot{S}_{gen} = \dot{m}(s_2 - s_1) + \frac{\dot{Q}_{out}}{T_{surr}}$$

where

$$\Delta s_{air} = s_2^o - s_1^o - R\ln\frac{P_2}{P_1} = (1.74302 - 1.88543)\text{kJ/kg}\cdot\text{K} - (0.287\ \text{kJ/kg}\cdot\text{K})\ln\frac{95\ \text{kPa}}{300\ \text{kPa}} = 0.1876\ \text{kJ/kg}\cdot\text{K}$$

Substituting, the entropy generation and exergy destruction per unit mass of air are determined to be

$$x_{destroyed} = T_0 s_{gen} = T_{surr} s_{gen} = T_0\left(s_2 - s_1 + \frac{q_{surr}}{T_{surr}}\right) = (290\ \text{K})\left(0.1876\ \text{kJ/kg}\cdot\text{K} + \frac{4\ \text{kJ/kg}}{290\ \text{K}}\right) = \mathbf{58.4\ kJ/kg}$$

Alternative solution The exergy destroyed during a process can be determined from an exergy balance applied on the *extended system* that includes the device and its immediate surroundings so that the boundary temperature of the extended system is environment temperature T_0 (or T_{surr}) at all times. Noting that exergy transfer with heat is zero when the temperature at the point of transfer is the environment temperature, the exergy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{X}_{\text{in}}-\dot{X}_{\text{out}}}_{\substack{\text{Rate of net exergy transfer}\\ \text{by heat, work, and mass}}} - \underbrace{\dot{X}_{\text{destroyed}}}_{\substack{\text{Rate of exergy}\\ \text{destruction}}} = \underbrace{\Delta\dot{X}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change}\\ \text{of exergy}}} = 0 \rightarrow \dot{X}_{\text{destroyed}} = \dot{X}_{in}-\dot{X}_{out} = \dot{m}\psi_1 - \dot{m}\psi_2 = \dot{m}(\psi_1-\psi_2)$$

$$= \dot{m}[(h_1-h_2)-T_0(s_1-s_2)-\Delta ke-\Delta pe^{\nearrow 0}] = \dot{m}[T_0(s_2-s_1)-(h_2-h_1+\Delta ke)]$$

$$= \dot{m}[T_0(s_2-s_1)+q_{\text{out}}] \quad \text{since, from energy balance,} \; -q_{\text{out}} = h_2-h_1+\Delta ke$$

$$= T_0\left(\dot{m}(s_2-s_1)+\frac{\dot{Q}_{\text{out}}}{T_0}\right) = T_0\dot{S}_{\text{gen}}$$

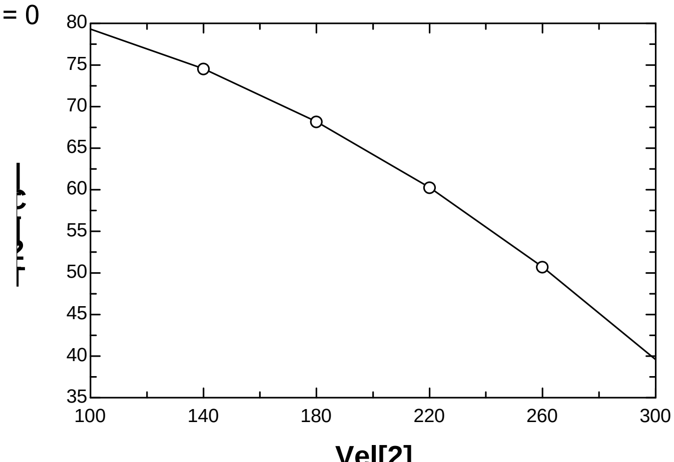
Therefore, the two approaches for the determination of exergy destruction are identical.

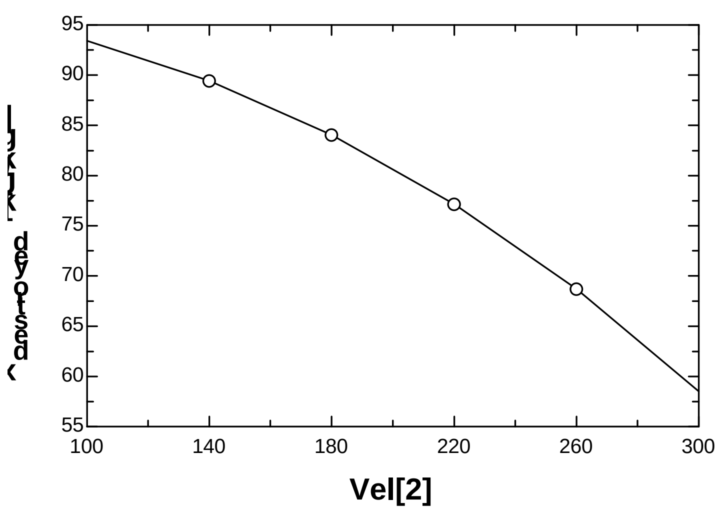
8-60 EES Problem 8-59 is reconsidered. The effect of varying the nozzle exit velocity on the exit temperature and exergy destroyed is to be investigated.

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

```
"Knowns:"
WorkFluid$ = 'Air'
P[1] = 300  [kPa]
T[1] =87 [C]
P[2] = 95  [kPa]
Vel[1] = 50 [m/s]
{Vel[2] = 300 [m/s]}
T_o = 17 [C]
T_surr = T_o
q_loss = 4 [kJ/kg]
"Conservation of Energy - SSSF energy balance for nozzle -- neglecting the change in potential energy:"
h[1]=enthalpy(WorkFluid$,T=T[1])
s[1]=entropy(WorkFluid$,P=P[1],T=T[1])
ke[1] = Vel[1]^2/2
ke[2]=Vel[2]^2/2
h[1]+ke[1]*convert(m^2/s^2,kJ/kg) = h[2] + ke[2]*convert(m^2/s^2,kJ/kg)+q_loss
T[2]=temperature(WorkFluid$,h=h[2])
s[2]=entropy(WorkFluid$,P=P[2],h=h[2])
"The entropy generated is detemined from the entropy balance:"
s[1] - s[2] - q_loss/(T_surr+273) + s_gen = 0
x_destroyed = (T_o+273)*s_gen
```

T_2 [C]	Vel_2 [m/s]	$x_{destroyed}$ [kJ/kg]
79.31	100	93.41
74.55	140	89.43
68.2	180	84.04
60.25	220	77.17
50.72	260	68.7
39.6	300	58.49





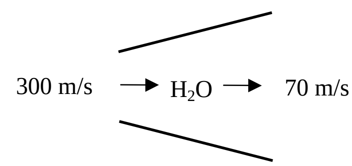
8-61 Steam is decelerated in a diffuser. The mass flow rate of steam and the wasted work potential during the process are to be determined.

Assumptions **1** The diffuser operates steadily. **2** The changes in potential energies are negligible.

Properties The properties of steam at the inlet and the exit of the diffuser are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 10\text{ kPa} \\ T_1 = 50°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 2592.0\text{ kJ/kg} \\ s_1 = 8.1741\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} T_2 = 50°\text{C} \\ \text{sat.vapor} \end{array}\right\} \begin{array}{l} h_2 = 2591.3\text{ kJ/kg} \\ s_2 = 8.0748\text{ kJ/kg}\cdot\text{K} \\ \boldsymbol{v}_2 = 12.026\text{ m}^3\text{/kg} \end{array}$$



Analysis (a) The mass flow rate of the steam is

$$\dot{m} = \frac{1}{\boldsymbol{v}_2} A_2 V_2 = \frac{1}{12.026\text{ m}^3\text{ / kg}} (3\text{ m}^2)(70\text{ m/s}) = \mathbf{17.46\text{ kg/s}}$$

(*b*) We take the diffuser to be the system, which is a control volume. Assuming the direction of heat transfer to be from the stem, the energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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

Substituting,

$$\dot{Q}_{\text{out}} = -(17.46\text{ kg/s})\left[2591.3 - 2592.0 + \frac{(70\text{ m/s})^2 - (300\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2\text{ / s}^2}\right)\right] = 754.8\text{ kJ/s}$$

The wasted work potential is equivalent to exergy destruction. The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{gen}$ where the entropy generation S_{gen} is determined from an entropy balance on an *extended system* that includes the device and its immediate surroundings so that the boundary temperature of the extended system is T_{surr} at all times. It gives

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

$$\dot{m}s_1 - \dot{m}s_2 - \frac{\dot{Q}_{\text{out}}}{T_{\text{b,surr}}} + \dot{S}_{\text{gen}} = 0 \rightarrow \dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1) + \frac{\dot{Q}_{\text{out}}}{T_{\text{surr}}}$$

Substituting, the exergy destruction is determined to be

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = T_0 \left(\dot{m}(s_2 - s_1) + \frac{\dot{Q}_{\text{out}}}{T_0} \right)$$

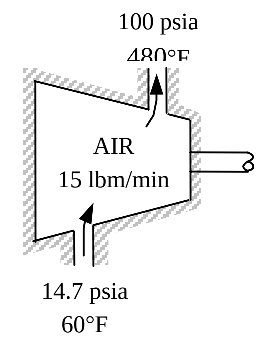
$$= (298\ \text{K}) \left((17.46\ \text{kg/s})(8.0748 - 8.1741)\text{kJ/kg} \cdot \text{K} + \frac{754.8\ \text{kW}}{298\ \text{K}} \right) = \mathbf{238.3\ kW}$$

8-62E Air is compressed steadily by a compressor from a specified state to another specified state. The minimum power input required for the compressor is to be determined.

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

Properties The gas constant of air is R = 0.06855 Btu/lbm.R (Table A-1E). From the air table (Table A-17E)

$$T_1 = 520\text{ R} \longrightarrow h_1 = 124.27\text{ Btu/lbm}$$
$$s_1^{\text{o}} = 0.59173\text{ Btu/lbm}\cdot\text{R}$$
$$T_2 = 940\text{ R} \longrightarrow h_2 = 226.11\text{ Btu/lbm}$$
$$s_2^{\text{o}} = 0.73509\text{ Btu/lbm}\cdot\text{R}$$



Analysis The reversible (or minimum) power input is determined from the rate form of the exergy balance applied on the compressor and setting the exergy destruction term equal to zero,

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

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

where

$$\Delta s_{\text{air}} = s_2^{\text{o}} - s_1^{\text{o}} - R\ln\frac{P_2}{P_1}$$
$$= (0.73509 - 0.59173)\text{Btu/lbm}\cdot\text{R} - (0.06855\text{ Btu/lbm}\cdot\text{R})\ln\frac{100\text{ psia}}{14.7\text{ psia}}$$
$$= 0.01193\text{ Btu/lbm}\cdot\text{R}$$

Substituting,

$$\dot{W}_{\text{rev,in}} = (22/60\text{ lbm/s})\big[(226.11 - 124.27)\text{Btu/lbm} - (520\text{ R})(0.01193\text{ Btu/lbm}\cdot\text{R})\big]$$
$$= 35.1\text{ Btu/s} = \mathbf{49.6\ \ hp}$$

Discussion Note that this is the minimum power input needed for this compressor.

8-63 Steam expands in a turbine from a specified state to another specified state. The actual power output of the turbine is given. The reversible power output and the second-law efficiency are to be determined.

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

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

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

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

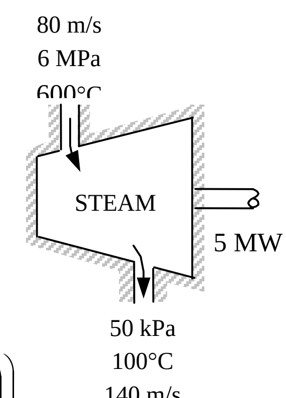
Analysis (*b*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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



Substituting,

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

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

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

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

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

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

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

Substituting,

$$\begin{aligned} \dot{W}_{\text{rev,out}} &= \dot{m}[(h_1 - h_2) - T_0(s_1 - s_2)] \\ &= (5.156\text{ kg/s})[3658.8 - 2682.4 - (298\text{ K})(7.1693 - 7.6953)\text{ kJ/kg}\cdot\text{K}] = \mathbf{5842\ \ kW} \end{aligned}$$

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

$$\eta_{\text{II}} = \frac{\dot{W}_{\text{out}}}{\dot{W}_{\text{rev,out}}} = \frac{5\text{ MW}}{5.842\text{ MW}} = \mathbf{85.6\%}$$

Discussion Note that 14.4% percent of the work potential of the steam is wasted as it flows through the turbine during this process.

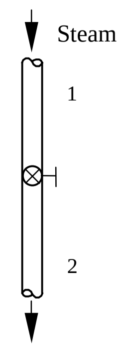
8-64 Steam is throttled from a specified state to a specified pressure. The decrease in the exergy of the steam during this throttling process is to be determined.

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 temperature of the surroundings is given to be 25°C. **4** Heat transfer is negligible.

Properties The properties of steam before and after throttling are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 9\ \text{MPa} \\ T_1 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3387.4\ \text{kJ/kg} \\ s_1 = 6.6603\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 7\ \text{MPa} \\ h_2 = h_1 \end{array}\right\} s_2 = 6.7687\ \text{kJ/kg}\cdot\text{K}$$



Analysis The decrease in exergy is of the steam is the difference between the inlet and exit flow exergies,

$$\begin{aligned} \text{Decrease in exergy} &= \psi_1 - \psi_2 = -[\Delta h^{\nearrow 0} - \Delta ke^{\nearrow 0} - \Delta pe^{\nearrow 0} - T_0(s_1 - s_2)] = T_0(s_2 - s_1) \\ &= (298\ \text{K})(6.7687 - 6.6603)\text{kJ/kg}\cdot\text{K} \\ &= \mathbf{32.3\ kJ/kg} \end{aligned}$$

Discussion Note that 32.3 kJ/kg of work potential is wasted during this throttling process.

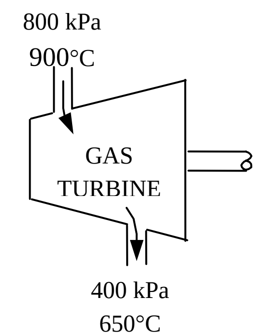
8-65 Combustion gases expand in a turbine from a specified state to another specified state. The exergy of the gases at the inlet and the reversible work output of the turbine are to be determined.

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

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

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

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



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

where

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

Thus,

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

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

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

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

where

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

and

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

Then the reversible work output on a unit mass basis becomes

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

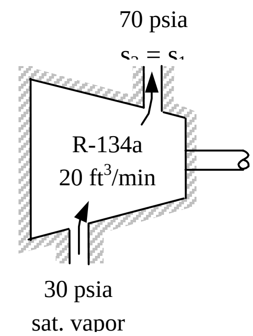
8-66E Refrigerant-134a enters an adiabatic compressor with an isentropic efficiency of 0.80 at a specified state with a specified volume flow rate, and leaves at a specified pressure. The actual power input and the second-law efficiency to the compressor are to be determined.

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.

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

$$\left.\begin{array}{l} P_1 = 30 \text{ psia} \\ \text{sat.vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\,@\,30\text{ psia}} = 105.32 \text{ Btu / lbm} \\ s_1 = s_{g\,@\,30\text{ psia}} = 0.2238 \text{ Btu/lbm} \cdot \text{R} \\ \boldsymbol{v}_1 = \boldsymbol{v}_{g\,@\,30\text{ psia}} = 1.5492 \text{ ft}^3\text{/lbm} \end{array}$$

$$\left.\begin{array}{l} P_2 = 70 \text{ psia} \\ s_{2s} = s_1 \end{array}\right\} h_{2s} = 112.80 \text{ Btu/lbm}$$



Analysis From the isentropic efficiency relation,

$$\eta_c = \frac{h_{2s} - h_1}{h_{2a} - h_1} \longrightarrow h_{2a} = h_1 + (h_{2s} - h_1)/\eta_c$$
$$= 105.32 + (112.80 - 105.32)/0.80$$
$$= 114.67 \text{ Btu/lbm}$$

Then,

$$\left.\begin{array}{l} P_2 = 70 \text{ psia} \\ h_{2a} = 114.67 \end{array}\right\} s_2 = 0.2274 \text{ Btu/lbm}$$

Also, $$\dot{m} = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{20/60 \text{ ft}^3\text{ / s}}{1.5492 \text{ ft}^3\text{ / lbm}} = 0.2152 \text{ lbm/s}$$

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. The energy balance for this steady-flow system can be expressed as

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

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

Substituting, the actual power input to the compressor becomes

$$\dot{W}_{\text{a,in}} = (0.2152 \text{ lbm/s})(114.67 - 105.32) \text{ Btu/lbm} \left(\frac{1 \text{ hp}}{0.7068 \text{ Btu/s}} \right) = \mathbf{2.85 \text{ hp}}$$

(*b*) The reversible (or minimum) power input is determined from the exergy balance applied on the compressor and setting the exergy destruction term equal to zero,

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

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

Substituting,
$$\dot{W}_{\text{rev,in}} = (0.2152 \text{ lbm/s})\left[(114.67 - 105.32)\text{Btu/lbm} - (535 \text{ R})(0.2274 - 0.2238)\text{Btu/lbm} \cdot \text{R}\right]$$
$$= 1.606 \text{ Btu/s} = \mathbf{2.27 \text{ hp}} \qquad (\text{since } 1 \text{ hp} = 0.7068 \text{ Btu/s})$$

Thus, $\eta_{\text{II}} = \dfrac{\dot{W}_{\text{rev,in}}}{\dot{W}_{\text{act,in}}} = \dfrac{2.27\ \text{hp}}{2.85\ \text{hp}} = \mathbf{79.8\%}$

8-67 Refrigerant-134a is compressed by an adiabatic compressor from a specified state to another specified state. The isentropic efficiency and the second-law efficiency of the compressor are to be determined.

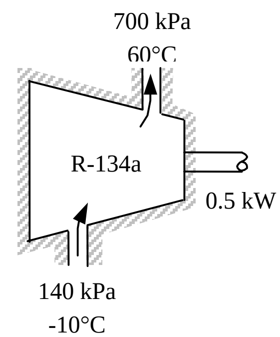
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

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

$$\left.\begin{array}{l} P_1 = 140\text{ kPa} \\ T_1 = -10°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 246.36\text{ kJ/kg} \\ s_1 = 0.97236\text{ kJ/kg}\cdot\text{K} \\ v_1 = 0.14605\text{ m}^3\text{/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 700\text{ kPa} \\ T_2 = 60°\text{C} \end{array}\right\} \begin{array}{l} h_2 = 298.42\text{ kJ/kg} \\ s_2 = 1.0256\text{ kJ/kg}\cdot\text{K} \end{array}$$

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



Analysis (*a*) The isentropic efficiency is

$$\eta_c = \frac{h_{2s}-h_1}{h_{2a}-h_1} = \frac{281.16-246.36}{298.42-246.36} = 0.668 = \mathbf{66.8\%}$$

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

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

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

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

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

Then the mass flow rate of the refrigerant becomes

$$\dot{m} = \frac{\dot{W}_{\text{a,in}}}{h_{2a}-h_1} = \frac{0.5\text{ kJ/s}}{(298.42-246.36)\text{kJ/kg}} = 0.009603\text{ kg/s}$$

The reversible (or minimum) power input is determined from the exergy balance applied on the compressor and setting the exergy destruction term equal to zero,

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

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

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

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

Substituting,

$$\dot{W}_{\text{rev, in}} = (0.009603\text{ kg/s})\left[(298.42 - 246.36)\text{kJ/kg} - (300\text{ K})(1.0256 - 0.97236)\text{kJ/kg}\cdot\text{K}\right] = 0.347\text{ kW}$$

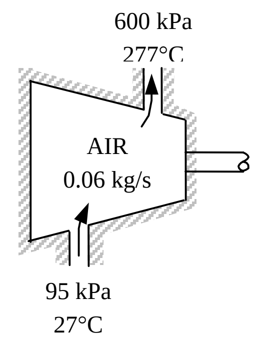
and

$$\eta_{\mathrm{II}} = \frac{\dot{W}_{\mathrm{rev,in}}}{\dot{W}_{\mathrm{a,in}}} = \frac{0.347\ \mathrm{kW}}{0.5\ \mathrm{kW}} = \mathbf{69.3\%}$$

8-68 Air is compressed steadily by a compressor from a specified state to another specified state. The increase in the exergy of air and the rate of exergy destruction are to be determined.

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

Properties The gas constant of air is R = 0.287 kJ/kg.K (Table A-1). From the air table (Table A-17)



$$T_1 = 300\text{ K} \longrightarrow h_1 = 300.19\text{ kJ/kg}$$
$$s_1^{\text{o}} = 1.702\text{ kJ/kg}\cdot\text{K}$$

$$T_2 = 550\text{ K} \longrightarrow h_2 = 555.74\text{ kJ/kg}$$
$$s_2^{\text{o}} = 2.318\text{ kJ/kg}\cdot\text{K}$$

Analysis The reversible (or minimum) power input is determined from the rate form of the exergy balance applied on the compressor and setting the exergy destruction term equal to zero,

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

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

where

$$s_2 - s_1 = s_2^o - s_1^o - R\ln\frac{P_2}{P_1}$$
$$= (2.318 - 1.702)\text{ kJ/kg}\cdot\text{K} - (0.287\text{ kJ/kg}\cdot\text{K})\ln\frac{600\text{ kPa}}{95\text{ kPa}}$$
$$= 0.0870\text{ kJ/kg}\cdot\text{K}$$

Substituting,

$$\dot{W}_{\text{rev,in}} = (0.06\text{ kg/s})\left[(555.74 - 300.19)\text{kJ/kg - }(298\text{ K})(0.0870\text{ kJ/kg}\cdot\text{K})\right] = \mathbf{13.7\text{ kW}}$$

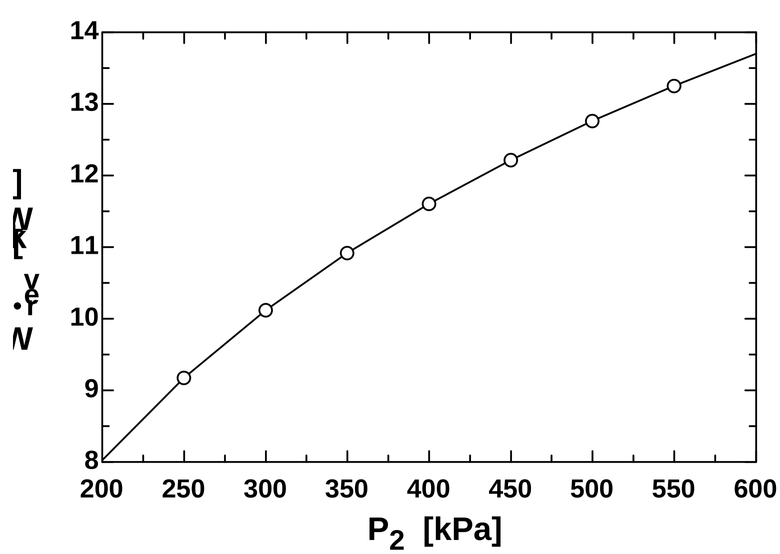
Discussion Note that a minimum of 13.7 kW of power input is needed for this compression process.

8-69 EES Problem 8-68 is reconsidered. The effect of compressor exit pressure on reversible power is to be investigated.

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

```
T_1=27 [C]
P_1=95 [kPa]
m_dot = 0.06 [kg/s]
{P_2=600 [kPa]}
T_2=277 [C]
T_o=25 [C]
P_o=100 [kPa]
m_dot_in=m_dot

"Steady-flow conservation of mass"
m_dot_in = m_dot_out
h_1 =enthalpy(air,T=T_1)
h_2 = enthalpy(air, T=T_2)
W_dot_rev=m_dot_in*(h_2 - h_1 -(T_1+273.15)*(s_2-s_1))
s_1=entropy(air, T=T_1,P=P_1)
s_2=entropy(air,T=T_2,P=P_2)
```

P_2 [kPa]	W_{rev} [kW]
200	8.025
250	9.179
300	10.12
350	10.92
400	11.61
450	12.22
500	12.76
550	13.25
600	13.7



8-70 Argon enters an adiabatic compressor at a specified state, and leaves at another specified state. The reversible power input and irreversibility are to be determined.

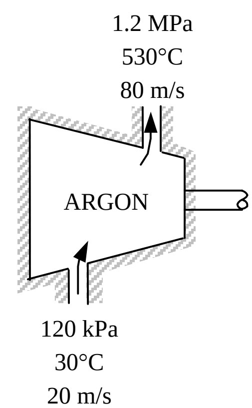
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Argon is an ideal gas with constant specific heats.

Properties For argon, the gas constant is R = 0.2081 kJ/kg.K; the specific heat ratio is k = 1.667; the constant pressure specific heat is c_p = 0.5203 kJ/kg.K (Table A-2).

Analysis The mass flow rate, the entropy change, and the kinetic energy change of argon during this process are

$$\boldsymbol{v}_1 = \frac{RT_1}{P_1} = \frac{(0.2081\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(303\,\text{K})}{(120\,\text{kPa})} = 0.5255\,\text{m}^3/kg$$

$$\dot{m} = \frac{1}{\boldsymbol{v}_1}\mathbf{A}_1 V_1 = \frac{1}{0.5255\,\text{m}^3/\text{kg}}(0.0130\,\text{m}^2)(20\,\text{m/s}) = 0.495\,\text{kg/s}$$



$$\begin{aligned} s_2 - s_1 &= c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} \\ &= (0.5203\,\text{kJ/kg}\cdot\text{K})\ln\frac{803\,\text{K}}{303\,\text{K}} - (0.2081\,\text{kJ/kg}\cdot\text{K})\ln\frac{1200\,\text{kPa}}{120\,\text{kPa}} \\ &= 0.02793\,\text{kJ/kg}\cdot\text{K} \end{aligned}$$

and

$$\Delta ke = \frac{V_2^2 - V_1^2}{2} = \frac{(80\,\text{m/s})^2 - (20\,\text{m/s})^2}{2}\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) = 3.0\,\text{kJ/kg}$$

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

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

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

Substituting,

$$\begin{aligned} \dot{W}_{\text{rev,in}} &= \dot{m}\left[c_p(T_2 - T_1) - T_0(s_2 - s_1) + \Delta ke\right] \\ &= (0.495\,\text{kg/s})\left[(0.5203\,\text{kJ/kg}\cdot\text{K})(530-30)\text{K} - (298\,\text{K})(0.02793\,\text{kJ/kg}\cdot\text{K}) + 3.0\right] = \mathbf{126\ kW} \end{aligned}$$

The exergy destruction (or irreversibility) can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ where the entropy generation is determined from an entropy balance on the system, which is an adiabatic steady-flow device,

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

$$\dot{m}s_1 - \dot{m}s_2 + \dot{S}_{\text{gen}} = 0 \;\rightarrow\; \dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1)$$

Substituting,

$$\dot{X}_{\text{destroyed}} = T_0\dot{m}(s_2 - s_1) = (298\ \text{K})(0.495\ \text{kg/s})(0.02793\ \text{kJ/kg}\cdot\text{K}) = \mathbf{4.12\ \ kW}$$

8-71 Steam expands in a turbine steadily at a specified rate from a specified state to another specified state. The power potential of the steam at the inlet conditions and the reversible power output are to be determined.

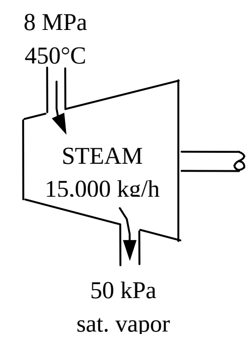
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The temperature of the surroundings is given to be 25°C.

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

$$\left.\begin{array}{l} P_1 = 8\text{ MPa} \\ T_1 = 450°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3273.3\text{ kJ/kg} \\ s_1 = 6.5579\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 50\text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_2 = 2645.2\text{ kJ/kg} \\ s_2 = 7.5931\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_0 = 100\text{ kPa} \\ T_0 = 25°\text{C} \end{array}\right\} \begin{array}{l} h_0 \cong h_{f@25°\text{C}} = 104.83\text{ kJ/kg} \\ s_0 \cong s_{f@25°\text{C}} = 0.36723\text{ kJ/kg}\cdot\text{K} \end{array}$$



Analysis (*a*) The power potential of the steam at the inlet conditions is equivalent to its exergy at the inlet state,

$$\dot{\Psi} = \dot{m}\psi_1 = \dot{m}\left(h_1 - h_0 - T_0(s_1 - s_0) + \frac{V_1^{2}\nearrow^0}{2} + gz_1\nearrow^0 \right) = \dot{m}\big(h_1 - h_0 - T_0(s_1 - s_0)\big)$$

$$= (15{,}000 / 3600\text{ kg/s})\big[(3273.3 - 104.83)\text{kJ/kg} - (298\text{ K})(6.5579 - 0.36723)\text{kJ/kg}\cdot\text{K}\big]$$

$$= \mathbf{5515\ kW}$$

(*b*) The power output of the turbine if there were no irreversibilities is the reversible power, is determined from the rate form of the exergy balance applied on the turbine and setting the exergy destruction term equal to zero,

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

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

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

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

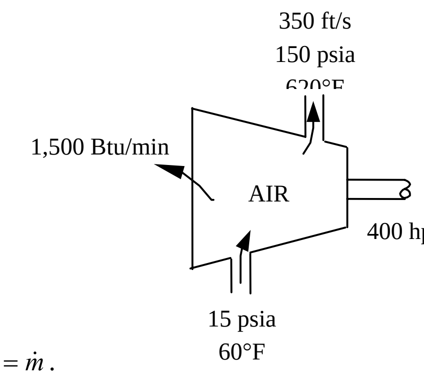
Substituting,

$$\dot{W}_{\text{rev,out}} = \dot{m}[(h_1 - h_2) - T_0(s_1 - s_2)]$$

$$= (15{,}000/3600\text{ kg/s})\big[(3273.3 - 2645.2)\text{ kJ/kg} - (298\text{ K})(6.5579 - 7.5931)\text{ kJ/kg}\cdot\text{K}\big]$$

$$= \mathbf{3902\ kW}$$

8-72E Air is compressed steadily by a 400-hp compressor from a specified state to another specified state while being cooled by the ambient air. The mass flow rate of air and the part of input power that is used to just overcome the irreversibilities are to be determined.



Assumptions **1** Air is an ideal gas with variable specific heats. **2** Potential energy changes are negligible. **3** The temperature of the surroundings is given to be 60°F.

Properties The gas constant of air is R = 0.06855 Btu/lbm.R (Table A-1E). From the air table (Table A-17E)

$$\left.\begin{array}{l} T_1 = 520\text{ R} \\ P_1 = 15\text{ psia} \end{array}\right\} \begin{array}{l} h_1 = 124.27\text{ Btu/lbm} \\ s_1^o = 0.59173\text{ Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} T_2 = 1080\text{ R} \\ P_2 = 150\text{ psia} \end{array}\right\} \begin{array}{l} h_2 = 260.97\text{ Btu/lbm} \\ s_1^o = 0.76964\text{ Btu/lbm}\cdot\text{R} \end{array}$$

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

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

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

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

Substituting, the mass flow rate of the refrigerant becomes

$$(400\text{ hp})\left(\frac{0.7068\text{ Btu/s}}{1\text{ hp}}\right) - (1500/60\text{ Btu/s}) = \dot{m}\left(260.97 - 124.27 + \frac{(350\text{ ft/s})^2 - 0}{2}\frac{1\text{ Btu/lbm}}{25{,}037\text{ ft}^2/\text{s}^2}\right)$$

It yields $\dot{m} = \mathbf{1.852\ lbm/s}$

(b) The portion of the power output that is used just to overcome the irreversibilities is equivalent to exergy destruction, which can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ where the entropy generation S_{gen} is determined from an entropy balance on an *extended system* that includes the device and its immediate surroundings. It gives

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

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

where

$$s_2 - s_1 = s_2^0 - s_1^0 - R\ln\frac{P_2}{P_1} = (0.76964 - 0.59173)\text{ Btu/lbm} - (0.06855\text{ Btu/lbm.R})\ln\frac{150\text{ psia}}{15\text{ psia}}$$
$$= 0.02007\ \text{Btu/lbm.R}$$

Substituting, the exergy destruction is determined to be

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = T_0 \left(\dot{m}(s_2 - s_1) + \frac{\dot{Q}_{\text{out}}}{T_0} \right)$$

$$= (520\text{ R})\left((1.852\text{ lbm/s})(0.02007\text{ Btu/lbm}\cdot\text{R}) + \frac{1500/60\text{ Btu/s}}{520\text{ R}} \right)\left(\frac{1\text{ hp}}{0.7068\text{ Btu/s}} \right) = \mathbf{62.72\ hp}$$

8-73 Hot combustion gases are accelerated in an adiabatic nozzle. The exit velocity and the decrease in the exergy of the gases are to be determined.

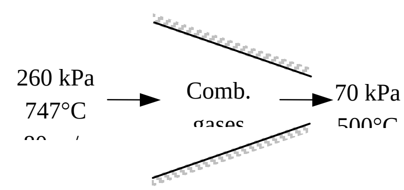
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** The combustion gases are ideal gases with constant specific heats.

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

$$R = c_p - c_v = c_p - c_p / k = c_p(1-1/k) = (1.15\,\text{kJ/kg}\cdot\text{K})(1-1/1.3) = 0.2654\,\text{kJ/kg}\cdot\text{K}$$

Analysis (a) 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. The energy balance for this steady-flow system can be expressed as

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



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

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

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

Then the exit velocity becomes

$$\begin{aligned} V_2 &= \sqrt{2c_p(T_1 - T_2) + V_1^2} \\ &= \sqrt{2(1.15\,\text{kJ/kg}\cdot\text{K})(747-500)\text{K}\left(\frac{1000\,\text{m}^2/\text{s}^2}{1\,\text{kJ/kg}}\right) + (80\,\text{m/s})^2} \\ &= \mathbf{758\,m/s} \end{aligned}$$

(b) The decrease in exergy of combustion gases is simply the difference between the initial and final values of flow exergy, and is determined to be

$$\psi_1 - \psi_2 = w_{\text{rev}} = h_1 - h_2 - \Delta ke - \Delta pe^{\nearrow 0} + T_0(s_2 - s_1) = c_p(T_1 - T_2) + T_0(s_2 - s_1) - \Delta\text{ke}$$

where

$$\Delta\text{ke} = \frac{V_2^2 - V_1^2}{2} = \frac{(758\,\text{m/s})^2 - (80\,\text{m/s})^2}{2}\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) = 284.1\,\text{kJ/kg}$$

and

$$\begin{aligned} s_2 - s_1 &= c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} \\ &= (1.15\,\text{kJ/kg}\cdot\text{K})\ln\frac{773\,\text{K}}{1020\,\text{K}} - (0.2654\,\text{kJ/kg}\cdot\text{K})\ln\frac{70\,\text{kPa}}{260\,\text{kPa}} \\ &= 0.02938\,\text{kJ/kg}\cdot\text{K} \end{aligned}$$

Substituting,

$$\begin{aligned} \text{Decrease in exergy} &= \psi_1 - \psi_2 \\ &= (1.15\,\text{kJ/kg}\cdot\text{K})(747-500)°\text{C} + (293\,\text{K})(0.02938\,\text{kJ/kg}\cdot\text{K}) - 284.1\,\text{kJ/kg} \\ &= \mathbf{8.56\ kJ/kg} \end{aligned}$$

8-74 Steam is accelerated in an adiabatic nozzle. The exit velocity of the steam, the isentropic efficiency, and the exergy destroyed within the nozzle are to be determined.

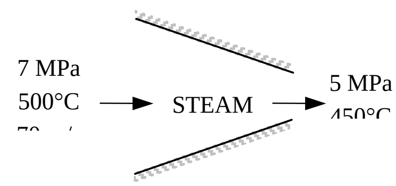
Assumptions **1** The nozzle operates steadily. **2** The changes in potential energies are negligible.

Properties The properties of steam at the inlet and the exit of the nozzle are (Tables A-4 through A-6)

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

$$\left.\begin{array}{l} P_2 = 5\text{ MPa} \\ T_2 = 450°\text{C} \end{array}\right\} \begin{array}{l} h_2 = 3317.2\text{ kJ/kg} \\ s_2 = 6.8210\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_{2s} = 5\text{ MPa} \\ s_{2s} = s_1 \end{array}\right\} h_{2s} = 3302.0\text{ kJ/kg}$$



Analysis (*a*) We take the nozzle to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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

Then the exit velocity becomes

$$V_2 = \sqrt{2(h_1 - h_2) + \mathbf{V}_1^2} = \sqrt{2(3411.4 - 3317.2)\text{ kJ/kg}\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right) + (70\text{ m/s})^2} = \mathbf{439.6\text{ m/s}}$$

(*b*) The exit velocity for the isentropic case is determined from

$$V_{2s} = \sqrt{2(h_1 - h_{2s}) + \mathbf{V}_1^2} = \sqrt{2(3411.4 - 3302.0)\text{ kJ/kg}\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right) + (70\text{ m/s})^2} = 472.9\text{ m/s}$$

Thus,

$$\eta_N = \frac{V_2^2/2}{V_{2s}^2/2} = \frac{(439.6\text{ m/s})^2/2}{(472.9\text{ m/s})^2/2} = \mathbf{86.4\%}$$

(*c*) The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{gen}$ where the entropy generation S_{gen} is determined from an entropy balance on the actual nozzle. It gives

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

$$\dot{m}s_1 - \dot{m}s_2 + \dot{S}_{\text{gen}} = 0 \rightarrow \dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1) \text{ or } \quad s_{\text{gen}} = s_2 - s_1$$

Substituting, the exergy destruction in the nozzle on a unit mass basis is determined to be

$$x_{\text{destroyed}} = T_0 s_{\text{gen}} = T_0(s_2 - s_1) = (298\text{ K})(6.8210 - 6.8000)\text{kJ/kg}\cdot\text{K} = \mathbf{6.28\ \text{kJ/kg}}$$

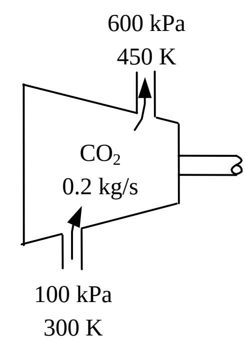
8-75 CO_2 gas is compressed steadily by a compressor from a specified state to another specified state. The power input to the compressor if the process involved no irreversibilities is to be determined.

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 = 1.261 and c_p = 0.917 kJ/kg.K (Table A-2).

Analysis The reversible (or minimum) power input is determined from the exergy balance applied on the compressor, and setting the exergy destruction term equal to zero,

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



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

where

$$\begin{aligned}
s_2 - s_1 &= c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} \\
&= (0.9175\ \text{kJ/kg}\cdot\text{K})\ln\frac{450\ \text{K}}{300\ \text{K}} - (0.1889\ \text{kJ/kg}\cdot\text{K})\ln\frac{600\ \text{kPa}}{100\ \text{kPa}} \\
&= 0.03335\ \text{kJ/kg}\cdot\text{K}
\end{aligned}$$

Substituting,

$$\dot{W}_{\text{rev,in}} = (0.2\ \text{kg/s})\left[(0.917\ \text{kJ/kg}\cdot\text{K})(450-300)\text{K} - (298\ \text{K})(0.03335\ \text{kJ/kg}\cdot\text{K})\right] = \mathbf{25.5\ kW}$$

Discussion Note that a minimum of 25.5 kW of power input is needed for this compressor.

8-76E A hot water stream is mixed with a cold water stream. For a specified mixture temperature, the mass flow rate of cold water stream and the rate of exergy destruction are to be determined.

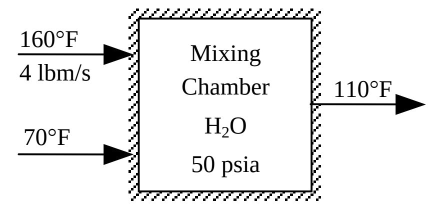
Assumptions **1** Steady operating conditions exist. **2** The mixing chamber is well-insulated so that heat loss to the surroundings is negligible. **3** Changes in the kinetic and potential energies of fluid streams are negligible.

Properties Noting that that T < $T_{sat@50\ psia}$ = 280.99°F, the water in all three streams exists as a compressed liquid, which can be approximated as a saturated liquid at the given temperature. Thus from Table A-4E,

$$\left.\begin{array}{l} P_1 = 50\text{ psia} \\ T_1 = 160°\text{F} \end{array}\right\} \begin{array}{l} h_1 \cong h_{f\,@160°\text{F}} = 128.00\text{ Btu/lbm} \\ s_1 \cong s_{f\,@160°\text{F}} = 0.23136\text{ Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_2 = 50\text{ psia} \\ T_2 = 70°\text{F} \end{array}\right\} \begin{array}{l} h_2 \cong h_{f\,@70°\text{F}} = 38.08\text{ Btu/lbm} \\ s_2 \cong s_{f\,@70°\text{F}} = 0.07459\text{ Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_3 = 50\text{ psia} \\ T_3 = 110°\text{F} \end{array}\right\} \begin{array}{l} h_3 \cong h_{f\,@110°\text{F}} = 78.02\text{ Btu/lbm} \\ s_3 \cong s_{f\,@110°\text{F}} = 0.14728\text{ Btu/lbm}\cdot\text{R} \end{array}$$



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

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

Energy balance:

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

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

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

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

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

$$\dot{m}_2 = \frac{h_1 - h_3}{h_3 - h_2}\dot{m}_1 = \frac{(128.00 - 78.02)\text{Btu/lbm}}{(78.02 - 38.08)\text{Btu/lbm}}(4.0\text{ lbm/s}) = \mathbf{5.0\ lbm/s}$$

Also,

$$\dot{m}_3 = \dot{m}_1 + \dot{m}_2 = 4 + 5 = 9\text{ lbm/s}$$

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

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

$$\dot{m}_1 s_1 + \dot{m}_2 s_2 - \dot{m}_3 s_3 + \dot{S}_{gen} = 0 \;\rightarrow\; \dot{S}_{gen} = \dot{m}_3 s_3 - \dot{m}_1 s_1 - \dot{m}_2 s_2$$

Substituting, the exergy destruction is determined to be

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = T_0 (\dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_1 s_1)$$

$$= (535\ \text{R})(9.0 \times 0.14728 - 5.0 \times 0.07459 - 4.0 \times 0.23136)\text{Btu/s} \cdot \text{R}$$

$$= \mathbf{14.7\ Btu/s}$$

8-77 Liquid water is heated in a chamber by mixing it with superheated steam. For a specified mixing temperature, the mass flow rate of the steam and the rate of exergy destruction are to be determined.

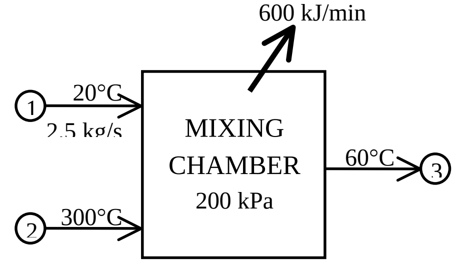
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions.

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

$$\left.\begin{array}{l} P_1 = 200\text{ kPa} \\ T_1 = 20°\text{C} \end{array}\right\} \begin{array}{l} h_1 \cong h_{f@20^\circ\text{C}} = 83.91\text{ kJ/kg} \\ s_1 \cong s_{f@20^\circ\text{C}} = 0.29649\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 200\text{ kPa} \\ T_2 = 300°\text{C} \end{array}\right\} \begin{array}{l} h_2 = 3072.1\text{ kJ/kg} \\ s_2 = 7.8941\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_3 = 200\text{ kPa} \\ T_3 = 60°\text{C} \end{array}\right\} \begin{array}{l} h_3 \cong h_{f@60^\circ\text{C}} = 251.18\text{ kJ/kg} \\ s_3 \cong s_{f@60^\circ\text{C}} = 0.83130\text{ kJ/kg}\cdot\text{K} \end{array}$$



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

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

Energy balance:

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

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

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

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

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

$$\dot{m}_2 = \frac{\dot{Q}_{\text{out}} - \dot{m}_1(h_1 - h_3)}{h_2 - h_3} = \frac{(600/60\text{ kJ/s}) - (2.5\text{ kg/s})(83.91 - 251.18)\text{kJ/kg}}{(3072.1 - 251.18)\text{kJ/kg}} = \mathbf{0.148\ kg/s}$$

Also, $\dot{m}_3 = \dot{m}_1 + \dot{m}_2 = 2.5 + 0.148 = 2.648\text{ kg/s}$

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

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

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

Substituting, the exergy destruction is determined to be

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = T_0 \left(\dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_1 s_1 + \frac{\dot{Q}_{out}}{T_{b,surr}} \right)$$

$$= (298\text{ K})(2.648 \times 0.83130 - 0.148 \times 7.8941 - 2.5 \times 0.29649 + 10 / 298)\text{kW/K} = \mathbf{96.4\ kW}$$

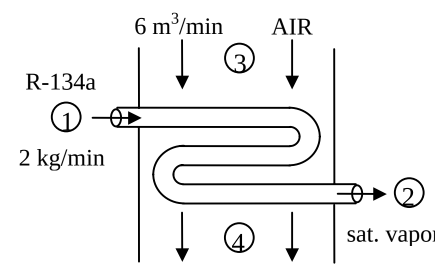
8-78 Refrigerant-134a is vaporized by air in the evaporator of an air-conditioner. For specified flow rates, the exit temperature of air and the rate of exergy destruction are to be determined for the cases of insulated and uninsulated evaporator.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** Air is an ideal gas with constant specific heats at room temperature.

Properties The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1). The constant pressure specific heat of air at room temperature is c_p = 1.005 kJ/kg.K (Table A-2). The properties of R-134a at the inlet and the exit states are (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_1 = 120\text{ kPa} \\ x_1 = 0.3 \end{array}\right\} \begin{array}{l} h_1 = h_f + x_1 h_{fg} = 22.49 + 0.3 \times 214.48 = 86.83\text{ kJ/kg} \\ s_1 = s_f + x_1 s_{fg} = 0.09275 + 0.3(0.85503) = 0.34926\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} T_2 = 120\text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_2 = h_{g\,@120\text{ kPa}} = 236.97\text{ kJ/kg} \\ s_2 = s_{g\,@120\text{ kPa}} = 0.94779\text{ kJ/kg}\cdot\text{K} \end{array}$$



Analysis Air at specified conditions can be treated as an ideal gas with specific heats at room temperature. The properties of the refrigerant are

$$\dot{m}_{\text{air}} = \frac{P_3 \dot{\boldsymbol{V}}_3}{RT_3} = \frac{(100\text{ kPa})(6\text{ m}^3/\text{min})}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{ K})} = 6.97\text{ kg/min}$$

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

Mass balance (*for each fluid stream*):

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

Energy balance (for the entire heat exchanger):

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

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

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

Combining the two, $\dot{m}_R (h_2 - h_1) = \dot{m}_{\text{air}} (h_3 - h_4) = \dot{m}_{\text{air}} c_p (T_3 - T_4)$

Solving for T_4, $T_4 = T_3 - \dfrac{\dot{m}_R (h_2 - h_1)}{\dot{m}_{\text{air}} c_p}$

Substituting, $T_4 = 27°\text{C} - \dfrac{(2\text{ kg/min})(236.97 - 86.83)\text{ kJ/kg}}{(6.97\text{ kg/min})(1.005\text{ kJ/kg}\cdot\text{K})} = \mathbf{-15.9°C} = 257.1\text{ K}$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ where the entropy generation S_{gen} is determined from an entropy balance on the evaporator. Noting that the condenser is well-insulated and thus heat transfer is negligible, the entropy balance for this steady-flow system can be expressed as

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

$$\dot m_1 s_1+\dot m_3 s_3-\dot m_2 s_2-\dot m_4 s_4+\dot S_{\rm gen}=0 \quad (\text{since } Q=0)$$

$$\dot m_R s_1+\dot m_{\rm air} s_3-\dot m_R s_2-\dot m_{\rm air} s_4+\dot S_{\rm gen}=0$$

or, $$\dot S_{\rm gen}=\dot m_R(s_2-s_1)+\dot m_{\rm air}(s_4-s_3)$$

where $$s_4-s_3=c_p\ln\frac{T_4}{T_3}-R\ln\frac{P_4^{\nearrow 0}}{P_3}=c_p\ln\frac{T_4}{T_3}=(1.005\ \text{kJ/kg}\cdot\text{K})\ln\frac{257.1\ \text{K}}{300\ \text{K}}=-0.1551\ \text{kJ/kg}\cdot\text{K}$$

Substituting, the exergy destruction is determined to be

$$\begin{aligned}\dot X_{\rm destroyed}&=T_0\dot S_{\rm gen}=T_0[\dot m_R(s_2-s_1)+\dot m_{\rm air}(s_4-s_3)]\\&=(305\ \text{K})\left[(2\ \text{kg/min})(0.94779-0.34926)\text{kJ/kg}\cdot\text{K}+(6.97\ \text{kg/min})(-0.1551\ \text{kJ/kg}\cdot\text{K})\right]\\&=35.4\ \text{kJ/min}=\mathbf{0.59\ kW}\end{aligned}$$

(*b*) When there is a heat gain from the surroundings, the steady-flow energy equation reduces to

$$\dot Q_{\rm in}=\dot m_R(h_2-h_1)+\dot m_{\rm air}c_p(T_4-T_3)$$

Solving for T_4, $$T_4=T_3+\frac{\dot Q_{\rm in}-\dot m_R(h_2-h_1)}{\dot m_{\rm air}c_p}$$

Substituting, $$T_4=27^\circ\text{C}+\frac{(30\ \text{kJ/min})-(2\ \text{kg/min})(236.97-86.83)\ \text{kJ/kg}}{(6.97\ \text{kg/min})(1.005\ \text{kJ/kg}\cdot\text{K})}=\mathbf{-11.6^\circ C}=261.4\ \text{K}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\rm destroyed}=T_0S_{gen}$ where the entropy generation $S_{\rm gen}$ is determined from an entropy balance on an *extended system* that includes the evaporator and its immediate surroundings. It gives

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

$$\frac{Q_{\rm in}}{T_{\rm b,in}}+\dot m_1 s_1+\dot m_3 s_3-\dot m_2 s_2-\dot m_4 s_4+\dot S_{\rm gen}=0$$

$$\frac{Q_{\rm in}}{T_0}+\dot m_R s_1+\dot m_{\rm air} s_3-\dot m_R s_2-\dot m_{\rm air} s_4+\dot S_{\rm gen}=0$$

or $$\dot S_{\rm gen}=\dot m_R(s_2-s_1)+\dot m_{\rm air}(s_4-s_3)-\frac{\dot Q_{\rm in}}{T_0}$$

where $$s_4-s_3=c_p\ln\frac{T_4}{T_3}-R\ln\frac{P_4^{\nearrow 0}}{P_3}=(1.005\ \text{kJ/kg}\cdot\text{K})\ln\frac{261.4\ \text{K}}{300\ \text{K}}=-0.1384\ \text{kJ/kg}\cdot\text{K}$$

Substituting, the exergy destruction is determined to be

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = T_0 \left[\dot{m}_R (s_2 - s_1) + \dot{m}_{air} (s_4 - s_3) - \frac{\dot{Q}_{\text{in}}}{T_0} \right]$$

$$= (305\ \text{K}) \left[(2\ \text{kg/min})(0.94779 - 0.34926)\text{kJ/kg} \cdot \text{K} + (6.97\text{kg/min})(-0.1384\ \text{kJ/kg} \cdot \text{K}) - \frac{30\ \text{kJ/min}}{305\ \text{K}} \right]$$

$$= 40.9\ \text{kJ/min} = \mathbf{0.68\ \ kW}$$