Chapter 5
THE FIRST LAW OF THERMODYNAMICS

Closed System Energy Balance: General Systems

5-1C No. This is the case for adiabatic systems only.

5-2C Warmer. Because energy is added to the room air in the form of electrical work.

5-3C Warmer. If we take the room that contains the refrigerator as our system, we will see that electrical work is supplied to this room to run the refrigerator, which is eventually dissipated to the room as waste heat.

5-4C Energy can be transferred to or from a control volume as heat, various forms of work, and by mass transport.

5-5 Water is heated in a pan on top of a range while being stirred. The energy of the water at the end of the process is to be determined.

Assumptions The pan is stationary and thus the changes in kinetic and potential energies are negligible.

Analysis We take the water in the pan as our system. This is a closed system since no mass enters or leaves. Applying the energy balance on this system gives

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

$$Q_{in} + W_{pw,in} - Q_{out} = \Delta U = U_2 - U_1$$

$$30\text{ kJ} + 0.5\text{ kJ} - 5\text{ kJ} = U_2 - 10\text{ kJ}$$

$$U_2 = \mathbf{35.5\ kJ}$$

Therefore, the final internal energy of the system is 35.5 kJ.

5-6E Water is heated in a cylinder on top of a range. The change in the energy of the water during this process is to be determined.

Assumptions The pan is stationary and thus the changes in kinetic and potential energies are negligible.

Analysis We take the water in the cylinder as the system. This is a closed system since no mass enters or leaves. Applying the energy balance on this system gives

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

$$Q_{in} - W_{b,out} - Q_{out} = \Delta U = U_2 - U_1$$

$$65\text{ Btu} - 5\text{ Btu} - 8\text{ Btu} = \Delta U$$

$$\Delta U = U_2 - U_1 = \mathbf{52\ Btu}$$

Therefore, the energy content of the system increases by 52 Btu during this process.

5-7 A classroom is to be air-conditioned using window air-conditioning units. The cooling load is due to people, lights, and heat transfer through the walls and the windows. The number of 5-kW window air conditioning units required is to be determined.

Assumptions There are no heat dissipating equipment (such as computers, TVs, or ranges) in the room.

Analysis The total cooling load of the room is determined from

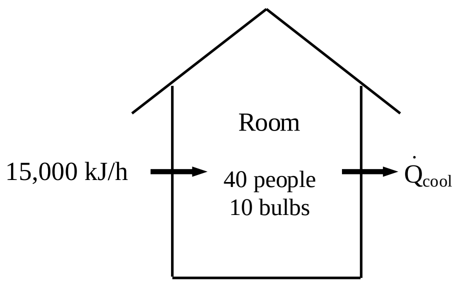
$$\dot{Q}_{cooling} = \dot{Q}_{lights} + \dot{Q}_{people} + \dot{Q}_{heat\ gain}$$

where

$$\dot{Q}_{lights} = 10 \times 100\text{ W} = 1\text{ kW}$$

$$\dot{Q}_{people} = 40 \times 360\text{ kJ/h} = 4\text{ kW}$$

$$\dot{Q}_{heat\ gain} = 15{,}000\text{ kJ/h} = 4.17\text{ kW}$$



Substituting,

$$\dot{Q}_{cooling} = 1 + 4 + 4.17 = 9.17\text{ kW}$$

Thus the number of air-conditioning units required is

$$\frac{9.17\text{ kW}}{5\text{ kw/unit}} = 1.83 \longrightarrow \mathbf{2\ units}$$

5-8 An industrial facility is to replace its 40-W standard fluorescent lamps by their 35-W high efficiency counterparts. The amount of energy and money that will be saved a year as well as the simple payback period are to be determined.

Analysis The reduction in the total electric power consumed by the lighting as a result of switching to the high efficiency fluorescent is

$$\begin{aligned}\text{Wattage reduction} &= \text{(Wattage reduction per lamp)(Number of lamps)}\\ &= \text{(40 - 34 W/lamp)(700 lamps)}\\ &= \text{4200 W}\end{aligned}$$

Then using the relations given earlier, the energy and cost savings associated with the replacement of the high efficiency fluorescent lamps are determined to be

$$\begin{aligned}\text{Energy Savings} &= \text{(Total wattage reduction)(Ballast factor)(Operating hours)}\\ &= \text{(4.2 kW)(1.1)(2800 h/year)}\\ &= \mathbf{12{,}936\ kWh/year}\end{aligned}$$

$$\begin{aligned}\text{Cost Savings} &= \text{(Energy savings)(Unit electricity cost)}\\ &= \text{(12,936 kWh/year)(\$0.08/kWh)}\\ &= \mathbf{\$1035}\end{aligned}$$

The implementation cost of this measure is simply the extra cost of the energy efficient fluorescent bulbs relative to standard ones, and is determined to be

$$\begin{aligned}\text{Implementation Cost} &= \text{(Cost difference of lamps)(Number of lamps)}\\ &= \text{[(\$2.26-\$1.77)/lamp](700 lamps)}\\ &= \text{\$343}\end{aligned}$$

This gives a simple payback period of

$$\text{Simple payback period} = \frac{\text{Implementation cost}}{\text{Annual cost savings}} = \frac{\$343}{\$1035\,/\,\text{year}} = \mathbf{0.33\ year}\ \ \text{(4.0 months)}$$

Discussion Note that if all the lamps were burned out today and are replaced by high-efficiency lamps instead of the conventional ones, the savings from electricity cost would pay for the cost differential in about 4 months. The electricity saved will also help the environment by reducing the amount of CO_2, CO, NO_x, etc. associated with the generation of electricity in a power plant.

5-9 The lighting energy consumption of a storage room is to be reduced by installing motion sensors. The amount of energy and money that will be saved as well as the simple payback period are to be determined.

Assumptions The electrical energy consumed by the ballasts is negligible.

Analysis The plant operates 12 hours a day, and thus currently the lights are on for the entire 12 hour period. The motion sensors installed will keep the lights on for 3 hours, and off for the remaining 9 hours every day. This corresponds to a total of 9×365 = 3285 off hours per year. Disregarding the ballast factor, the annual energy and cost savings become

Energy Savings = (Number of lamps)(Lamp wattage)(Reduction of annual operating hours)
= (24 lamps)(60 W/lamp)(3285 hours/year)
= **4730 kWh/year**

Cost Savings = (Energy Savings)(Unit cost of energy)
= (4730 kWh/year)($0.08/kWh)
= **$378/year**

The implementation cost of this measure is the sum of the purchase price of the sensor plus the labor,

Implementation Cost = Material + Labor = $32 + $40 = $72

This gives a simple payback period of

$$\text{Simple payback period} = \frac{\text{Implementation cost}}{\text{Annual cost savings}} = \frac{\$72}{\$378/\text{ year}} = \mathbf{0.19\ year}\ \ (2.3\,\text{months})$$

Therefore, the motion sensor will pay for itself in about 2 months.

5-10 The classrooms and faculty offices of a university campus are not occupied an average of 4 hours a day, but the lights are kept on. The amounts of electricity and money the campus will save per year if the lights are turned off during unoccupied periods are to be determined.

Analysis The total electric power consumed by the lights in the classrooms and faculty offices is

$$\dot{E}_{\text{lighting, classroom}} = (\text{Power consumed per lamp})\times(\text{No. of lamps}) = (200\times12\times110\text{ W}) = 264{,}000 = 264\text{ kW}$$

$$\dot{E}_{\text{lighting, offices}} = (\text{Power consumed per lamp})\times(\text{No. of lamps}) = (400\times6\times110\text{ W}) = 264{,}000 = 264\text{ kW}$$

$$\dot{E}_{\text{lighting, total}} = \dot{E}_{\text{lighting, classroom}} + \dot{E}_{\text{lighting, offices}} = 264 + 264 = 528\text{ kW}$$

Noting that the campus is open 240 days a year, the total number of unoccupied work hours per year is

Unoccupied hours = (4 hours/day)(240 days/year) = 960 h/yr

Then the amount of electrical energy consumed per year during unoccupied work period and its cost are

$$\text{Energy savings} = (\dot{E}_{\text{lighting, classroom}})(\text{Unoccupied hours}) = (528\text{ kW})(960\text{ h/yr}) = 506{,}880\text{ kWh}$$

$$\text{Cost savings} = (\text{Energy savings})(\text{Unit cost of energy}) = (506{,}880\text{ kWh})(\$0.082/\text{kWh}) = \mathbf{\$41{,}564/yr}$$

Discussion Note that simple conservation measures can result in significant energy and cost savings.

5-11 The radiator of a steam heating system is initially filled with superheated steam. The valves are closed, and steam is allowed to cool until the pressure drops to a specified value by transferring heat to the room. The amount of heat transfer is to be determined, and the process is to be shown on a *P-v* diagram.

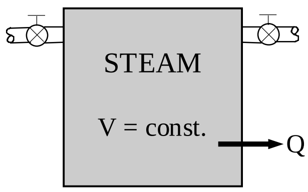
Assumptions **1** The tank is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions.

Analysis We take the radiator as the system. This is a closed system since no mass enters or leaves. Noting that the volume of the system is constant and thus there is no boundary work, the energy balance for this stationary closed system can be expressed as

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

$$-Q_{out} = \Delta U = m(u_2 - u_1) \quad (\text{since } W = \text{KE} = \text{PE} = 0)$$

$$Q_{out} = m(u_1 - u_2)$$



Using data from the steam tables (Tables A-4 through A-6), some properties are determined to be

$$\left.\begin{array}{l} P_1 = 300 \text{kPa} \\ T_1 = 250^{\circ}\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.7964 \text{m}^3/\text{kg} \\ u_1 = 2728.7 \text{kJ/kg} \end{array}$$

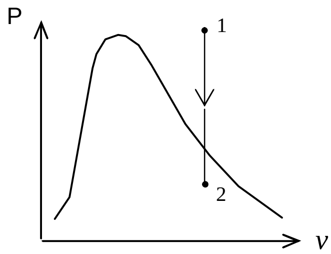
$$P_2 = 100 \text{kPa} \rightarrow \begin{array}{ll} v_f = 0.001043, & v_g = 1.6940 \text{m}^3/\text{kg} \\ u_f = 417.36, & u_{fg} = 2088.7 \text{kJ/kg} \end{array}$$

Noting that $v_1 = v_2$ and $v_f < v_2 < v_g$, the mass and the final internal energy becomes

$$m = \frac{V_1}{v_1} = \frac{0.020 \text{ m}^3}{0.7964 \text{ m}^3/\text{kg}} = 0.0251 \text{ kg}$$

$$x_2 = \frac{v_2 - v_f}{v_{fg}} = \frac{0.7964 - 0.001043}{1.6940 - 0.001043} = 0.470$$

$$u_2 = u_f + x_2 u_{fg} = 417.36 + (0.470 \times 2088.7) = 1399.0 \text{ kJ/kg}$$



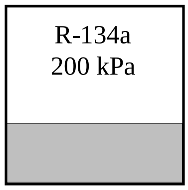
Substituting,

$$\begin{aligned} Q_{out} &= m(u_1 - u_2) \\ &= (0.0251 \text{ kg})(2728.7 - 1399.0) \text{ kJ/kg} \\ &= \mathbf{33.4 \text{ kJ}} \end{aligned}$$

5-12 A rigid tank is initially filled with superheated R-134a. Heat is transferred to the tank until the pressure inside rises to a specified value. The mass of the refrigerant and the amount of heat transfer are to be determined, and the process is to be shown on a *P*-*v* diagram. √

Assumptions **1** The tank is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions.

Analysis (*a*) We take the tank as the system. This is a closed system since no mass enters or leaves. Noting that the volume of the system is constant and thus there is no boundary work, the energy balance for this stationary closed system can be expressed as



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

$$Q_{\text{in}} = \Delta U = m(u_2 - u_1) \quad (\text{since } W = \text{KE} = \text{PE} = 0)$$

Using data from the refrigerant tables (Tables A-11 through A-13), the properties of R-134a are determined to be

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ x_1 = 0.4 \end{array}\right\} \begin{array}{ll} v_f = 0.0007532, & v_g = 0.0993 \text{ m}^3/\text{kg} \\ u_f = 36.69, & u_g = 221.43 \text{ kJ/kg} \end{array}$$

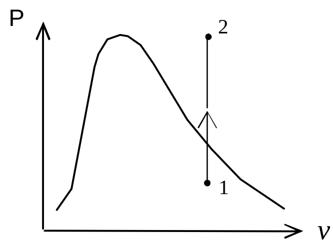
$$v_1 = v_f + x_1 v_{fg} = 0.0007532 + [0.4 \times (0.0993 - 0.0007532)] = 0.04017 \text{ m}^3/\text{kg}$$

$$u_1 = u_f + x_1 u_{fg} = 36.69 + [0.4 \times (221.43 - 36.69)] = 110.59 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 800 \text{ kPa} \\ (v_2 = v_1) \end{array}\right\} u_2 = 349.82 \text{ kJ/kg} \quad (\text{Superheated vapor})$$

Then the mass of the refrigerant is determined to be

$$m = \frac{V_1}{v_1} = \frac{0.5 \text{ m}^3}{0.04017 \text{ m}^3/\text{kg}} = \mathbf{12.45 \text{ kg}}$$



(*b*) Then the heat transfer to the tank becomes

$$\begin{aligned} Q_{\text{in}} &= m(u_2 - u_1) \\ &= (12.45 \text{ kg})(349.82 - 110.59) \text{ kJ/kg} \\ &= \mathbf{2978 \text{ kJ}} \end{aligned}$$

5-13E A rigid tank is initially filled with saturated R-134a vapor. Heat is transferred from the refrigerant until the pressure inside drops to a specified value. The final temperature, the mass of the refrigerant that has condensed, and the amount of heat transfer are to be determined. Also, the process is to be shown on a *P-v* diagram.

Assumptions **1** The tank is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions.

Analysis (*a*) We take the tank as the system. This is a closed system since no mass enters or leaves. Noting that the volume of the system is constant and thus there is no boundary work, the energy balance for this stationary closed system can be expressed as

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

$$-Q_{out} = \Delta U = m(u_2 - u_1) \quad (\text{since } W = \text{KE} = \text{PE} = 0)$$

$$Q_{out} = m(u_1 - u_2)$$

Using data from the refrigerant tables (Tables A-11 through A-13), the properties of R-134a are determined to be

$$\left.\begin{array}{l} P_1 = 120 \text{ psia} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} v_1 = v_{g@120\text{psia}} = 0.3941 \text{ ft}^3/\text{lbm} \\ u_1 = u_{g@120\text{psia}} = 105.06 \text{ Btu/lbm} \end{array}$$

R-134a
120 psia
Sat. vapor

$$\left.\begin{array}{l} P_2 = 30 \text{ psia} \\ (v_2 = v_1) \end{array}\right\} \begin{array}{ll} v_f = 0.01209, & v_g = 1.5408 \text{ ft}^3/\text{lbm} \\ u_f = 16.24, & u_g = 95.40 \text{ Btu/lbm} \end{array}$$

The final state is saturated mixture. Thus,

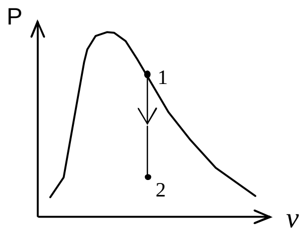
$$T_2 = T_{sat\ @\ 30\ psia} = \mathbf{15.38\ °F}$$

(*b*) The total mass and the amount of refrigerant that has condensed are

$$m = \frac{V_1}{v_1} = \frac{20 \text{ ft}^3}{0.3941 \text{ ft}^3/\text{lbm}} = 50.8 \text{ lbm}$$

$$x_2 = \frac{v_2 - v_f}{v_{fg}} = \frac{0.3941 - 0.01209}{1.5408 - 0.01209} = 0.250$$

$$m_f = (1 - x_2)m = (1 - 0.250)(50.8 \text{ lbm}) = \mathbf{38.1\ lbm}$$



Also,

$$u_2 = u_f + x_2 u_{fg} = 16.24 + [0.250 \times (95.40 - 16.24)] = 36.03 \text{ Btu/lbm}$$

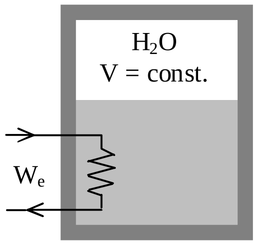
(*c*) Substituting,

$$\begin{aligned} Q_{out} &= m(u_1 - u_2) \\ &= (50.8 \text{ lbm})(105.06 - 36.03) \text{ Btu/lbm} \\ &= \mathbf{3507\ Btu} \end{aligned}$$

5-14 An insulated rigid tank is initially filled with a saturated liquid-vapor mixture of water. An electric heater in the tank is turned on, and the entire liquid in the tank is vaporized. The length of time the heater was kept on is to be determined, and the process is to be shown on a *P-v* diagram.

Assumptions **1** The tank is stationary and thus the kinetic and potential energy changes are zero. **2** The device is well-insulated and thus heat transfer is negligible. **3** The energy stored in the resistance wires, and the heat transferred to the tank itself is negligible.

Analysis We take the contents of the tank as the system. This is a closed system since no mass enters or leaves. Noting that the volume of the system is constant and thus there is no boundary work, the energy balance for this stationary closed system can be expressed as



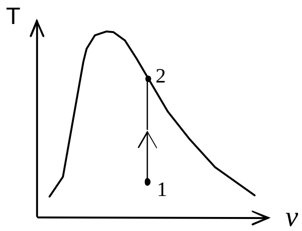
$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{e,\text{in}} = \Delta U = m(u_2 - u_1) \quad (\text{since } Q = \text{KE} = \text{PE} = 0)$$

$$VI\Delta t = m(u_2 - u_1)$$

The properties of water are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 100 \text{kPa} \\ x_1 = 0.25 \end{array}\right\} \begin{array}{l} v_f = 0.001043, \quad v_g = 1.6940 \text{m}^3/\text{kg} \\ u_f = 417.36, \quad u_{fg} = 2088.7 \text{kJ/kg} \end{array}$$



$$v_1 = v_f + x_1 v_{fg} = 0.001043 + [0.25 \times (1.6940 - 0.001043)] = 0.42428 \text{m}^3/\text{kg}$$

$$u_1 = u_f + x_1 u_{fg} = 417.36 + (0.25 \times 2088.7) = 939.5 \text{kJ/kg}$$

$$\left.\begin{array}{l} v_2 = v_1 = 0.42428 \text{m}^3/\text{kg} \\ \text{sat.vapor} \end{array}\right\} u_2 = u_{g@0.42428 \text{m}^3/\text{kg}} = 2556.7 \text{kJ/kg}$$

Substituting,

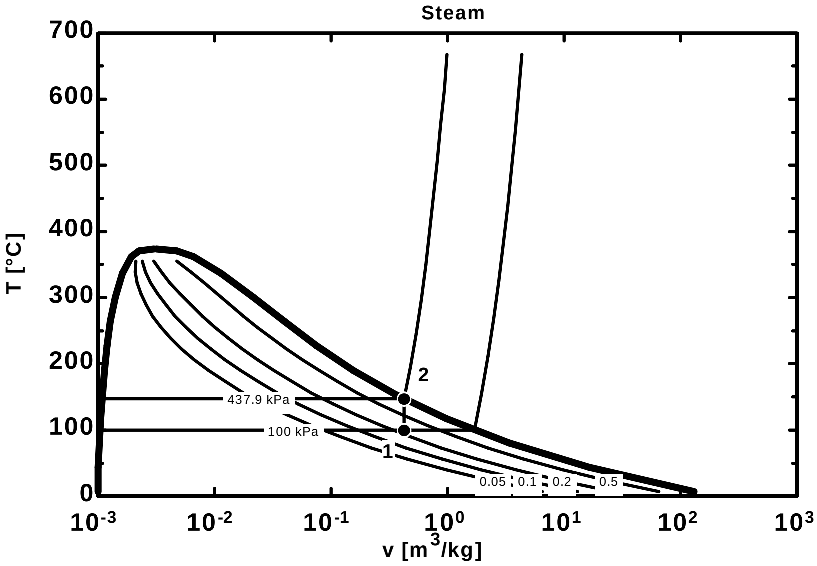
$$(110\text{V})(8\text{A})\Delta t = (5\text{kg})(2556.7 - 939.5)\,\text{kJ/kg}\left(\frac{1000\text{VA}}{1\text{kJ/s}}\right)$$

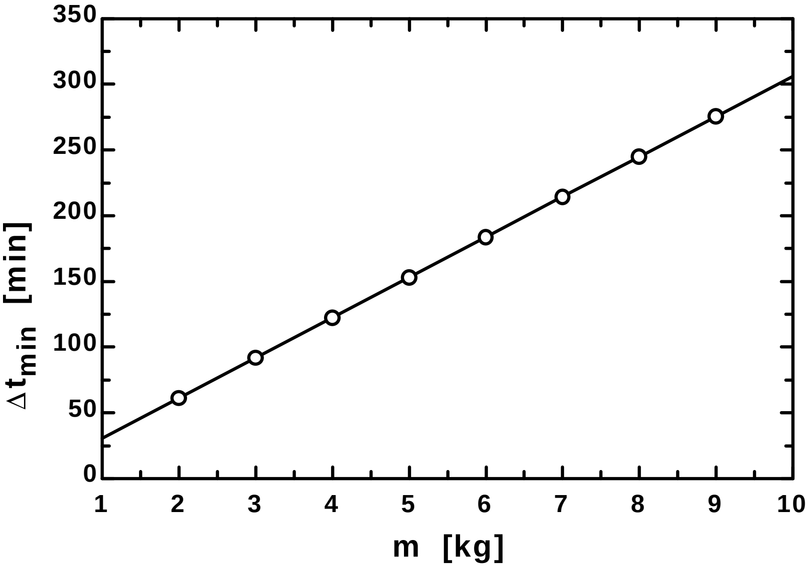
$$\Delta t = 9189 \text{s} \cong \mathbf{153.2 min}$$

5-15 Problem 5-14 is reconsidered. The effect of the initial mass of water on the length of time required to completely vaporize the liquid as the initial mass varies from 1 kg to 10 kg is to be investigated. The vaporization time is to be plotted against the initial mass.

```
PROCEDURE P2X2(v[1]:P[2],x[2])

If v[1] > V_CRIT(Steam) then
P[2]=pressure(Steam,v=v[1],x=1)
x[2]=1
else
P[2]=pressure(Steam,v=v[1],x=0)
x[2]=0
EndIf
End

"Knowns"

{m=5"[kg]"}
P[1]=100"[kPa]"
y=0.75 "moisture"
Volts=110"[V]"
I=8"[amps]"

"Solution"

"Conservation of Energy for the closed tank:"
E_dot_in-E_dot_out=DELTAE_dot
E_dot_in=W_dot_ele"[kW]"
W_dot_ele=Volts*I*CONVERT(J/s,kW)"[kW]"
E_dot_out=0"[kW]"
DELTAE_dot=m*(u[2]-u[1])/DELTAt_s"[kW]"
DELTAt_min=DELTAt_s*convert(s,min)"[min]"

"The quality at state 1 is:"
x[1]=1-y
u[1]=INTENERGY(Steam,P=P[1], x=x[1])"[kJ/kg]"
v[1]=volume(Steam,P=P[1], x=x[1])"[m^3/kg]"
T[1]=temperature(Steam,P=P[1], x=x[1])"[C]"

"Check to see if state 2 is on the saturated liquid line or saturated vapor line:"
Call P2X2(v[1]:P[2],x[2])
u[2]=INTENERGY(Steam,P=P[2], x=x[2])"[kJ/kg]"
v[2]=volume(Steam,P=P[2], x=x[2])"[m^3/kg]"
T[2]=temperature(Steam,P=P[2], x=x[2])"[C]"
```

Δt_{min} [min]	m [kg]
30.63	1
61.26	2
91.89	3
122.5	4
153.2	5
183.8	6
214.4	7
245	8
275.7	9
306.3	10

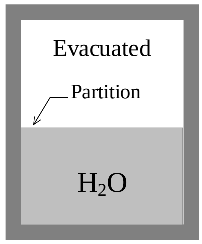

Steam
T [°C]
v [m³/kg]
437.9 kPa
100 kPa
2
1
0.05
0.1
0.2
0.5


Δt_min [min]
m [kg]

5-16 One part of an insulated tank contains compressed liquid while the other side is evacuated. The partition is then removed, and water is allowed to expand into the entire tank. The final temperature and the volume of the tank are to be determined.

Assumptions **1** The tank is stationary and thus the kinetic and potential energy changes are zero. **2** The tank is insulated and thus heat transfer is negligible. **3** There are no work interactions.

Analysis We take the entire contents of the tank as the system. This is a closed system since no mass enters or leaves. Noting that the volume of the system is constant and thus there is no boundary work, the energy balance for this stationary closed system can be expressed as



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

$$0 = \Delta U = m(u_2 - u_1) \quad (\text{since } W = Q = \text{KE} = \text{PE} = 0)$$

$$u_1 = u_2$$

The properties of water are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 600\text{kPa} \\ T_1 = 60^\circ\text{C} \end{array}\right\} \begin{array}{l} v_1 \cong v_{f@60^\circ\text{C}} = 0.001017\text{m}^3/\text{kg} \\ u_1 \cong u_{f@60^\circ\text{C}} = 251.11\text{kJ/kg} \end{array}$$

We now assume the final state in the tank is saturated liquid-vapor mixture and determine quality. This assumption will be verified if we get a quality between 0 and 1.

$$\left.\begin{array}{l} P_2 = 10\text{kPa} \\ (u_2 = u_1) \end{array}\right\} \begin{array}{ll} v_f = 0.00101, & v_g = 14.67 \text{ m}^3/\text{kg} \\ u_f = 191.82, & u_{fg} = 2246.1 \text{ kJ/kg} \end{array}$$

$$x_2 = \frac{u_2 - u_f}{u_{fg}} = \frac{251.11 - 191.82}{2246.1} = 0.0264$$

Thus,

$$T_2 = = T_{\text{sat @ 10 kPa}} = \mathbf{45.81\ ^\circ C}$$

$$v_2 = v_f + x_2 v_{fg} = 0.0010 + [0.0264 \times (14.67 - 0.00101)] = 0.388\text{m}^3/\text{kg}$$

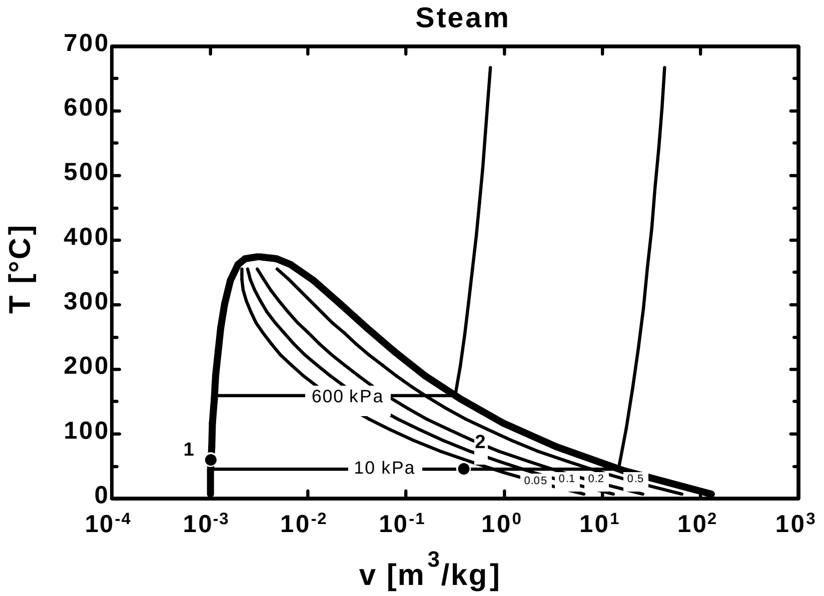
and,

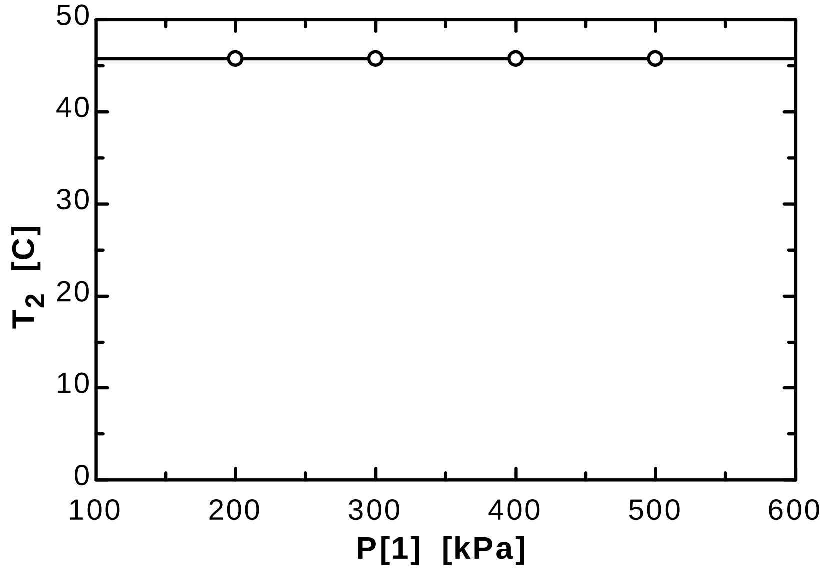
$$V = mv_2 = (2.5 \text{ kg})(0.388 \text{ m}^3/\text{kg}) = \mathbf{0.97\ m^3}$$

5-17 Problem 5-16 is reconsidered. The effect of the initial pressure of water on the final temperature in the tank as the initial pressure varies from 100 kPa to 600 kPa is to be investigated. The final temperature is to be plotted against the initial pressure.

```
"Knowns"
m=2.5"[kg]"
{P[1]=600"[kPa]"}
T[1]=60"[C]"
P[2]=10"[kPa]"

"Solution"
"Conservation of Energy for the closed tank:"
E_in-E_out=DELTAE
E_in=0"[kJ]"
E_out=0"[kJ]"

DELTAE=m*(u[2]-u[1])"[kJ]"

u[1]=INTENERGY(Steam,P=P[1], T=T[1])"[kJ/kg]"
v[1]=volume(Steam,P=P[1], T=T[1])"[m^3/kg]"

T[2]=temperature(Steam,P=P[2], u=u[2])"[kJ/kg]"
T_2=T[2]"[C]"
v[2]=volume(Steam,P=P[2], u=u[2])"[m^3/kg]"
V_total=m*v[2]"[m^3]"
```

P_1 [kPa]	T_2 [C]
100	45.79
200	45.79
300	45.79
400	45.79
500	45.79
600	45.79

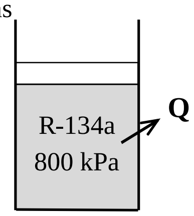



50
40
30
20
10
0
T2 [C]
100
200
300
400
500
600
P[1] [kPa]

5-18 A cylinder is initially filled with R-134a at a specified state. The refrigerant is cooled at constant pressure. The amount of heat loss is to be determined, and the process is to be shown on a *T*-*v* diagram.

Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved other than the boundary work. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

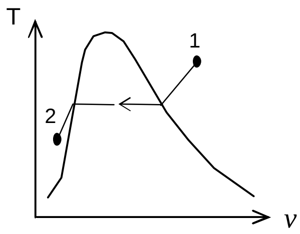


$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{out} - W_{b,out} = \Delta U = m(u_2 - u_1) \quad \text{(since KE = PE = 0)}$$

$$-Q_{out} = m(h_2 - h_1)$$

since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. The properties of R-134a are (Tables A-11 through A-13)



$$\left.\begin{array}{l} P_1 = 800\text{kPa} \\ T_1 = 60^{\circ}\text{C} \end{array}\right\} h_1 = 294.98\text{kJ/kg}$$

$$\left.\begin{array}{l} P_1 = 800\text{kPa} \\ T_1 = 20^{\circ}\text{C} \end{array}\right\} h_2 = h_{f@20^{\circ}\text{C}} = 77.26\text{kJ/kg}$$

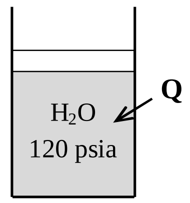
Substituting,

Q_{out} = - (5 kg)(77.26 - 294.98) kJ/kg = **1089 kJ**

5-19E A cylinder contains water initially at a specified state. The water is heated at constant pressure. The final temperature of the water is to be determined, and the process is to be shown on a *T-v* diagram.

Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** The thermal energy stored in the cylinder itself is negligible. **3** The compression or expansion process is quasi-equilibrium.

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

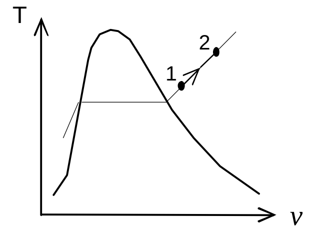


$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{in} - W_{b,out} = \Delta U = m(u_2 - u_1) \quad \text{(since KE = PE = 0)}$$

$$Q_{in} = m(h_2 - h_1)$$

since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. The properties of water are (Tables A-11E through A-13E)



$$v_1 = \frac{V_1}{m} = \frac{2\text{ft}^3}{0.5\text{lbm}} = 4\text{ft}^3/\text{lbm}$$

$$\left.\begin{aligned} P_1 &= 120\text{psia} \\ v_1 &= 4\text{ft}^3/\text{lbm} \end{aligned}\right\} h_1 = 1216.9\text{Btu/lbm}$$

Substituting,

$$200\text{ Btu} = (0.5\text{lbm})(h_2 - 1216.9)\text{Btu/lbm}$$

$$h_2 = 1616.9\text{ Btu/lbm}$$

Then,

$$\left.\begin{aligned} P_2 &= 120\text{ psia} \\ h_2 &= 1616.9\text{ Btu/lbm} \end{aligned}\right\} T_2 = \mathbf{1161.4^{\circ}F}$$

5-20 A cylinder is initially filled with saturated liquid water at a specified pressure. The water is heated electrically as it is stirred by a paddle-wheel at constant pressure. The voltage of the current source is to be determined, and the process is to be shown on a *P-v* diagram.

Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** The cylinder is well-insulated and thus heat transfer is negligible. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

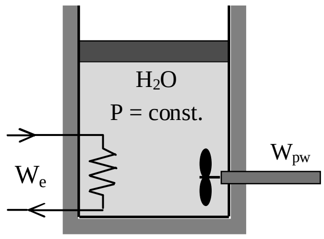
Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

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

$$W_{e,in} + W_{pw,in} - W_{b,out} = \Delta U \quad (\text{since } Q = \text{KE} = \text{PE} = 0)$$

$$W_{e,in} + W_{pw,in} = m(h_2 - h_1)$$

$$(VI\Delta t) + W_{pw,in} = m(h_2 - h_1)$$



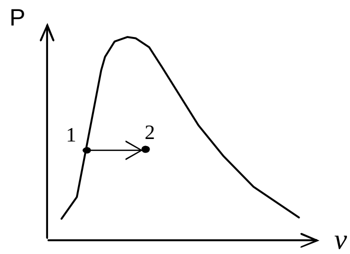
since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. The properties of water are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 150\text{ kPa} \\ \text{sat.liquid} \end{array}\right\} \begin{array}{l} h_1 = h_{f@150\text{ kPa}} = 467.11\text{ kJ/kg} \\ v_1 = v_{f@150\text{ kPa}} = 0.0010528\text{ m}^3/\text{kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 150\text{ kPa} \\ x_2 = 0.5 \end{array}\right\} h_2 = h_f + x_2 h_{fg} = 467.11 + (0.5 \times 2226.5) = 1580.36\text{ kJ/kg}$$

$$m = \frac{V_1}{v_1} = \frac{0.005\text{ m}^3}{0.0010528\text{ m}^3/\text{kg}} = 4.75\text{ kg}$$

Substituting,



$$VI\Delta t + (300\text{kJ}) = (4.75\text{kg})(1580.36 - 467.11)\text{kJ/kg}$$

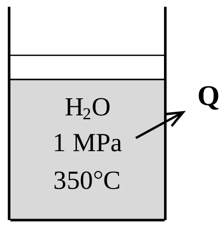
$$VI\Delta t = 4988\text{kJ}$$

$$V = \frac{4988\text{kJ}}{(8\text{A})(45 \times 60\text{s})}\left(\frac{1000\text{VA}}{1\text{kJ/s}}\right) = \mathbf{230.9\,V}$$

5-21 A cylinder is initially filled with steam at a specified state. The steam is cooled at constant pressure. The mass of the steam, the final temperature, and the amount of heat transfer are to be determined, and the process is to be shown on a *T-v* diagram.

Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved other than the boundary work. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as



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

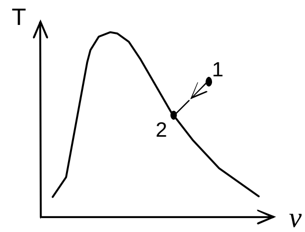
$$-Q_{\text{out}} - W_{b,\text{out}} = \Delta U = m(u_2 - u_1) \quad \text{(since KE = PE = 0)}$$

$$-Q_{\text{out}} = m(h_2 - h_1)$$

since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. The properties of water are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 1\text{MPa} \\ T_2 = 350\,^{\circ}\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.2825\text{m}^3/\text{kg} \\ h_1 = 3157.7\text{kJ/kg} \end{array}$$

$$m = \frac{V_1}{v_1} = \frac{1.5\text{m}^3}{0.2825\text{m}^3/\text{kg}} = \mathbf{5.31kg}$$



(*b*) The final temperature is determined from

$$\left.\begin{array}{l} P_2 = 1\text{MPa} \\ sat.vapor \end{array}\right\} \begin{array}{l} T_2 = T_{sat@1\text{MPa}} = \mathbf{179.91\,^{\circ}C} \\ h_2 = h_{\text{g@1MPa}} = 2778.1\text{kJ/kg} \end{array}$$

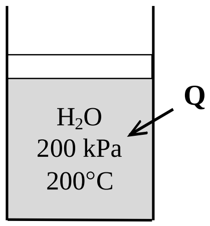
(*c*) Substituting, the energy balance gives

Q_{out} = - (5.31 kg)(2778.1 - 3157.7) kJ/kg = **2016 kJ**

5-22 [*Also solved by EES on enclosed CD***]** A cylinder equipped with an external spring is initially filled with steam at a specified state. Heat is transferred to the steam, and both the temperature and pressure rise. The final temperature, the boundary work done by the steam, and the amount of heat transfer are to be determined, and the process is to be shown on a *P-v* diagram.

Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** The thermal energy stored in the cylinder itself is negligible. **3** The compression or expansion process is quasi-equilibrium. **4** The spring is a linear spring.

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. Noting that the spring is not part of the system (it is external), the energy balance for this stationary closed system can be expressed as

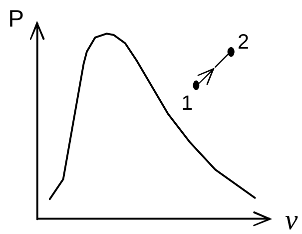


$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\\text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\\text{potential, etc. energies}}}$$

$$Q_{in} - W_{b,\text{out}} = \Delta U = m(u_2 - u_1) \quad \text{(since KE = PE = 0)}$$

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

The properties of steam are (Tables A-4 through A-6)



$$\left.\begin{aligned} P_1 &= 200\text{kPa} \\ T_1 &= 200^{\circ}\text{C} \end{aligned}\right\} \begin{aligned} v_1 &= 1.0803\text{m}^3/\text{kg} \\ u_1 &= 2654.4\text{kJ/kg} \end{aligned}$$

$$m = \frac{V_1}{v_1} = \frac{0.5\text{m}^3}{1.0803\text{m}^3/\text{kg}} = 0.463\text{kg}$$

$$v_2 = \frac{V_2}{m} = \frac{0.6\text{m}^3}{0.463\text{kg}} = 1.296\text{m}^3/\text{kg}$$

$$\left.\begin{aligned} P_2 &= 500\text{kPa} \\ v_2 &= 1.296\text{m}^3/\text{kg} \end{aligned}\right\} \begin{aligned} T_2 &= \mathbf{1131^{\circ}C} \\ u_2 &= 4321.9\text{kJ/kg} \end{aligned}$$

(*b*) The pressure of the gas changes linearly with volume, and thus the process curve on a P-V diagram will be a straight line. The boundary work during this process is simply the area under the process curve, which is a trapezoidal. Thus,

$$W_b = Area = \frac{P_1 + P_2}{2}(V_2 - V_1) = \frac{(200 + 500)\text{kPa}}{2}(0.6 - 0.5)\text{m}^3\left(\frac{1\text{kJ}}{1\text{kPa}\cdot\text{m}^3}\right) = \mathbf{35kJ}$$

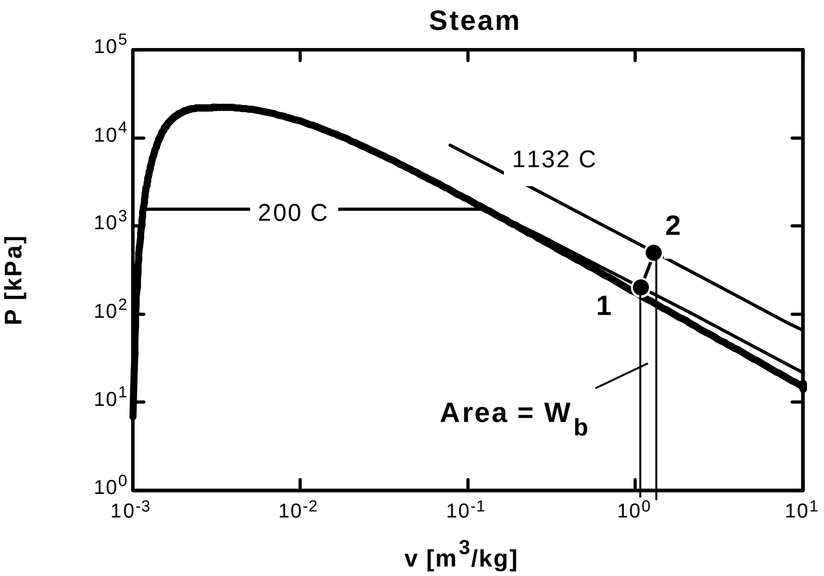
(*c*) From the energy balance we have

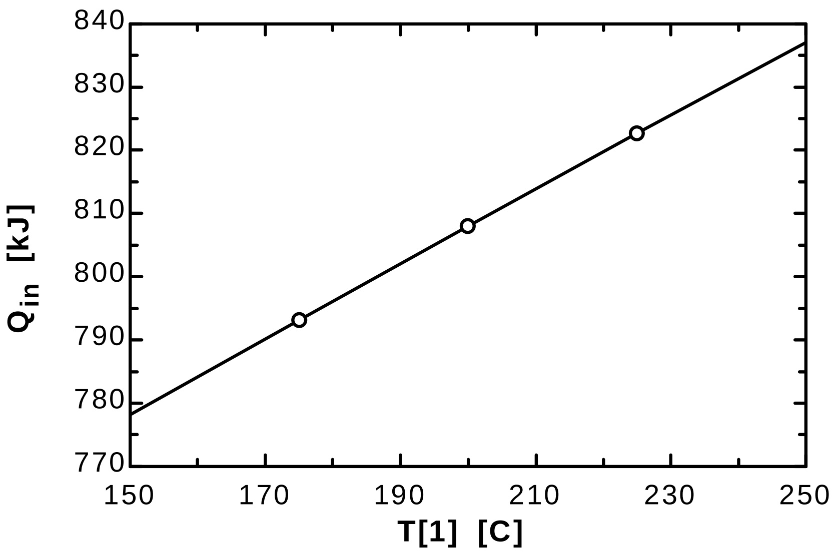
$$Q_{\text{in}} = (0.463 \text{ kg})(4321.9 - 2654.4)\text{kJ/kg} + 35 \text{ kJ} = \mathbf{807 \ kJ}$$

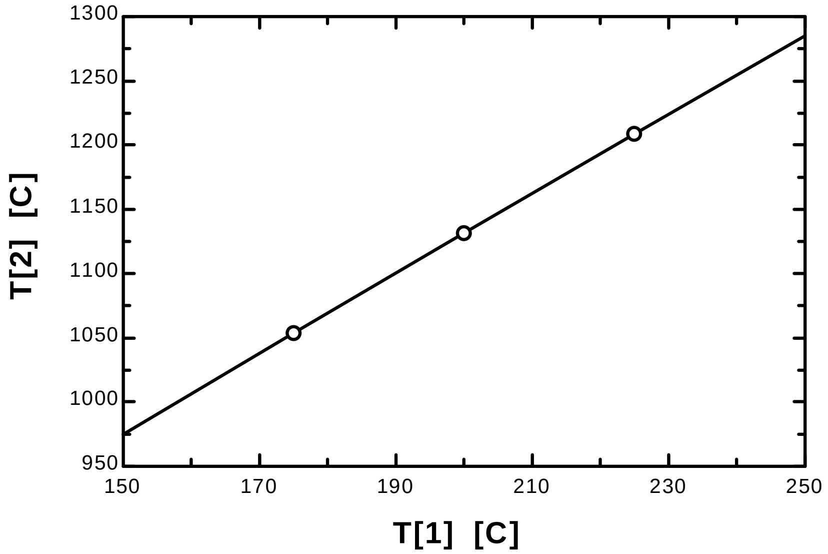
5-23 Problem 5-22 is reconsidered. The effect of the initial temperature of steam on the final temperature, the work done, and the total heat transfer as the initial temperature varies from 150°C to 250°C is to be investigated. The final results are to be plotted against the initial temperature.

```
 "The process is given by:"
"P[2]=P[1]+k*x*A/A, and as the spring moves 'x' amount, the volume changes by V[2]-V[1]."
P[2]=P[1]+(Spring_const)*(V[2] - V[1]) "P[2] is a linear function of V[2]"
"where Spring_const = k/A, the actual spring constant divided by the piston face area"

"Conservation of mass for the closed system is:"
m[2]=m[1]

"The conservation of energy for the closed system is"
"E_in - E_out = DeltaE,  neglect DeltaKE and DeltaPE for the system"
Q_in - W_out = m[1]*(u[2]-u[1])
DELTAU=m[1]*(u[2]-u[1])
"Input Data"
P[1]=200"kPa"
V[1]=0.5"m^3"
T[1]=200"C"
P[2]=500"kPa"
V[2]=0.6"m^3"

m[1]=V[1]/spvol[1]
spvol[1]=volume(Steam,T=T[1], P=P[1])
u[1]=intenergy(Steam, T=T[1], P=P[1])
spvol[2]=V[2]/m[2]

"The final temperature is:"
T[2]=temperature(Steam,P=P[2],v=spvol[2])

u[2]=intenergy(Steam, P=P[2], T=T[2])
Wnet_other = 0
W_out=Wnet_other + W_b
"W_b = integral of P[2]*dV[2] for 0.5<V[2]<0.6 and is given by:"
 W_b=P[1]*(V[2]-V[1])+Spring_const/2*(V[2]-V[1])^2
"Compare this with the result given in Example 3-13"
```

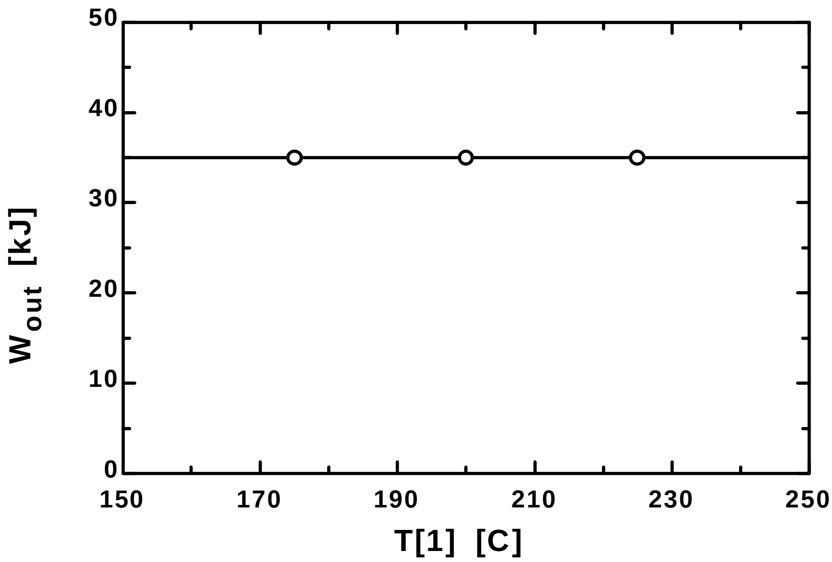
Q_{in} [kJ]	T_1 [C]	T_2 [C]	W_{out} [kJ]
778.2	150	975	35
793.2	175	1054	35
808	200	1131	35
822.7	225	1209	35
837.1	250	1285	35


Steam
1132 C
200 C
2
1
Area = W_b
P [kPa]
v [m³/kg]


Q_in [kJ]
T[1] [C]



1300
1250
1200
1150
1100
1050
1000
950
150
170
190
210
230
250
T[2] [C]
T[1] [C]

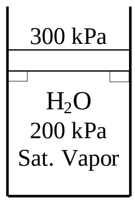


50
40
30
20
10
0
150
170
190
210
230
250
W out [kJ]
T[1] [C]

5-24 A cylinder equipped with a set of stops for the piston to rest on is initially filled with saturated water vapor at a specified pressure. Heat is transferred to water until the volume doubles. The final temperature, the boundary work done by the steam, and the amount of heat transfer are to be determined, and the process is to be shown on a *P*-*v* diagram.

Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved other than the boundary work. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as



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

$$Q_{\text{in}} - W_{b,\text{out}} = \Delta U = m(u_3 - u_1) \quad (\text{since KE = PE = 0})$$

$$Q_{\text{in}} = m(u_3 - u_1) + W_{b,\text{out}}$$

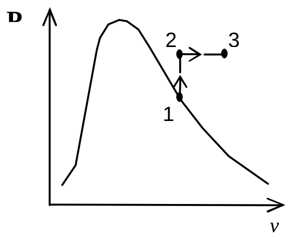
The properties of steam are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 200\text{kPa} \\ sat.vapor \end{array}\right\} \begin{array}{l} v_1 = v_{g@200\text{kPa}} = 0.8857\text{m}^3/\text{kg} \\ u_1 = u_{g@200\text{kPa}} = 2529.5\text{kJ/kg} \end{array}$$

$$m = \frac{V_1}{v_1} = \frac{0.5\text{m}^3}{0.8857\text{m}^3/\text{kg}} = 0.5645\text{kg}$$

$$v_3 = \frac{V_3}{m} = \frac{1\text{m}^3}{0.5645\text{kg}} = 1.7715\text{m}^3/\text{kg}$$

$$\left.\begin{array}{l} P_3 = 300\text{kPa} \\ v_3 = 1.7715\text{m}^3/\text{kg} \end{array}\right\} \begin{array}{l} T_3 = \mathbf{878.9\,^{\circ}C} \\ u_3 = 3813.8\text{kJ/kg} \end{array}$$



(*b*) The work done during process 1-2 is zero (since *V* = const) and the work done during the constant pressure process 2-3 is

$$W_{b,\text{out}} = \int_2^3 P\,dV = P(V_3 - V_2) = (300\text{ kPa})(1.0 - 0.5)\text{m}^3 \left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) = \mathbf{150\ \ kJ}$$

(*c*) Heat transfer is determined from the energy balance,

$$Q_{\text{in}} = m(u_3 - u_1) + W_{b,\text{out}}$$
$$= (0.5645\text{ kg})(3813.8 - 2529.5)\text{ kJ/kg} + 150\text{ kJ} = \mathbf{875.0\ \ kJ}$$

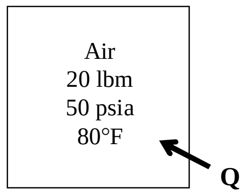
Closed System Energy Analysis: Ideal Gases

5-25C No, it isn't. This is because the first law relation Q - W = ΔU reduces to W = 0 in this case since the system is adiabatic (Q = 0) and ΔU = 0 for the isothermal processes of ideal gases. Therefore, this adiabatic system cannot receive any net work at constant temperature.

5-26E The air in a rigid tank is heated until its pressure doubles. The volume of the tank and the amount of heat transfer are to be determined.

Assumptions **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta pe \cong \Delta ke \cong 0$. **3** Constant specific heats at room temperature can be used for air. This assumption results in negligible error in heating and air-conditioning applications.

Properties The gas constant of air is R = 0.3704 psia.ft^3/lbm.R (Table A-1E).



Analysis (a) The volume of the tank can be determined from the ideal gas relation,

$$V = \frac{mRT_1}{P_1} = \frac{(20\text{ lbm})(0.3704\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(540\text{ R})}{50\text{ psia}} = \mathbf{80.0\ ft^3}$$

(b) We take the air in the tank as our system. The energy balance for this stationary closed system can be expressed as

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

$$Q_{in} = \Delta U$$

$$Q_{in} = m(u_2 - u_1) \cong mC_v(T_2 - T_1)$$

The final temperature of air is

$$\frac{P_1 V}{T_1} = \frac{P_2 V}{T_2} \longrightarrow T_2 = \frac{P_2}{P_1} T_1 = 2 \times (540\text{ R}) = 1080\text{ R}$$

The internal energies are (Table A-17E)

$$u_1 = u_{@\,540\text{ R}} = 92.04\text{ Btu / lbm}$$

$$u_2 = u_{@\,1080\text{ R}} = 186.93\text{ Btu / lbm}$$

Substituting,

Q_{in} = (20 lbm)(186.93 - 92.04)Btu/lbm = **1898 Btu**

Alternative solutions The specific heat of air at the average temperature of T_{ave} = (540+1080)/2= 810 R = 350°F is, from Table A-2Eb, $C_{v,ave}$ = 0.175 Btu/lbm.R. Substituting,

Q_{in} = (20 lbm)(0.175 Btu/lbm.R)(1080 - 540) R = **1890 Btu**

Discussion Both approaches resulted in almost the same solution in this case.

5-27 The hydrogen gas in a rigid tank is cooled until its temperature drops to 300 K. The final pressure in the tank and the amount of heat transfer are to be determined.

Assumptions **1** Hydrogen is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -240°C and 1.30 MPa. **2** The tank is stationary, and thus the kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$.

Properties The gas constant of hydrogen is R = 4.124 kPa.m^3/kg.K (Table A-1). The constant volume specific heat of hydrogen at the average temperature of 400 K is , $C_{v,ave}$ = 10.352 kJ/kg.K (Table A-2).

Analysis (*a*) The final pressure of hydrogen can be determined from the ideal gas relation,

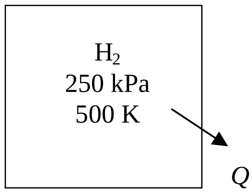
$$\frac{P_1 V}{T_1} = \frac{P_2 V}{T_2} \longrightarrow P_2 = \frac{T_2}{T_1} P_1 = \frac{300\text{ K}}{500\text{ K}}(250\text{ kPa}) = \mathbf{150\ kPa}$$

(*b*) We take the hydrogen in the tank as the system. This is a *closed system* since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

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

$$-Q_{out} = \Delta U$$

$$Q_{out} = -\Delta U = -m(u_2 - u_1) \cong mC_v(T_1 - T_2)$$



where

$$m = \frac{P_1 V}{RT_1} = \frac{(250\text{ kPa})(3.0\text{ m}^3)}{(4.124\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(500\text{ K})} = 0.3637\text{ kg}$$

Substituting into the energy balance,

Q_{out} = (0.3637 kg)(10.352 kJ/kg·K)(500 - 300)K = **753.0 kJ**

5-28 A resistance heater is to raise the air temperature in the room from 7 to 23°C within 20 min. The required power rating of the resistance heater is to be determined. √

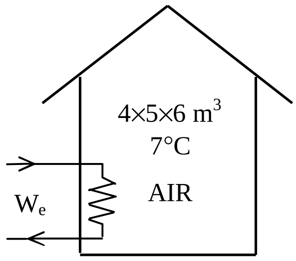
Assumptions **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats at room temperature can be used for air. This assumption results in negligible error in heating and air-conditioning applications. **4** Heat losses from the room are negligible. **5** The room is air-tight so that no air leaks in and out during the process.

Properties The gas constant of air is R = 0.287 kPa.m^3/kg.K (Table A-1). Also, C_v = 0.718 kJ/kg.K for air at room temperature (Table A-2).

Analysis We take the air in the room to be the system. This is a closed system since no mass crosses the system boundary. The energy balance for this stationary constant-volume closed system can be expressed as

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

$$W_{e,in} = \Delta U \cong mC_{v,ave}(T_2 - T_1) \quad (\text{since } Q = KE = PE = 0)$$



or,

$$\dot{W}_{e,in}\Delta t = mC_{v,\,ave}(T_2 - T_1)$$

The mass of air is

$$V = 4\times5\times6 = 120 \text{ m}^3$$

$$m = \frac{P_1V}{RT_1} = \frac{(100 \text{ kPa})(120 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(280 \text{ K})} = 149.3 \text{ kg}$$

Substituting, the power rating of the heater becomes

$$\dot{W}_{e,in} = \frac{(149.3 \text{ kg})(0.718 \text{ kJ/kg}\cdot{}^{o}\text{C})(23-7)^{o}\text{C}}{15\times60 \text{ s}} = \mathbf{1.91\ kW}$$

Discussion In practice, the pressure in the room will remain constant during this process rather than the volume, and some air will leak out as the air expands. As a result, the air in the room will undergo a constant pressure expansion process. Therefore, it is more proper to be conservative and to use ΔH instead of using ΔU in heating and air-conditioning applications.

5-29 A room is heated by a radiator, and the warm air is distributed by a fan. Heat is lost from the room. The time it takes for the air temperature to rise to 20°C is to be determined. √

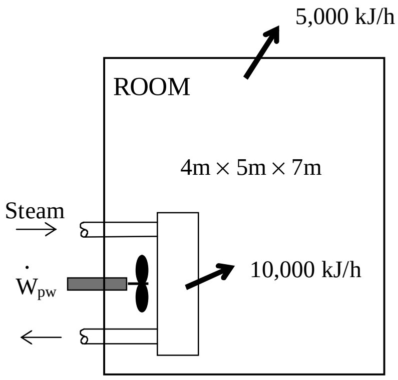
Assumptions **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats at room temperature can be used for air. This assumption results in negligible error in heating and air-conditioning applications. **4** The local atmospheric pressure is 100 kPa. **5** The room is air-tight so that no air leaks in and out during the process.

Properties The gas constant of air is R = 0.287 kPa.m^3/kg.K (Table A-1). Also, C_v = 0.718 kJ/kg.K for air at room temperature (Table A-2).

Analysis We take the air in the room to be the system. This is a closed system since no mass crosses the system boundary. The energy balance for this stationary constant-volume closed system can be expressed as

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

$$Q_{in} + W_{fan,in} - Q_{out} = \Delta U \cong mC_{v,ave}(T_2 - T_1) \text{ (since } KE = PE = 0)$$



or,

$$(\dot{Q}_{in} + \dot{W}_{fan,in} - \dot{Q}_{out})\Delta t = mC_{v,ave}(T_2 - T_1)$$

The mass of air is

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

$$m = \frac{P_1 V}{RT_1} = \frac{(100 \text{ kPa})(140 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(283 \text{ K})} = 172.4 \text{ kg}$$

Using the C_v value at room temperature,

$$\left[(10{,}000 - 5{,}000)/3600 \text{ kJ/s} + 0.1 \text{ kJ/s}\right]\Delta t = (172.4\text{kg})(0.718 \text{ kJ/kg}\cdot^\circ\text{C})(20-10)^\circ\text{C}$$

It yields

Δt = **831 s**

Discussion In practice, the pressure in the room will remain constant during this process rather than the volume, and some air will leak out as the air expands. As a result, the air in the room will undergo a constant pressure expansion process. Therefore, it is more proper to be conservative and to using ΔH instead of use ΔU in heating and air-conditioning applications.

5-30 A student living in a room turns her 150-W fan on in the morning. The temperature in the room when she comes back 10 h later is to be determined. √

Assumptions **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats at room temperature can be used for air. This assumption results in negligible error in heating and air-conditioning applications. **4** All the doors and windows are tightly closed, and heat transfer through the walls and the windows is disregarded.

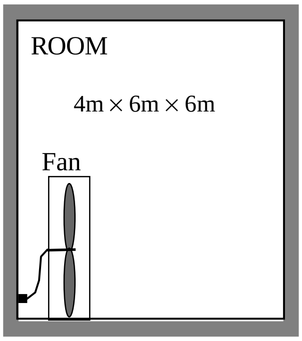
Properties The gas constant of air is R = 0.287 kPa.m^3/kg.K (Table A-1). Also, C_v = 0.718 kJ/kg.K for air at room temperature (Table A-2).

Analysis We take the room as the system. This is a *closed system* since the doors and the windows are said to be tightly closed, and thus no mass crosses the system boundary during the process. The energy balance for this system can be expressed as

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

$$W_{e,in} = \Delta U$$

$$W_{e,in} = m(u_2 - u_1) \cong mC_v(T_2 - T_1)$$



The mass of air is

$$V = 4 \times 6 \times 6 = 144 \text{ m}^3$$

$$m = \frac{P_1 V}{RT_1} = \frac{(100 \text{ kPa})(144 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(288 \text{ K})} = 174.2 \text{ kg}$$

The electrical work done by the fan is

$$W_e = \dot{W}_e \Delta t = (0.15 \text{ kJ/s})(10 \times 3600 \text{ s}) = 5400 \text{ kJ}$$

Substituting and using the C_v value at room temperature,

$$5400 \text{ kJ} = (174.2 \text{ kg})(0.718 \text{ kJ/kg} \cdot °\text{C})(T_2 - 15)°\text{C}$$

$$T_2 = \mathbf{58.2°C}$$

5-31E A paddle wheel in an oxygen tank is rotated until the pressure inside rises to 20 psia while some heat is lost to the surroundings. The paddle wheel work done is to be determined.

Assumptions **1** Oxygen is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -181°F and 736 psia. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** The energy stored in the paddle wheel is negligible. **4** This is a rigid tank and thus its volume remains constant.

Properties The gas constant and molar mass of oxygen are R = 0.3353 psia.ft^3/lbm.R and M = 32 lbm/lbmol (Table A-1E). The specific heat of oxygen at the average temperature of T_{ave} = (735+540)/2= 638 R is $C_{v,ave}$ = 0.160 Btu/lbm.R (Table A-2E).

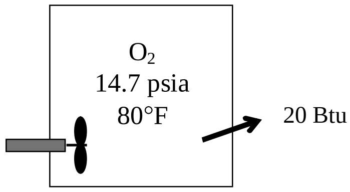
Analysis We take the oxygen in the tank as our system. This is a *closed system* since no mass enters or leaves. The energy balance for this system can be expressed as

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

$$W_{pw,in} - Q_{out} = \Delta U$$

$$W_{pw,in} = Q_{out} + m(u_2 - u_1)$$

$$\cong Q_{out} + mC_v(T_2 - T_1)$$



The final temperature and the mass of oxygen are

$$\frac{P_1 V}{T_1} = \frac{P_2 V}{T_2} \longrightarrow T_2 = \frac{P_2}{P_1} T_1 = \frac{20 \text{ psia}}{14.7 \text{ psia}}(540 \text{ R}) = 735 \text{ R}$$

$$m = \frac{P_1 V}{R T_1} = \frac{(14.7 \text{ psia})(10 \text{ ft}^3)}{(0.3353 \text{ psia}\cdot\text{ft}^3/\text{lbmol}\cdot\text{R})(540 \text{ R})} = 0.812 \text{ lbm}$$

Substituting,

$W_{pw,in}$ = (20 Btu) + (0.812 lbm)(0.160 Btu/lbm.R)(735 - 540) R = **45.3 Btu**

Discussion Note that a fan actually causes the internal temperature of a confined space to rise. In fact, a 100-W fan supplies a room with as much energy as a 100-W resistance heater.

5-32 One part of an insulated rigid tank contains an ideal gas while the other side is evacuated. The final temperature and pressure in the tank are to be determined when the partition is removed.

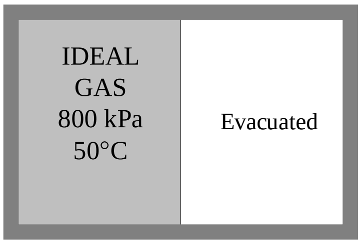
Assumptions **1** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **2** The tank is insulated and thus heat transfer is negligible.

Analysis We take the entire tank as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system can be expressed as

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

$$0 = \Delta U = m(u_2 - u_1)$$

$$u_2 = u_1$$



Therefore,

$$T_2 = T_1 = \mathbf{50°C}$$

Since $u = u(T)$ for an ideal gas. Then,

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \longrightarrow P_2 = \frac{V_1}{V_2} P_1 = \frac{1}{2}(800 \text{ kPa}) = \mathbf{400 kPa}$$

5-33 A cylinder equipped with a set of stops for the piston to rest on is initially filled with helium gas at a specified state. The amount of heat that must be transferred to raise the piston is to be determined. √

Assumptions **1** Helium is an ideal gas with constant specific heats. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** There are no work interactions involved. **4** The thermal energy stored in the cylinder itself is negligible.

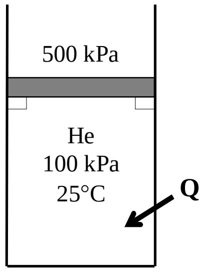
Properties The specific heat of helium at room temperature is C_v = 3.1156 kJ/kg.K (Table A-2).

Analysis We take the helium gas in the cylinder as the system. This is a closed system since no mass crosses the boundary of the system. The energy balance for this constant volume closed system can be expressed as

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

$$Q_{in} = \Delta U = m(u_2 - u_1)$$

$$Q_{in} = m(u_2 - u_1) = mC_v(T_2 - T_1)$$



The final temperature of helium can be determined from the ideal gas relation to be

$$\frac{P_1 V}{T_1} = \frac{P_2 V}{T_2} \longrightarrow T_2 = \frac{P_2}{P_1} T_1 = \frac{500\text{ kPa}}{100\text{ kPa}}(298\text{ K}) = 1490\text{ K}$$

Substituting into the energy balance relation gives

Q_{in} = (0.5 kg)(3.1156 kJ/kg·K)(1490 - 298)K = **1857 kJ**

5-34 An insulated cylinder is initially filled with air at a specified state. A paddle-wheel in the cylinder stirs the air at constant pressure. The final temperature of air is to be determined.

Assumptions **1** Air is an ideal gas with variable specific heats. **2** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **3** There are no work interactions involved other than the boundary work. **4** The cylinder is well-insulated and thus heat transfer is negligible. **5** The thermal energy stored in the cylinder itself and the paddle-wheel is negligible. **6** The compression or expansion process is quasi-equilibrium.

Properties The gas constant of air is R = 0.287 kPa.m^3/kg.K (Table A-1). Also, C_p = 1.005 kJ/kg.K for air at room temperature (Table A-2). The enthalpy of air at the initial temperature is

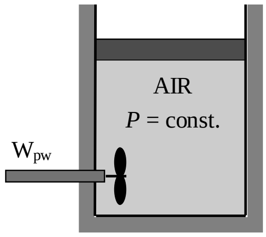
$$h_1 = h_{@298\text{ K}} = 298.18 \text{ kJ/kg}$$

Analysis We take the air in the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

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

$$W_{pw,in} - W_{b,out} = \Delta U$$

$$W_{pw,in} = m(h_2 - h_1)$$



since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. The mass of air is

$$m = \frac{P_1 V}{RT_1} = \frac{(400 \text{ kPa})(0.1 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298 \text{ K})} = 0.468 \text{ kg}$$

Substituting into the energy balance,

$$15 \text{ kJ} = (0.468 \text{ kg})(h_2 - 298.18 \text{ kJ/kg})$$
$$h_2 = 330.23 \text{ kJ/kg}$$

From Table A-17,

$$T_2 = \mathbf{329.9\ K}$$

Alternative solution Using specific heats at room temperature, C_p = 1.005 kJ/kg.°C, the final temperature is determined to be

$$W_{pw,in} = m(h_2 - h_1) \cong mC_p(T_2 - T_1)$$
$$15 \text{ kJ} = (0.468 \text{ kg})(1.005 \text{ kJ/kg.°C})(T_2 - 25)\text{°C}$$

which gives

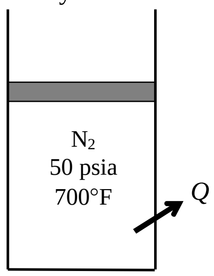
$$T_2 = \mathbf{56.9°C}$$

5-35E A cylinder is initially filled with nitrogen gas at a specified state. The gas is cooled by transferring heat from it. The amount of heat transfer is to be determined.

Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved other than the boundary work. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium. **5** Nitrogen is an ideal gas with constant specific heats.

Properties The specific heat of nitrogen at the average temperature of T_{ave} = (700+ 140)/2 = 420°F is C_{pave} = 0.252 Btu/lbm.°F (Table A-2Eb).

Analysis We take the nitrogen gas in the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this closed system can be expressed as



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

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

$$-Q_{out} = m(h_2 - h_1) = mC_p(T_2 - T_1)$$

since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process.

The mass of nitrogen is

$$m = \frac{P_1 V}{RT_1} = \frac{(50\text{ psia})(25\text{ ft}^3)}{(0.3830\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(1160\text{ R})} = 2.814\text{ lbm}$$

Substituting,

Q_{out} = (2.814 lbm)(0.252 Btu/lbm.°F)(700 - 140)°F = **397 Btu**

5-36 A cylinder is initially filled with air at a specified state. Air is heated electrically at constant pressure, and some heat is lost in the process. The amount of electrical energy supplied is to be determined. √

Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** Air is an ideal gas with variable specific heats. **3** The thermal energy stored in the cylinder itself and the resistance wires is negligible. **4** The compression or expansion process is quasi-equilibrium.

Properties The initial and final enthalpies of air are (Table A-17)

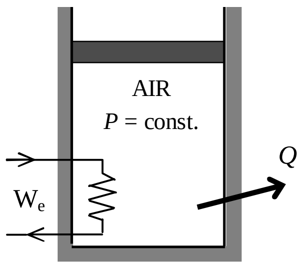
$$h_1 = h_{@\,298\text{ K}} = 298.18\text{ kJ/kg}$$
$$h_2 = h_{@\,350\text{ K}} = 350.49\text{ kJ/kg}$$

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this closed system can be expressed as

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

$$W_{e,in} - Q_{out} - W_{b,out} = \Delta U$$

$$W_{e,in} = m(h_2 - h_1) + Q_{out}$$



since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. Substituting,

$$W_{e,in} = (15\text{ kg})(350.49 - 298.18)\text{kJ/kg} + (60\text{ kJ}) = 845\text{ kJ}$$

or,

$$W_{e,in} = (845\text{kJ})\left(\frac{1\text{ kWh}}{3600\text{ kJ}}\right) = \mathbf{0.235\ kWh}$$

Alternative solution The specific heat of air at the average temperature of T_{ave} = (25+ 77)/2 51°C = 324 K is, from Table A-2b, $C_{p,ave}$ = 1.0065 kJ/kg.°C. Substituting,

$$W_{e,in} = mC_p(T_2 - T_1) + Q_{out}$$
$$= (15\text{ kg})(1.0065\text{ kJ/kg.°C})(77 - 25)\text{°C} + 60\text{ kJ} = 845\text{ kJ}$$

or,

$$W_{e,in} = (845\text{ kJ})\left(\frac{1\text{ kWh}}{3600\text{ kJ}}\right) = \mathbf{0.235\ kWh}$$

Discussion Note that for small temperature differences, both approaches give the same result.

5-37 An insulated cylinder initially contains CO_2 at a specified state. The CO_2 is heated electrically for 10 min at constant pressure until the volume doubles. The electric current is to be determined.

Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** The CO_2 is an ideal gas with constant specific heats. **3** The thermal energy stored in the cylinder itself and the resistance wires is negligible. **4** The compression or expansion process is quasi-equilibrium.

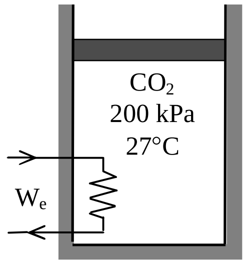
Properties The gas constant and molar mass of CO_2 are R = 0.1889 kPa.m^3/kg.K and M = 44 kg/kmol (Table A-1). The specific heat of CO_2 at the average temperature of T_{ave} = (300 + 600)/2 = 450 K is $C_{p,ave}$ = 0.978 kJ/kg.°C (Table A-2b).

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this closed system can be expressed as

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

$$W_{e,in} - W_{b,out} = \Delta U$$

$$W_{e,in} = m(h_2 - h_1) \cong mC_p(T_2 - T_1)$$



since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. The final temperature of CO_2 is

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2} \longrightarrow T_2 = \frac{P_2}{P_1}\frac{V_2}{V_1}T_1 = 1\times 2\times(300\text{K}) = 600\text{K}$$

The mass of CO_2 is

$$m = \frac{P_1V_1}{RT_1} = \frac{(200\text{ kPa})(0.3\text{ m}^3)}{(0.1889\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{ K})} = 1.059\text{ kg}$$

Substituting,

$$W_{e,in} = (1.059\text{ kg})(0.978\text{ kJ/kg.K})(600 - 300)\text{K} = 311\text{ kJ}$$

Then,

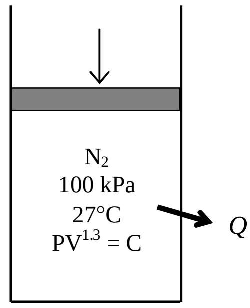
$$I = \frac{W_{e,in}}{V\Delta t} = \frac{311\text{kJ}}{(110\text{V})(10\times 60\text{s})}\left(\frac{1000\text{VA}}{1\text{kJ/s}}\right) = \mathbf{4.71A}$$

5-38 A cylinder initially contains nitrogen gas at a specified state. The gas is compressed polytropically until the volume is reduced by one-half. The work done and the heat transfer are to be determined.

Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** The N_2 is an ideal gas with constant specific heats. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

Properties The gas constant of N_2 are R = 0.2968 kPa.m^3/kg.K (Table A-1). The C_v value of N_2 at the average temperature (369+300)/2 = 335 K is 0.744 kJ/kg.K (Table A-2b).

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass crosses the system boundary. The energy balance for this closed system can be expressed as



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

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

$$W_{b,in} - Q_{out} = mC_v(T_2 - T_1)$$

The final pressure and temperature of nitrogen are

$$P_2V_2^{1.3} = P_1V_1^{1.3} \longrightarrow P_2 = \left(\frac{V_1}{V_2}\right)^{1.3} P_1 = 2^{1.3}(100\text{kPa}) = 246.2\text{kPa}$$

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2} \longrightarrow T_2 = \frac{P_2}{P_1}\frac{V_2}{V_1}T_1 = \frac{246.2\text{kPa}}{100\text{kPa}} \times 0.5 \times (300\text{K}) = 369.3\text{K}$$

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

$$W_{b,in} = -\int_1^2 P dV = -\frac{P_2V_2 - P_1V_1}{1-n} = -\frac{mR(T_2 - T_1)}{1-n}$$

$$= -\frac{(0.8\text{ kg})(0.2968\text{ kJ/kg}\cdot\text{K})(369.3 - 300)\text{K}}{1 - 1.3} = \mathbf{54.8\ kJ}$$

Substituting into the energy balance gives

$$Q_{out} = W_{b,in} - mC_v(T_2 - T_1)$$

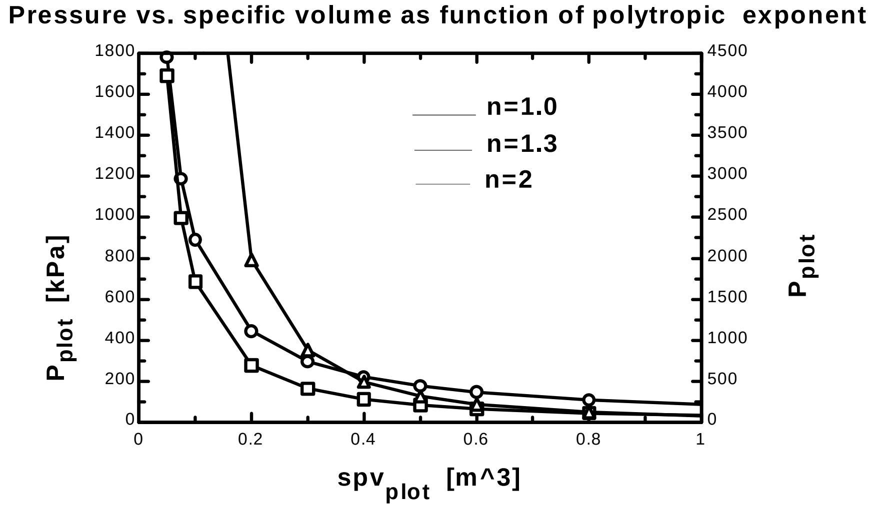
$$= 54.8\text{ kJ} - (0.8\text{ kg})(0.744\text{ kJ/kg.K})(369.3 - 360)\text{K}$$

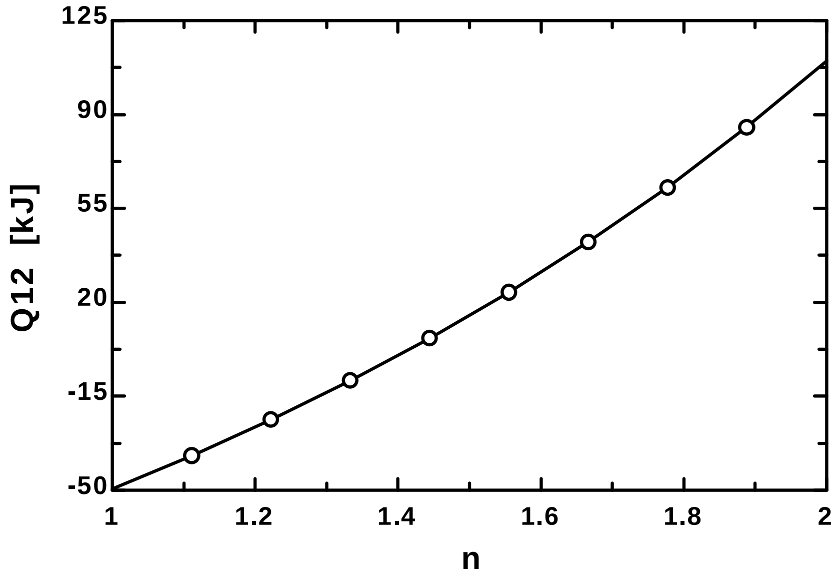
$$= \mathbf{13.6\ kJ}$$

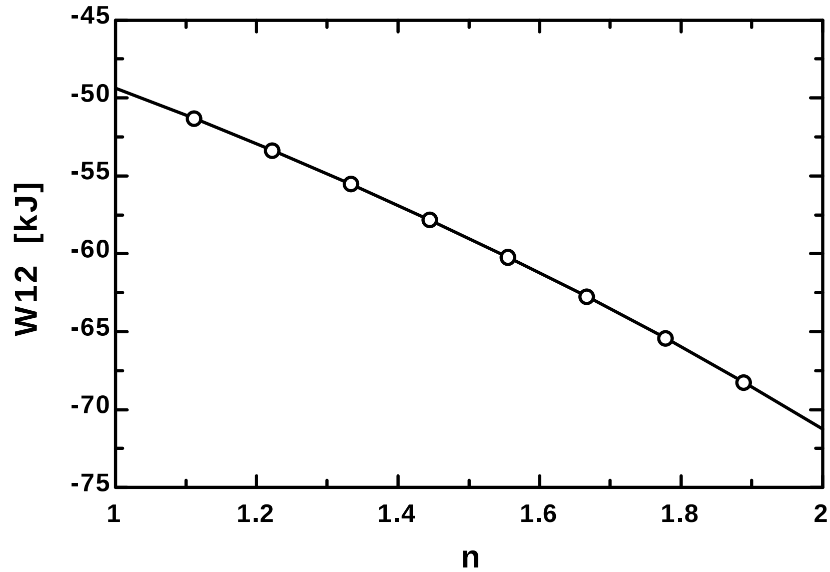
5-39 Problem 5-38 is reconsidered. The process is to be plotted on a P-V diagram, and the effect of the polytropic exponent n on the boundary work and heat transfer as the polytropic exponent varies from 1.1 to 1.6 is to be investigated. The boundary work and the heat transfer are to be plotted versus the polytropic exponent.

```
Procedure     Work(P[2],V[2],P[1],V[1],n:W12)
If n=1 then
W12=P[1]*V[1]*ln(V[2]/V[1])"[kJ]"
Else
W12=(P[2]*V[2]-P[1]*V[1])/(1-n)"[kJ]"
endif
End

"Input Data"
Vratio=0.5  "V[2]/V[1] = Vratio"
n=1.3       "Polytropic exponent"
P[1] = 100"[kPa]"
T[1] = (27+273)"[K]"
m=0.8"[kg]"
MM=molarmass(nitrogen)"[kg/kmol]"
R_u=8.314"[kJ/kmol-K]"
R=R_u/MM"[kJ/kg-K]"
V[1]=m*R*T[1]/P[1]"[m^3]"

"Process equations"
V[2]=Vratio*V[1]
P[2]*V[2]/T[2]=P[1]*V[1]/T[1]"The combined ideal gas law for
states 1 and 2 plus the polytropic process relation give P[2] and T[2]"
P[2]*V[2]^n=P[1]*V[1]^n

"Conservation of Energy for the closed system:"
"E_in - E_out = DeltaE,  we neglect Delta KE and Delta PE for the system, the nitrogen."

Q12 - W12 = m*(u[2]-u[1])
u[1]=intenergy(N2, T=T[1])"[kJ/kg]" "internal energy for nitrogen as an ideal gas, kJ/kg"
u[2]=intenergy(N2, T=T[2])"[kJ/kg]"
Call Work(P[2],V[2],P[1],V[1],n:W12)

"The following is required for the P-v plots"
{P_plot*spv_plot/T_plot=P[1]*V[1]/m/T[1]"The combined ideal gas law for
states 1 and 2 plus the polytropic process relation give P[2] and T[2]"
P_plot*spv_plot^n=P[1]*(V[1]/m)^n}
{spV_plot=R*T_plot/P_plot"[m^3]"}
```

n	Q12 [kJ]	W12 [kJ]
1	-49.37	-49.37
1.111	-37	-51.32
1.222	-23.59	-53.38
1.333	-9.067	-55.54
1.444	6.685	-57.82
1.556	23.81	-60.23
1.667	42.48	-62.76
1.778	62.89	-65.43
1.889	85.27	-68.25
2	109.9	-71.23


Pressure vs. specific volume as function of polytropic exponent
n=1.0
n=1.3
n=2
P_{plot} [kPa]
P_{plot}
spv_{plot} [m^3]
0
200
400
600
800
1000
1200
1400
1600
1800
500
1000
1500
2000
2500
3000
3500
4000
4500
0.2
0.4
0.6
0.8
1


Q12 [kJ]
n
125
90
55
20
-15
-50
1
1.2
1.4
1.6
1.8
2


W12 [kJ]
n
-45
-50
-55
-60
-65
-70
-75
1
1.2
1.4
1.6
1.8
2

5-40 It is observed that the air temperature in a room heated by electric baseboard heaters remains constant even though the heater operates continuously when the heat losses from the room amount to 8,000 kJ/h. The power rating of the heater is to be determined.

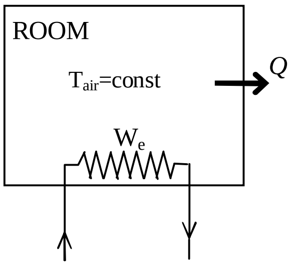
Assumptions **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** The temperature of the room is said to remain constant during this process.

Analysis We take the room as the system. This is a closed system since no mass crosses the boundary of the system. The energy balance for this system reduces to

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

$$W_{e,in} - Q_{out} = \Delta U = 0$$

$$W_{e,in} = Q_{out}$$



since $\Delta U = mC_v\Delta T = 0$ for isothermal processes of ideal gases. Thus,

$$\dot{W}_{e,in} = \dot{Q}_{out} = (6500 \text{ kJ/h})\left(\frac{1 \text{ kW}}{3600 \text{ kJ/h}}\right) = \mathbf{1.81\ kW}$$

5-41E A cylinder initially contains air at a specified state. Heat is transferred to the air, and air expands isothermally. The boundary work done is to be determined.

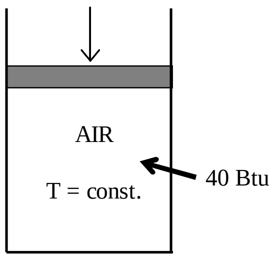
Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** The air is an ideal gas with constant specific heats. **3** The compression or expansion process is quasi-equilibrium.

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass crosses the system boundary. The energy balance for this closed system can be expressed as

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

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

$$= mC_v(T_2 - T_1) = 0$$



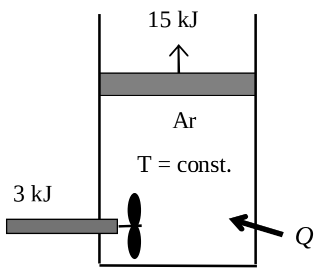
since $u = u(T)$ for ideal gases, and thus $u_2 = u_1$ when $T_1 = T_2$. Therefore,

$$W_{b,out} = Q_{in} = \mathbf{40\ Btu}$$

5-42 A cylinder initially contains argon gas at a specified state. The gas is stirred while being heated and expanding isothermally. The amount of heat transfer is to be determined.

Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** The air is an ideal gas with constant specific heats. **3** The compression or expansion process is quasi-equilibrium.

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass crosses the system boundary. The energy balance for this closed system can be expressed as



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

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

$$= mC_v(T_2 - T_1) = 0$$

since $u = u(T)$ for ideal gases, and thus $u_2 = u_1$ when $T_1 = T_2$. Therefore,

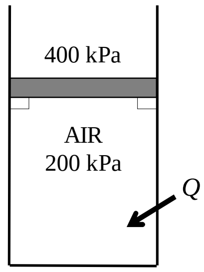
$$Q_{in} = W_{b,out} - W_{pw,in} = 15 - 3 = \mathbf{12\ kJ}$$

5-43 A cylinder equipped with a set of stops for the piston is initially filled with air at a specified state. Heat is transferred to the air until the volume doubled. The work done by the air and the amount of heat transfer are to be determined, and the process is to be shown on a *P-v* diagram.

Assumptions **1** Air is an ideal gas with variable specific heats. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

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

Analysis We take the air in the cylinder as the system. This is a closed system since no mass crosses the boundary of the system. The energy balance for this closed system can be expressed as



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

$$Q_{in} - W_{b,out} = \Delta U = m(u_3 - u_1)$$

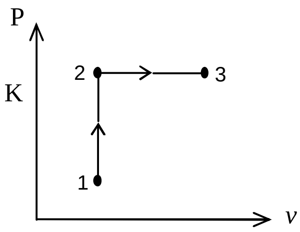
$$Q_{in} = m(u_3 - u_1) + W_{b,out}$$

The initial and the final volumes and the final temperature of air are

$$V_1 = \frac{mRT_1}{P_1} = \frac{(3\text{ kg})(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{ K})}{200\text{ kPa}} = 1.29\text{ m}^3$$

$$V_3 = 2V_1 = 2\times 1.29 = 2.58\text{ m}^3$$

$$\frac{P_1V_1}{T_1} = \frac{P_3V_3}{T_3} \longrightarrow T_3 = \frac{P_3}{P_1}\frac{V_3}{V_1}T_1 = \frac{400\text{ kPa}}{200\text{ kPa}}\times 2\times(300\text{ K}) = 1200\text{ K}$$



No work is done during process 1-2 since $V_1 = V_2$. The pressure remains constant during process 2-3 and the work done during this process is

$$W_{b,out} = \int_1^2 PdV = P_2(V_3 - V_2) = (400\text{ kPa})(2.58 - 1.29)\text{m}^3 = \mathbf{516\ \ kJ}$$

The initial and final internal energies of air are (Table A-17)

$$u_1 = u_{@\,300\text{ K}} = 214.07\text{ kJ/kg}$$

$$u_2 = u_{@1200\text{ K}} = 933.33\text{ kJ/kg}$$

Then from the energy balance,

Q_{in} = (3 kg)(933.33 - 214.07)kJ/kg + 516 kJ = **2674 kJ**

Alternative solution The specific heat of air at the average temperature of T_{ave} = (300 + 1200)/2 = 750 K is, from Table A-2b, C_{vave} = 0.800 kJ/kg.K. Substituting,

$$Q_{in} = m(u_3 - u_1) + W_{b,out} \cong mC_v(T_3 - T_1) + W_{b,out}$$

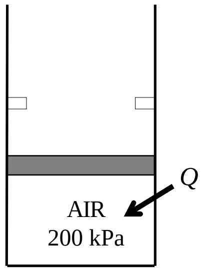
Q_{in} = (3 kg)(0.800 kJ/kg.K)(1200 - 300) K + 516 kJ = **2676 kJ**

5-44 [*Also solved by EES on enclosed CD***]** A cylinder equipped with a set of stops on the top is initially filled with air at a specified state. Heat is transferred to the air until the piston hits the stops, and then the pressure doubles. The work done by the air and the amount of heat transfer are to be determined, and the process is to be shown on a *P-v* diagram.

Assumptions **1** Air is an ideal gas with variable specific heats. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** There are no work interactions involved. **3** The thermal energy stored in the cylinder itself is negligible.

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

Analysis We take the air in the cylinder to be the system. This is a closed system since no mass crosses the boundary of the system. The energy balance for this closed system can be expressed as



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

$$Q_{in} - W_{b,out} = \Delta U = m(u_3 - u_1)$$

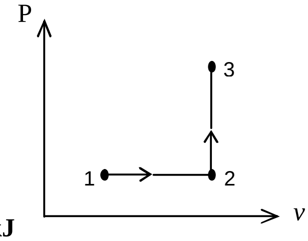
$$Q_{in} = m(u_3 - u_1) + W_{b,out}$$

The initial and the final volumes and the final temperature of air are determined from

$$V_1 = \frac{mRT_1}{P_1} = \frac{(3\text{ kg})(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{ K})}{200\text{ kPa}} = 1.29\text{ m}^3$$

$$V_3 = 2V_1 = 2 \times 1.29 = 2.58\text{ m}^3$$

$$\frac{P_1V_1}{T_1} = \frac{P_3V_3}{T_3} \longrightarrow T_3 = \frac{P_3}{P_1}\frac{V_3}{V_1}T_1 = \frac{400\text{ kPa}}{200\text{ kPa}} \times 2 \times (300\text{ K}) = 1200\text{ K}$$



No work is done during process 2-3 since $V_2 = V_3$. The pressure remains constant during process 1-2 and the work done during this process is

$$W_b = \int_1^2 P\,dV = P_2(V_3 - V_2) = (200\text{kPa})(2.58 - 1.29)\text{m}^3\left(\frac{1\text{kJ}}{1\text{kPa}\cdot\text{m}^3}\right) = \mathbf{258 kJ}$$

The initial and final internal energies of air are (Table A-17)

$$u_1 = u_{@\,300\text{ K}} = 214.07\text{ kJ/kg}$$

$$u_2 = u_{@\,1200\text{ K}} = 933.33\text{ kJ/kg}$$

Substituting,

$$Q_{in} = (3\text{ kg})(933.33 - 214.07)\text{kJ/kg} + 258\text{ kJ} = \mathbf{2416\text{ kJ}}$$

Alternative solution The specific heat of air at the average temperature of T_{ave} = (300 + 1200)/2 = 750 K is, from Table A-2b, C_{vave} = 0.800 kJ/kg.K. Substituting

$$Q_{in} = m(u_3 - u_1) + W_{b,out} \cong mC_v(T_3 - T_1) + W_{b,out}$$

$$= (3\text{ kg})(0.800\text{ kJ/kg.K})(1200 - 300)\text{ K} + 258\text{ kJ} = \mathbf{2418\text{ kJ}}$$

Closed System Energy Analysis: Solids and Liquids

5-45 A number of brass balls are to be quenched in a water bath at a specified rate. The rate at which heat needs to be removed from the water in order to keep its temperature constant is to be determined.

Assumptions **1** The thermal properties of the balls are constant. **2** The balls are at a uniform temperature before and after quenching. **3** The changes in kinetic and potential energies are negligible.

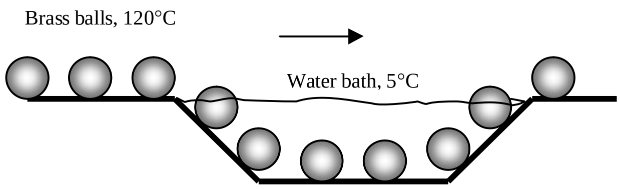
Properties The density and specific heat of the brass balls are given to be ρ = 8522 kg/m^3 and C_p = 0.385 kJ/kg.°C.

Analysis We take a single ball as the system. The energy balance for this closed system can be expressed as

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

$$-Q_{out} = \Delta U_{ball} = m(u_2 - u_1)$$

$$Q_{out} = mC(T_1 - T_2)$$



The total amount of heat transfer from a ball is

$$m = \rho V = \rho \frac{\pi D^3}{6} = (8522 \text{ kg/m}^3)\frac{\pi (0.05 \text{ m})^3}{6} = 0.558 \text{ kg}$$

$$Q_{out} = mC(T_1 - T_2) = (0.558 \text{ kg})(0.385 \text{ kJ/kg.°C})(120 - 74)\text{°C} = 9.88 \text{ kJ/ball}$$

Then the rate of heat transfer from the balls to the water becomes

$$\dot{Q}_{total} = \dot{n}_{\text{ball}} Q_{\text{ball}} = (100 \text{ balls/min}) \times (9.88 \text{ kJ/ball}) = \mathbf{988 \text{ kJ/min}}$$

Therefore, heat must be removed from the water at a rate of 988 kJ/min in order to keep its temperature constant at 50°C since energy input must be equal to energy output for a system whose energy level remains constant. That is, $E_{in} = E_{out}$ when $\Delta E_{\text{system}} = 0$.

5-46 A number of aluminum balls are to be quenched in a water bath at a specified rate. The rate at which heat needs to be removed from the water in order to keep its temperature constant is to be determined.

Assumptions **1** The thermal properties of the balls are constant. **2** The balls are at a uniform temperature before and after quenching. **3** The changes in kinetic and potential energies are negligible.

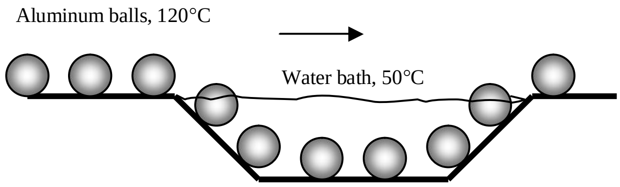
Properties The density and specific heat of aluminum at the average temperature of (120+74)/2 = 97°C = 370 K are ρ = 2700 kg/m^3 and C_p = 0.937 kJ/kg.°C (Table A-3).

Analysis We take a single ball as the system. The energy balance for this closed system can be expressed as

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

$$-Q_{out} = \Delta U_{ball} = m(u_2 - u_1)$$

$$Q_{out} = mC(T_1 - T_2)$$



The total amount of heat transfer from a ball is

$$m = \rho V = \rho \frac{\pi D^3}{6} = (2700 \text{ kg/m}^3)\frac{\pi(0.05 \text{ m})^3}{6} = 0.1767 \text{ kg}$$

$$Q_{out} = mC(T_1 - T_2) = (0.1767 \text{ kg})(0.937 \text{ kJ/kg.°C})(120 - 74)\text{°C} = 7.62 \text{ kJ/ball}$$

Then the rate of heat transfer from the balls to the water becomes

$$\dot{Q}_{total} = \dot{n}_{\text{ball}} Q_{\text{ball}} = (100 \text{ balls/min}) \times (7.62 \text{ kJ/ball}) = \mathbf{762 \text{ kJ/min}}$$

Therefore, heat must be removed from the water at a rate of 762 kJ/min in order to keep its temperature constant at 50°C since energy input must be equal to energy output for a system whose energy level remains constant. That is, $E_{in} = E_{out}$ when $\Delta E_{\text{system}} = 0$.

5-47E A person shakes a canned of drink in a iced water to cool it. The mass of the ice that will melt by the time the canned drink is cooled to a specified temperature is to be determined.

Assumptions **1** The thermal properties of the drink are constant, and are taken to be the same as those of water. **2** The effect of agitation on the amount of ice melting is negligible. **3** The thermal energy capacity of the can itself is negligible, and thus it does not need to be considered in the analysis.

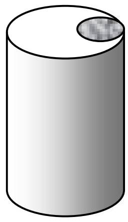
Properties The density and specific heat of water at the average temperature of (75+45)/2 = 60°F are ρ = 62.3 lbm/ft^3, and C_p = 1.0 Btu/lbm.°F (Table A-3E). The heat of fusion of water is 143.5 Btu/lbm.

Analysis We take a canned drink as the system. The energy balance for this closed system can be expressed as

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

$$-Q_{out} = \Delta U_{\text{canned drink}} = m(u_2 - u_1)$$

$$Q_{out} = mC(T_1 - T_2)$$



Cola
75°F

Noting that 1 gal = 128 oz and 1 ft^3 = 7.48 gal = 957.5 oz, the total amount of heat transfer from a ball is

$$m = \rho V = (62.3\,\text{lbm/ft}^3)(12\text{ oz/can})\left(\frac{1\,\text{ft}^3}{7.48\text{ gal}}\right)\left(\frac{1\,\text{gal}}{128\,\text{fluid oz}}\right) = 0.781\,\text{lbm/can}$$

$$Q_{out} = mC(T_1 - T_2) = (0.781\,\text{lbm/can})(1.0\text{ Btu/lbm.°F})(75-45)\text{°F} = 23.4\text{ Btu/can}$$

Noting that the heat of fusion of water is 14.5 Btu/lbm, the amount of ice that will melt to cool the drink is

$$m_{ice} = \frac{Q_{out}}{h_{if}} = \frac{23.4\text{ Btu/can}}{143.5\text{ Btu/lbm}} = \mathbf{0.163\ lbm} \quad \text{(per can of drink)}$$

since heat transfer to the ice must be equal to heat transfer from the can.

Discussion The actual amount of ice melted will be greater since agitation will also cause some ice to melt.

5-48 An iron whose base plate is made of an aluminum alloy is turned on. The minimum time for the plate to reach a specified temperature is to be determined.

Assumptions **1** It is given that 85 percent of the heat generated in the resistance wires is transferred to the plate. **2** The thermal properties of the plate are constant. **3** Heat loss from the plate during heating is disregarded since the minimum heating time is to be determined. **4** There are no changes in kinetic and potential energies. **5** The plate is at a uniform temperature at the end of the process.

Properties The density and specific heat of the aluminum alloy plate are given to be ρ = 2770 kg/m^3 and C_p = 875 kJ/kg.°C.

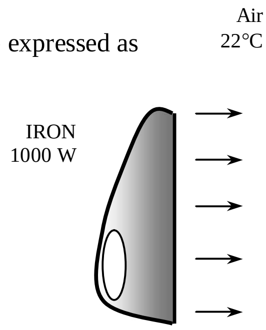
Analysis The mass of the iron's base plate is

$$m = \rho V = \rho LA = (2770\ \text{kg/m}^3)(0.005\ \text{m})(0.03\ \text{m}^2) = 0.4155\ \text{kg}$$

Noting that only 85 percent of the heat generated is transferred to the plate, the rate of heat transfer to the iron's base plate is

$$\dot{Q}_{\text{in}} = 0.85 \times 1000\ \text{W} = 850\ \text{W}$$

We take plate to be the system. The energy balance for this closed system can be expressed as



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

$$Q_{in} = \Delta U_{\text{plate}} = m(u_2 - u_1)$$

$$\dot{Q}_{in}\Delta t = mC(T_2 - T_1)$$

Solving for Δ*t* and substituting,

$$\Delta t = \frac{mC\Delta T_{\text{plate}}}{\dot{Q}_{\text{in}}} = \frac{(0.4155\ \text{kg})(875\ \text{J/kg.°C})(140 - 22)\text{°C}}{850\ \text{J/s}} = \mathbf{50.5\ s}$$

which is the time required for the plate temperature to reach the specified temperature.

5-49 Stainless steel ball bearings leaving the oven at a specified uniform temperature at a specified rate are exposed to air and are cooled before they are dropped into the water for quenching. The rate of heat transfer from the ball bearing to the air is to be determined. √

Assumptions **1** The thermal properties of the bearing balls are constant. **2** The kinetic and potential energy changes of the balls are negligible. **3** The balls are at a uniform temperature at the end of the process

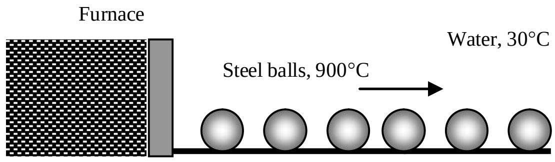
Properties The density and specific heat of the ball bearings are given to be ρ = 8085 kg/m^3 and C_p = 0.480 kJ/kg.°C.

Analysis We take a single bearing ball as the system. The energy balance for this closed system can be expressed as

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

$$-Q_{out} = \Delta U_{ball} = m(u_2 - u_1)$$

$$Q_{out} = mC(T_1 - T_2)$$



The total amount of heat transfer from a ball is

$$m = \rho V = \rho \frac{\pi D^3}{6} = (8085 \text{ kg/m}^3)\frac{\pi (0.012 \text{ m})^3}{6} = 0.007315 \text{ kg}$$

$$Q_{out} = mC(T_1 - T_2) = (0.007315 \text{ kg})(0.480 \text{ kJ/kg.°C})(900 - 850)\text{°C} = 0.1756 \text{ kJ/ball}$$

Then the rate of heat transfer from the balls to the air becomes

$$\dot{Q}_{total} = \dot{n}_{\text{ball}} Q_{\text{out (per ball)}} = (1400 \text{ balls/min}) \times (0.1756 \text{ kJ/ball}) = \mathbf{245.8\ kJ/min = 4.10\ kW}$$

Therefore, heat is lost to the air at a rate of 4.10 kW.

5-50 Carbon steel balls are to be annealed at a rate of 2500/h by heating them first and then allowing them to cool slowly in ambient air at a specified rate. The total rate of heat transfer from the balls to the ambient air is to be determined.

Assumptions **1** The thermal properties of the balls are constant. **2** There are no changes in kinetic and potential energies. **3** The balls are at a uniform temperature at the end of the process

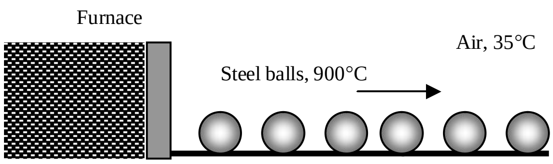
Properties The density and specific heat of the balls are given to be ρ = 7833 kg/m^3 and C_p = 0.465 kJ/kg.°C.

Analysis We take a single ball as the system. The energy balance for this closed system can be expressed as

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

$$-Q_{out} = \Delta U_{ball} = m(u_2 - u_1)$$

$$Q_{out} = mC(T_1 - T_2)$$



(*b*) The amount of heat transfer from a single ball is

$$m = \rho V = \rho \frac{\pi D^3}{6} = (7833\ \text{kg/m}^3)\frac{\pi (0.008\ \text{m})^3}{6} = 0.00210\ \text{kg}$$

$$Q_{out} = mC_p(T_1 - T_2) = (0.0021\ \text{kg})(0.465\ \text{kJ/kg.°C})(900 - 100)\text{°C} = 0.781\ \text{kJ (per ball)}$$

Then the total rate of heat transfer from the balls to the ambient air becomes

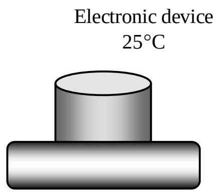
$$\dot{Q}_{out} = \dot{n}_{\text{ball}} Q_{out} = (2500\ \text{balls/h}) \times (0.781\ \text{kJ/ball}) = 1{,}953\ \text{kJ/h} = \mathbf{542\ W}$$

5-51 An electronic device is on for 5 minutes, and off for several hours. The temperature of the device at the end of the 5-min operating period is to be determined for the cases of operation with and without a heat sink.

Assumptions **1** The device and the heat sink are isothermal. **2** The thermal properties of the device and of the sink are constant. **3** Heat loss from the device during on time is disregarded since the highest possible temperature is to be determined.

Properties The specific heat of the device is given to be C_p = 850 J/kg.°C. The specific heat of aluminum at room temperature of 300 K is 902 J/kg.°C (Table A-3).

Analysis We take the device to be the system. Noting that electrical energy is supplied, the energy balance for this closed system can be expressed as



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

$$W_{e,in} = \Delta U_{\text{device}} = m(u_2 - u_1)$$

$$\dot{W}_{e,in}\Delta t = mC(T_2 - T_1)$$

Substituting, the temperature of the device at the end of the process is determined to be

$$(30 \text{ J/s})(5\times 60 \text{ s}) = (0.020 \text{ kg})(850 \text{ J/kg.°C})(T_2 - 25)\text{°C} \quad \rightarrow \quad T_2 = \mathbf{554°C} \text{ (without the heat sink)}$$

Case 2 When a heat sink is attached, the energy balance can be expressed as

$$W_{e,in} = \Delta U_{\text{device}} + \Delta U_{\text{heat sink}}$$

$$\dot{W}_{e,in}\Delta t = mC(T_2 - T_1)_{\text{device}} + mC(T_2 - T_1)_{\text{heat sink}}$$

Substituting, the temperature of the device-heat sink combination is determined to be

$$(30 \text{ J/s})(5\times 60 \text{ s}) = (0.020 \text{ kg})(850 \text{ J/kg.°C})(T_2 - 25)\text{°C} + (0.200 \text{ kg})(902 \text{ J/kg.°C})(T_2 - 25)\text{°C}$$

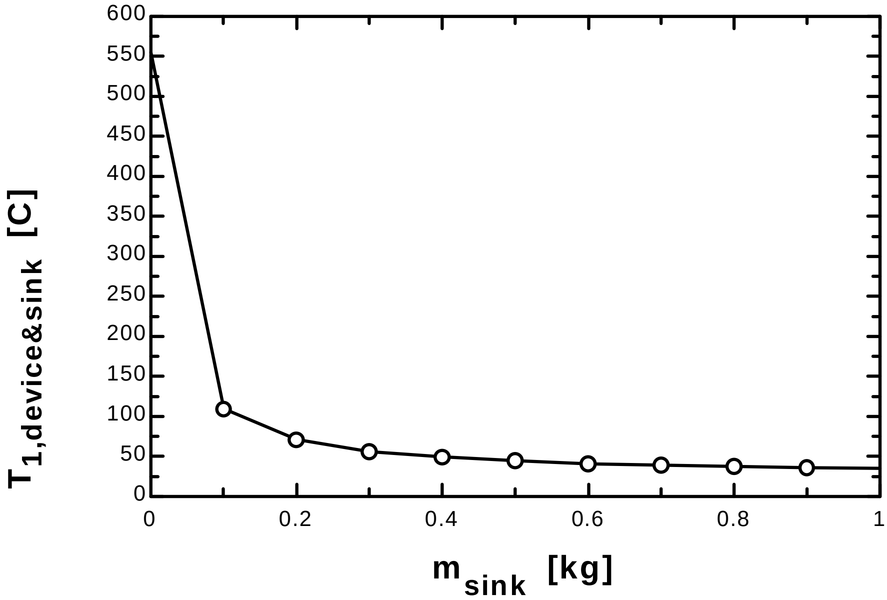
$$T_2 = \mathbf{70.6°C} \quad \text{(with heat sink)}$$

Discussion These are the maximum temperatures. In reality, the temperatures will be lower because of the heat losses to the surroundings.

5-52 Problem 5-52 is reconsidered. The effect of the mass of the heat sink on the maximum device temperature as the mass of heat sink varies from 0 kg to 1 kg is to be investigated. The maximum temperature is to be plotted against the mass of heat sink.

```
"Knowns:"
"T_1 is the maximum temperature of the device"
Q_dot_out = 30"[W]"
m_device=20"[g]"
Cp_device=850"[J/kg-C]"
A=5"[cm^2]"
DELTAt=5"[min]"
T_amb=25"[C]"
{m_sink=0.2"[kg]"}
"Cp_al taken from Table A-3(b) at 300K"
Cp_al=0.902"[kJ/kg-C]"
T_2=T_amb"[C]"

"Solution:"
"The device without the heat sink is considered to be a closed system."
"Conservation of Energy for the closed system:"
"E_dot_in - E_dot_out = DELTAE_dot,  we neglect DELTA KE and DELTA PE for the
system, the device."
E_dot_in - E_dot_out = DELTAE_dot
E_dot_in =0"[W]"
E_dot_out = Q_dot_out"[W]"

"Use the solid material approximation to find the energy change of the device."
DELTAE_dot= m_device*convert(g,kg)*Cp_device*(T_2-
T_1_device)/(DELTAt*convert(min,s))"[W]"

"The device with the heat sink is considered to be a closed system."
"Conservation of Energy for the closed system:"
"E_dot_in - E_dot_out = DELTAE_dot,  we neglect DELTA KE and DELTA PE for the device
with the heat sink."
E_dot_in - E_dot_out = DELTAE_dot_combined

"Use the solid material approximation to find the energy change of the device."
DELTAE_dot_combined= (m_device*convert(g,kg)*Cp_device*(T_2-
T_1_device&sink)+m_sink*Cp_al*(T_2-
T_1_device&sink)*convert(kJ,J))/(DELTAt*convert(min,s))"[W]"
```

m_{sink} [kg]	$T_{1,device\&sink}$ [C]
0	554.4
0.1	109
0.2	70.59
0.3	56.29
0.4	48.82
0.5	44.23
0.6	41.12
0.7	38.88
0.8	37.19
0.9	35.86
1	34.79


600
550
500
450
400
350
300
250
200
150
100
50
0
0
0.2
0.4
0.6
0.8
1
T1,device&sink [C]
msink [kg]

5-53 An egg is dropped into boiling water. The amount of heat transfer to the egg by the time it is cooked is to be determined.

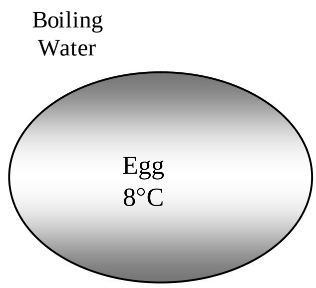
Assumptions **1** The egg is spherical in shape with a radius of r_0 = 2.75 cm. **2** The thermal properties of the egg are constant. **3** Energy absorption or release associated with any chemical and/or phase changes within the egg is negligible. **4** There are no changes in kinetic and potential energies.

Properties The density and specific heat of the egg are given to be ρ = 1020 kg/m^3 and C_p = 3.32 kJ/kg.°C.

Analysis We take the egg as the system. This is a closes system since no mass enters or leaves the egg. The energy balance for this closed system can be expressed as

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

$$Q_{in} = \Delta U_{egg} = m(u_2 - u_1) = mC(T_2 - T_1)$$



Then the mass of the egg and the amount of heat transfer become

$$m = \rho V = \rho \frac{\pi D^3}{6} = (1020\,\text{kg/m}^3)\frac{\pi(0.055\,\text{m})^3}{6} = 0.0889\,\text{kg}$$

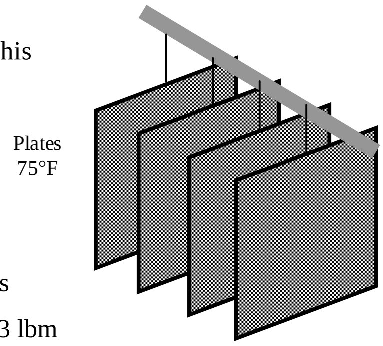
$$Q_{in} = mC_p(T_2 - T_1) = (0.0889\,\text{kg})(3.32\,\text{kJ/kg.°C})(70-8)\text{°C} = \mathbf{18.3\,kJ}$$

5-54E Large brass plates are heated in an oven at a rate of 300/min. The rate of heat transfer to the plates in the oven is to be determined.

Assumptions **1** The thermal properties of the plates are constant. **2** The changes in kinetic and potential energies are negligible.

Properties The density and specific heat of the brass are given to be ρ = 532.5 lbm/ft^3 and C_p = 0.091 Btu/lbm.°F.

Analysis We take the plate to be the system. The energy balance for this closed system can be expressed as



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

$$Q_{in} = \Delta U_{\text{plate}} = m(u_2 - u_1) = mC(T_2 - T_1)$$

The mass of each plate and the amount of heat transfer to each plate is

$$m = \rho V = \rho LA = (532.5 \text{ lbm/ft}^3)[(1.2/12 \text{ ft})(2 \text{ ft})(2 \text{ ft})] = 213 \text{ lbm}$$

$$Q_{in} = mC(T_2 - T_1) = (213 \text{ lbm/plate})(0.091 \text{ Btu/lbm.°F})(1000 - 75)\text{°F} = 17{,}930 \text{ Btu/plate}$$

Then the total rate of heat transfer to the plates becomes

$$\dot{Q}_{total} = \dot{n}_{\text{plate}} Q_{\text{in, per plate}} = (300 \text{ plates/min}) \times (17{,}930 \text{ Btu/plate}) = \mathbf{5{,}379{,}000 \text{ Btu/min} = 89{,}650 \text{ Btu/s}}$$

5-55 Long cylindrical steel rods are heat-treated in an oven. The rate of heat transfer to the rods in the oven is to be determined.

Assumptions **1** The thermal properties of the rods are constant. **2** The changes in kinetic and potential energies are negligible.

Properties The density and specific heat of the steel rods are given to be ρ = 7833 kg/m^3 and C_p = 0.465 kJ/kg.°C.

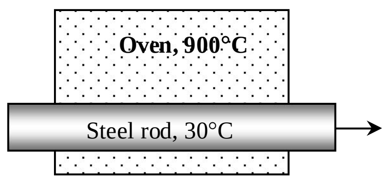
Analysis Noting that the rods enter the oven at a velocity of 3 m/min and exit at the same velocity, we can say that a 3-m long section of the rod is heated in the oven in 1 min. Then the mass of the rod heated in 1 minute is

$$m = \rho V = \rho LA = \rho L(\pi D^2/4) = (7833 \text{ kg/m}^3)(3 \text{ m})[\pi(0.1 \text{ m})^2/4] = 184.6 \text{ kg}$$

We take the 3-m section of the rod in the oven as the system. The energy balance for this closed system can be expressed as

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

$$Q_{in} = \Delta U_{\text{rod}} = m(u_2 - u_1) = mC(T_2 - T_1)$$



Substituting,

$$Q_{in} = mC(T_2 - T_1) = (184.6 \text{ kg})(0.465 \text{ kJ/kg.°C})(700 - 30)\text{°C} = 57{,}512 \text{ kJ}$$

Noting that this much heat is transferred in 1 min, the rate of heat transfer to the rod becomes

$$\dot{Q}_{in} = Q_{\text{in}}/\Delta t = (57{,}512 \text{ kJ})/(1 \text{ min}) = 57{,}512 \text{ kJ/min} = \mathbf{958.5 \text{ kW}}$$

Steady Flow Energy Balance: Nozzles and Diffusers

5-56C A steady-flow system involves no changes with time anywhere within the system or at the system boundaries

---

5-57C No.

5-58C It is mostly converted to internal energy as shown by a rise in the fluid temperature.

5-59C The kinetic energy of a fluid increases at the expense of the internal energy as evidenced by a decrease in the fluid temperature.

5-60C Heat transfer to the fluid as it flows through a nozzle is desirable since it will probably increase the kinetic energy of the fluid. Heat transfer from the fluid will decrease the exit velocity.

5-61 Air is accelerated in a nozzle from 30 m/s to 180 m/s. The mass flow rate, the exit temperature, and the exit area of the nozzle are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with constant specific heats. **3** Potential energy changes are negligible. **4** The device is adiabatic and thus heat transfer is negligible. **5** There are no work interactions.

Properties The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1). The specific heat of air at the anticipated average temperature of 450 K is C_p = 1.02 kJ/kg.°C (Table A-2).

Analysis (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Using the ideal gas relation, the specific volume and the mass flow rate of air are determined to be

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(473\ \text{K})}{300\ \text{kPa}} = 0.4525\ \text{m}^3/\text{kg}$$

$$\dot{m} = \frac{1}{v_1}A_1\mathbf{V}_1 = \frac{1}{0.4525\text{m}^3/\text{kg}}(0.008\text{m}^2)(30\text{m/s}) = \mathbf{0.5304\ kg/s}$$

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

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

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

$$\dot{m}(h_1 + \mathbf{V}_1^2/2) = \dot{m}(h_2 + \mathbf{V}_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta\text{pe} \cong 0)$$

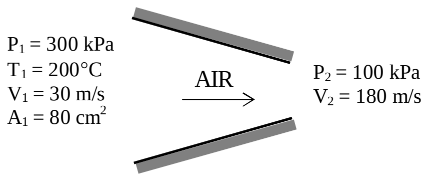
$$0 = h_2 - h_1 + \frac{\mathbf{V}_2^2 - \mathbf{V}_1^2}{2} \longrightarrow 0 = C_{p,ave}\left(T_2 - T_1\right) + \frac{\mathbf{V}_2^2 - \mathbf{V}_1^2}{2}$$

Substituting,

$$0 = (1.02\ \text{kJ/kg}\cdot\text{K})(T_2 - 200^\circ\text{C}) + \frac{(180\ \text{m/s})^2 - (30\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)$$

It yields $\qquad T_2$ = **184.6°C**

(*c*) The specific volume of air at the nozzle exit is



$$v_2 = \frac{RT_2}{P_2} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(184.6 + 273\ \text{K})}{100\ \text{kPa}} = 1.313\ \text{m}^3/\text{kg}$$

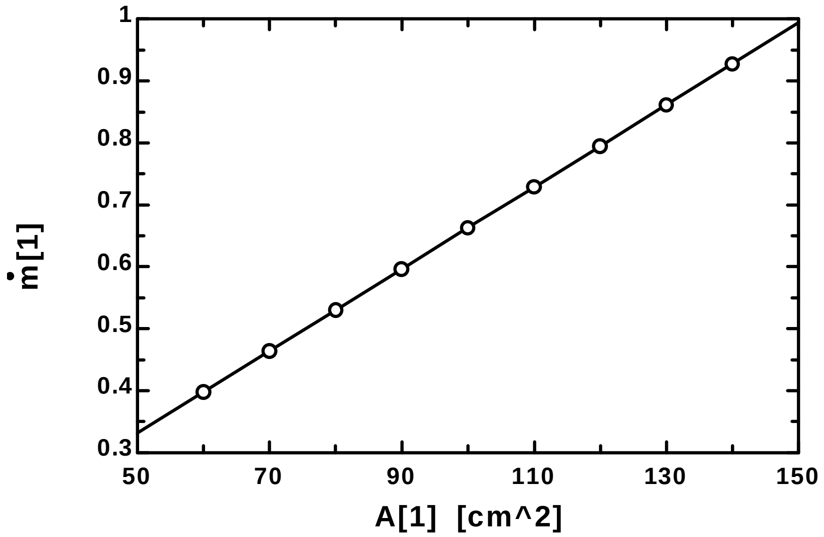
$$\dot{m} = \frac{1}{v_2}A_2\mathbf{V}_2 \longrightarrow 0.5304\text{kg/s} = \frac{1}{1.313\text{m}^3/\text{kg}}A_2\left(180\text{m/s}\right)$$

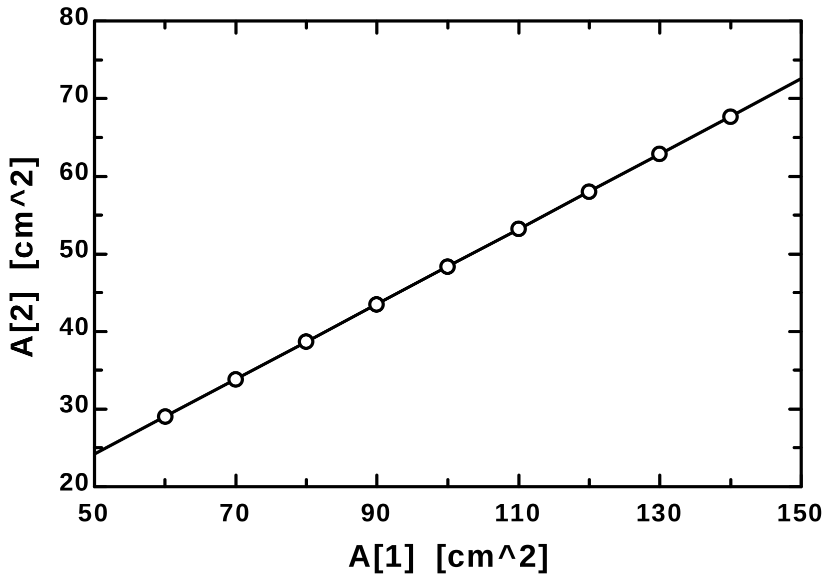
$$A_2 = 0.00387\ \text{m}^2 = \mathbf{38.7\ cm^2}$$

5-62 Problem 5-61 is reconsidered. The effect of the inlet area on the mass flow rate, exit velocity, and the exit area as the inlet area varies from 50 cm^2 to 150 cm^2 is to be investigated, and the final results are to be plotted against the inlet area.

```
Function HCal(WorkFluid$, Tx, Px)
"Function to calculate the enthalpy of an ideal gas or real gas"
   If  'Air' = WorkFluid$  then
       HCal:=ENTHALPY('Air',T=Tx) "Ideal gas equ."
   else
      HCal:=ENTHALPY(WorkFluid$,T=Tx, P=Px)"Real gas equ."
   endif
end  HCal

"System: control volume for the nozzle, Property relation: Air is an ideal gas"
"Process: Steady state, steady flow, adiabatic, no work"
"Knowns - obtain from the input diagram"
WorkFluid$ = 'Air'
T[1] = 200  "[C]"
P[1] = 300   "[kPa]"
Vel[1] = 30  "[m/s]"
P[2] = 100  "[kPa]"
Vel[2] = 180  "[m/s]"
A[1]=80 "[cm^2]"
Am[1]=A[1]*convert(cm^2,m^2)"[m^2]"

"Property Data - since the Enthalpy function has different parameters
for ideal gas and real fluids, a function was used to determine h."
h[1]=HCal(WorkFluid$,T[1],P[1])
h[2]=HCal(WorkFluid$,T[2],P[2])

"The Volume function has the same form for an ideal gas as for a real fluid."
v[1]=volume(workFluid$,T=T[1],p=P[1])
v[2]=volume(WorkFluid$,T=T[2],p=P[2])

"Conservation of mass: "
m_dot[1]= m_dot[2]
"Mass flow rate"
m_dot[1]=Am[1]*Vel[1]/v[1]
m_dot[2]= Am[2]*Vel[2]/v[2]

"Conservation of Energy - SSSF energy balance"
h[1]+Vel[1]^2/(2*1000) = h[2]+Vel[2]^2/(2*1000)
"Definition"
A_ratio=A[1]/A[2]
A[2]=Am[2]*convert(m^2,cm^2)
```

A_1 [cm^2]	A_2 [cm^2]	m_1	T_2
50	24.19	0.3314	184.6
60	29.02	0.3976	184.6
70	33.86	0.4639	184.6
80	38.7	0.5302	184.6
90	43.53	0.5964	184.6
100	48.37	0.6627	184.6
110	53.21	0.729	184.6
120	58.04	0.7952	184.6
130	62.88	0.8615	184.6
140	67.72	0.9278	184.6
150	72.56	0.9941	184.6


1
0.9
0.8
0.7
0.6
0.5
0.4
0.3
ṁ[1]
50
70
90
110
130
150
A[1] [cm^2]

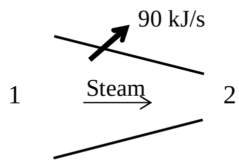

80
70
60
50
40
30
20
A[2] [cm^2]
50
70
90
110
130
150
A[1] [cm^2]

5-63 Steam is accelerated in a nozzle from a velocity of 80 m/s. The mass flow rate, the exit velocity, and the exit area of the nozzle 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** There are no work interactions.

Properties From the steam tables (Table A-6)

$$\left.\begin{array}{l} P_1 = 5 \text{ MPa} \\ T_1 = 500^\circ\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.06857 \text{ m}^3\text{/kg} \\ h_1 = 3433.8 \text{ kJ/kg} \end{array}$$



and

$$\left.\begin{array}{l} P_2 = 2\text{MPa} \\ T_2 = 400^\circ\text{C} \end{array}\right\} \begin{array}{l} v_2 = 0.15120\text{m}^3\text{/kg} \\ h_2 = 3247.6\text{kJ/kg} \end{array}$$

Analysis (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$.

The mass flow rate of steam is

$$\dot{m} = \frac{1}{v_1}\mathbf{V}_1 A_1 = \frac{1}{0.06857\text{m}^3\text{/kg}}(80 \text{ m/s})(50\times10^{-4}\text{ m}^2) = \mathbf{5.833\ kg/s}$$

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

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

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

$$\dot{m}(h_1 + \mathbf{V}_1^2/2) = \dot{Q}_{\text{out}} + \dot{m}(h_2 + \mathbf{V}_2^2/2) \quad (\text{since } \dot{W} \cong \Delta\text{pe} \cong 0)$$

$$-\dot{Q}_{\text{out}} = \dot{m}\left(h_2 - h_1 + \frac{\mathbf{V}_2^2 - \mathbf{V}_1^2}{2}\right)$$

Substituting, the exit velocity of the steam is determined to be

$$-90\text{kJ/s} = (5.833\text{kg/s})\left(3247.6 - 3433.8 + \frac{\mathbf{V}_2^2 - (80\text{m/s})^2}{2}\left(\frac{1\text{kJ/kg}}{1000\text{m}^2/\text{s}^2}\right)\right)$$

It yields $\mathbf{V_2 = 589.9\ m/s}$

(*c*) The exit area of the nozzle is determined from

$$\dot{m} = \frac{1}{v_2}\mathbf{V}_2 A_2 \longrightarrow A_2 = \frac{\dot{m}v_2}{\mathbf{V}_2} = \frac{(5.833 \text{ kg/s})(0.1512 \text{ m}^3\text{/kg})}{589.9 \text{ m/s}} = \mathbf{15.0\times10^{-4}\ m^2}$$

5-64E Air is accelerated in a nozzle from 150 ft/s to 900 ft/s. The exit temperature of air and the exit area of the nozzle are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with variable specific heats. **3** Potential energy changes are negligible. **4** There are no work interactions.

Properties The enthalpy of air at the inlet is h_1 = 143.47 Btu/lbm (Table A-17E).

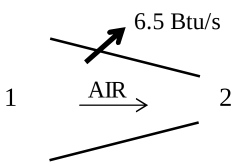
Analysis (a) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

$$\dot{m}(h_1 + \mathbf{V}_1^2/2) = \dot{Q}_{\text{out}} + \dot{m}(h_2 + \mathbf{V}_2^2/2) \quad (\text{since } \dot{W} \cong \Delta\text{pe} \cong 0)$$

$$-\dot{Q}_{\text{out}} = \dot{m}\left(h_2 - h_1 + \frac{\mathbf{V}_2^2 - \mathbf{V}_1^2}{2}\right)$$



or,

$$h_2 = -q_{\text{out}} + h_1 - \frac{\mathbf{V}_2^2 - \mathbf{V}_1^2}{2}$$

$$= -6.5 \text{ Btu/lbm} + 143.47 \text{ Btu/lbm} - \frac{(900 \text{ ft/s})^2 - (150 \text{ ft/s})^2}{2}\left(\frac{1 \text{ Btu/lbm}}{25{,}037 \text{ ft}^2/\text{s}^2}\right)$$

$$= 121.2 \text{ Btu/lbm}$$

Thus, from Table A-17E, T_2 = **507 R**

(*b*) The exit area is determined from the conservation of mass relation,

$$\frac{1}{v_2}A_2\mathbf{V}_2 = \frac{1}{v_1}A_1\mathbf{V}_1 \longrightarrow A_2 = \frac{v_2}{v_1}\frac{\mathbf{V}_1}{\mathbf{V}_2}A_1 = \left(\frac{RT_2/P_2}{RT_1/P_1}\right)\frac{\mathbf{V}_1}{\mathbf{V}_2}A_1$$

$$A_2 = \frac{(508/14.7)(150 \text{ft/s})}{(600/50)(900 \text{ft/s})}(0.1 \text{ft}^2) = \mathbf{0.048 \text{ ft}^2}$$

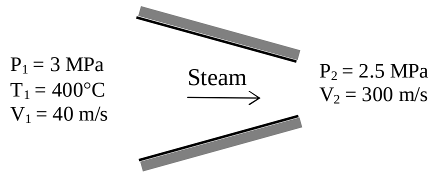
5-65 [*Also solved by EES on enclosed CD***]** Steam is accelerated in a nozzle from a velocity of 40 m/s to 300 m/s. The exit temperature and the ratio of the inlet-to-exit area of the nozzle are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

Properties From the steam tables (Table A-6),

$$\left.\begin{array}{l} P_1 = 3\text{MPa} \\ T_1 = 400^{\circ}\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.09936\text{m}^3/\text{kg} \\ h_1 = 3230.9\text{kJ/kg} \end{array}$$

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



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

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

$$\dot{m}(h_1 + \mathbf{V}_1^2/2) = \dot{m}(h_2 + \mathbf{V}_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta\text{pe} \cong 0)$$

$$0 = h_2 - h_1 + \frac{\mathbf{V}_2^2 - \mathbf{V}_1^2}{2}$$

or,

$$h_2 = h_1 - \frac{\mathbf{V}_2^2 - \mathbf{V}_1^2}{2} = 3230.9 \text{ kJ/kg} - \frac{(300 \text{ m/s})^2 - (40 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) = 3186.7 \text{ kJ/kg}$$

Thus,
$$\left.\begin{array}{l} P_2 = 2.5\text{MPa} \\ h_2 = 3186.7\text{kJ/kg} \end{array}\right\} \begin{array}{l} T_2 = \mathbf{376.7^{\circ}C} \\ v_2 = 0.1153\text{m}^3/\text{kg} \end{array}$$

(*b*) The ratio of the inlet to exit area is determined from the conservation of mass relation,

$$\frac{1}{v_2}A_2\mathbf{V}_2 = \frac{1}{v_1}A_1\mathbf{V}_1 \longrightarrow \frac{A_1}{A_2} = \frac{v_1}{v_2}\frac{\mathbf{V}_2}{\mathbf{V}_1} = \frac{(0.09936 \text{ m}^3/\text{kg})(300 \text{ m/s})}{(0.1153 \text{ m}^3/\text{kg})(40 \text{ m/s})} = \mathbf{6.46}$$

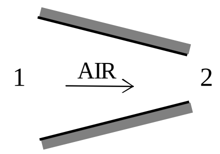
5-66 Air is accelerated in a nozzle from 120 m/s to 380 m/s. The exit temperature and pressure of air are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with variable specific heats. **3** Potential energy changes are negligible. **4** The device is adiabatic and thus heat transfer is negligible. **5** There are no work interactions.

Properties The enthalpy of air at the inlet temperature of 500 K is h_1 = 503.02 kJ/kg (Table A-17).

Analysis (a) There is only one inlet and one exit, and thus $\dot m_1 = \dot m_2 = \dot m$. We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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



$$\dot E_{in} = \dot E_{out}$$

$$\dot m(h_1 + \mathbf{V}_1^2/2) = \dot m(h_2 + \mathbf{V}_2^2/2) \quad (\text{since } \dot Q \cong \dot W \cong \Delta\text{pe} \cong 0)$$

$$0 = h_2 - h_1 + \frac{\mathbf{V}_2^2 - \mathbf{V}_1^2}{2}$$

or,

$$h_2 = h_1 - \frac{\mathbf{V}_2^2 - \mathbf{V}_1^2}{2} = 503.02\text{kJ/kg} - \frac{(380\text{m/s})^2 - (120\text{m/s})^2}{2}\left(\frac{1\text{kJ/kg}}{1000\text{m}^2/\text{s}^2}\right) = 438.02\text{kJ/kg}$$

Then from Table A-17 we read T_2 = **436.5 K**

(b) The exit pressure is determined from the conservation of mass relation,

$$\frac{1}{v_2}A_2\mathbf{V}_2 = \frac{1}{v_1}A_1\mathbf{V}_1 \longrightarrow \frac{1}{RT_2/P_2}A_2\mathbf{V}_2 = \frac{1}{RT_1/P_1}A_1\mathbf{V}_1$$

Thus,

$$P_2 = \frac{A_1 T_2 \mathbf{V}_1}{A_2 T_1 \mathbf{V}_2}P_1 = \frac{2}{1}\frac{(436.5\text{ K})(120\text{ m/s})}{(500\text{ K})(380\text{ m/s})}(600\text{ kPa}) = \mathbf{330.8\ kPa}$$

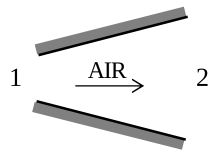
5-67 Air is decelerated in a diffuser from 230 m/s to 30 m/s. The exit temperature of air and the exit area of the diffuser are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with variable specific heats. **3** Potential energy changes are negligible. **4** The device is adiabatic and thus heat transfer is negligible. **5** There are no work interactions.

Properties The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1). The enthalpy of air at the inlet temperature of 400 K is h_1 = 400.98 kJ/kg (Table A-17).

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

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



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

$$\dot{m}(h_1 + \mathbf{V}_1^2/2) = \dot{m}(h_2 + \mathbf{V}_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta\text{pe} \cong 0)$$

$$0 = h_2 - h_1 + \frac{\mathbf{V}_2^2 - \mathbf{V}_1^2}{2} ,$$

or,

$$h_2 = h_1 - \frac{\mathbf{V}_2^2 - \mathbf{V}_1^2}{2} = 400.98 \text{kJ/kg} - \frac{(30 \text{m/s})^2 - (230 \text{m/s})^2}{2}\left(\frac{1 \text{kJ/kg}}{1000 \text{m}^2/\text{s}^2}\right) = 426.98 \text{kJ/kg}$$

From Table A-17, T_2 = **425.6 K**

(b) The specific volume of air at the diffuser exit is

$$v_2 = \frac{RT_2}{P_2} = \frac{(0.287 \text{kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(425.6 \text{K})}{(100 \text{kPa})} = 1.221 \text{m}^3/\text{kg}$$

From conservation of mass,

$$\dot{m} = \frac{1}{v_2} A_2 \mathbf{V}_2 \longrightarrow A_2 = \frac{\dot{m} v_2}{\mathbf{V}_2} = \frac{(6000/3600 \text{ kg/s})(1.221 \text{ m}^3/\text{kg})}{30 \text{ m/s}} = \mathbf{0.0678 \ m^2}$$

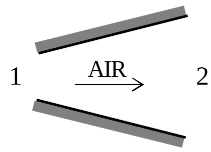
5-68E Air is decelerated in a diffuser from 600 ft/s to a low velocity. The exit temperature and the exit velocity of air are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with variable specific heats. **3** Potential energy changes are negligible. **4** The device is adiabatic and thus heat transfer is negligible. **5** There are no work interactions.

Properties The enthalpy of air at the inlet temperature of 20°F is h_1 = 114.69 Btu/lbm (Table A-17E).

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

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



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

$$\dot{m}(h_1 + \mathbf{V}_1^2/2) = \dot{m}(h_2 + \mathbf{V}_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta\text{pe} \cong 0)$$

$$0 = h_2 - h_1 + \frac{\mathbf{V}_2^2 - \mathbf{V}_1^2}{2},$$

or,

$$h_2 = h_1 - \frac{\mathbf{V}_2^2 - \mathbf{V}_1^2}{2} = 114.69\,\text{Btu/lbm} - \frac{0 - (600\,\text{ft/s})^2}{2}\left(\frac{1\,\text{Btu/lbm}}{25{,}037\,\text{ft}^2/\text{s}^2}\right) = 121.88\,\text{Btu/lbm}$$

From Table A-17E, T_2 = **510.0 R**

(b) The exit velocity of air is determined from the conservation of mass relation,

$$\frac{1}{v_2}A_2\mathbf{V}_2 = \frac{1}{v_1}A_1\mathbf{V}_1 \longrightarrow \frac{1}{RT_2/P_2}A_2\mathbf{V}_2 = \frac{1}{RT_1/P_1}A_1\mathbf{V}_1$$

Thus,

$$\mathbf{V}_2 = \frac{A_1 T_2 P_1}{A_2 T_1 P_2}\mathbf{V}_1 = \frac{1}{5}\frac{(510\text{ R})(13\text{ psia})}{(480\text{ R})(14.5\text{ psia})}(600\text{ ft/s}) = \mathbf{114.3\ ft/s}$$

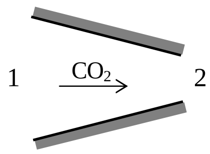
5-69 CO_2 gas is accelerated in a nozzle to 450 m/s. The inlet velocity and the exit temperature are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** CO_2 is an ideal gas with variable specific heats. **3** Potential energy changes are negligible. **4** The device is adiabatic and thus heat transfer is negligible. **5** There are no work interactions.

Properties The gas constant of CO_2 is 0.1889 kPa.m^3/kg.K (Table A-1). The enthalpy of CO_2 at 500°C is $\bar{h}_1 = 30{,}797$ kJ/kmol (Table A-20).

Analysis (a) There is only one inlet and one exit, and thus $\dot m_1 = \dot m_2 = \dot m$. Using the ideal gas relation, the specific volume is determined to be

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.1889\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(773\ \text{K})}{1000\ \text{kPa}} = 0.146\ \text{m}^3/\text{kg}$$



Thus,

$$\dot m = \frac{1}{v_1}A_1\mathbf{V}_1 \longrightarrow \mathbf{V}_1 = \frac{\dot m v_1}{A_1} = \frac{(6000/3600\ \text{kg/s})(0.146\ \text{m}^3/\text{kg})}{40\times10^{-4}\ \text{m}^2} = \mathbf{60.8\ m/s}$$

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

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

$$\dot E_{in} = \dot E_{out}$$

$$\dot m(h_1 + \mathbf{V}_1^2/2) = \dot m(h_2 + \mathbf{V}_2^2/2) \quad (\text{since } \dot Q \cong \dot W \cong \Delta\text{pe} \cong 0)$$

$$0 = h_2 - h_1 + \frac{\mathbf{V}_2^2 - \mathbf{V}_1^2}{2}$$

Substituting,

$$\bar h_2 = \bar h_1 - \frac{\mathbf{V}_2^2 - \mathbf{V}_1^2}{2} \times M$$

$$= 30{,}797\ \text{kJ/kmol} - \frac{(450\ \text{m/s})^2 - (60.8\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)(44\ \text{kg/kmol})$$

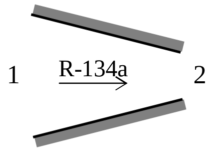
$$= 26{,}423\ \text{kJ/kmol}$$

Then the exit temperature of CO_2 from Table A-20 is obtained to be T_2 = **685.8 K**

5-70 R-134a is accelerated in a nozzle from a velocity of 20 m/s. The exit velocity of the refrigerant and the ratio of the inlet-to-exit area of the nozzle 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** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

Properties From the refrigerant tables (Table A-13)



$$\left.\begin{array}{l} P_1 = 700\text{kPa} \\ T_1 = 100^\circ\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.04064\text{m}^3/\text{kg} \\ h_1 = 338.19\text{kJ/kg} \end{array}$$

and

$$\left.\begin{array}{l} P_2 = 300\text{ kPa} \\ T_2 = 30^\circ\text{C} \end{array}\right\} \begin{array}{l} v_2 = 0.07767\text{ m}^3/\text{kg} \\ h_2 = 274.70\text{ kJ/kg} \end{array}$$

Analysis (*a*) There is only one inlet and one exit, and thus $\dot m_1 = \dot m_2 = \dot m$. We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

$$\dot E_{in} = \dot E_{out}$$

$$\dot m(h_1 + \mathbf{V}_1^2/2) = \dot m(h_2 + \mathbf{V}_2^2/2) \quad (\text{since } \dot Q \cong \dot W \cong \Delta\text{pe} \cong 0)$$

$$0 = h_2 - h_1 + \frac{\mathbf{V}_2^2 - \mathbf{V}_1^2}{2}$$

Substituting,

$$0 = (274.70 - 338.19)\text{kJ/kg} + \frac{\mathbf{V}_2^2 - (20\text{m/s})^2}{2}\left(\frac{1\text{kJ/kg}}{1000\text{m}^2/\text{s}^2}\right)$$

It yields $\mathbf{V_2 = 356.9\ m/s}$

(*b*) The ratio of the inlet to exit area is determined from the conservation of mass relation,

$$\frac{1}{v_2}A_2\mathbf{V}_2 = \frac{1}{v_1}A_1\mathbf{V}_1 \longrightarrow \frac{A_1}{A_2} = \frac{v_2}{v_1}\frac{\mathbf{V}_1}{\mathbf{V}_2} = \frac{(0.04064\text{m}^3/\text{kg})(356.9\text{m/s})}{(0.07767\text{m}^3/\text{kg})(20\text{m/s})} = \mathbf{9.34}$$

5-71 Air is decelerated in a diffuser from 220 m/s. The exit velocity and the exit pressure of air are to be determined.

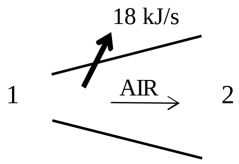
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with variable specific heats. **3** Potential energy changes are negligible. **4** There are no work interactions.

Properties The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1). The enthalpies are (Table A-17)

$$T_1 = 27°\text{C} = 300 \text{ K} \quad \rightarrow \quad h_1 = 300.19 \text{ kJ/kg}$$
$$T_2 = 42°\text{C} = 315 \text{ K} \quad \rightarrow \quad h_2 = 315.27 \text{ kJ/kg}$$

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

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



$$\dot{E}_{in} = \dot{E}_{out}$$
$$\dot{m}(h_1 + \mathbf{V}_1^2/2) = \dot{Q}_{out} + \dot{m}(h_2 + \mathbf{V}_2^2/2) \quad (\text{since } \dot{W} \cong \Delta\text{pe} \cong 0)$$
$$-\dot{Q}_{out} = \dot{m}\left(h_2 - h_1 + \frac{\mathbf{V}_2^2 - \mathbf{V}_1^2}{2}\right)$$

Substituting, the exit velocity of the air is determined to be

$$-18 \text{ kJ/s} = (2.5 \text{ kg/s})\left(315.27 - 300.19 + \frac{\mathbf{V}_2^2 - (220 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)\right)$$

It yields **$\mathbf{V_2}$ = 62.0 m/s**

(*b*) The exit pressure of air is determined from the conservation of mass and the ideal gas relations,

$$\dot{m} = \frac{1}{v_2}A_2\mathbf{V}_2 \longrightarrow v_2 = \frac{A_2\mathbf{V}_2}{\dot{m}} = \frac{(0.04\text{m}^2)(62\text{m/s})}{2.5\text{kg/s}} = 0.992\text{m}^3/\text{kg}$$

and

$$P_2 v_2 = RT_2 \longrightarrow P_2 = \frac{RT_2}{v_2} = \frac{(0.287\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(315\text{K})}{0.992\text{m}^3/\text{kg}} = \mathbf{91.1kPa}$$

5-72 Nitrogen is decelerated in a diffuser from 200 m/s to a lower velocity. The exit velocity of nitrogen and the ratio of the inlet-to-exit area are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Nitrogen is an ideal gas with variable specific heats. **3** Potential energy changes are negligible. **4** The device is adiabatic and thus heat transfer is negligible. **5** There are no work interactions.

Properties The molar mass of nitrogen is M = 28 kg/kmol (Table A-1). The enthalpies are (Table A-18)

$$T_1 = 7°\text{C} = 280\,\text{K} \;\rightarrow\; \bar{h}_1 = 8141\,\text{kJ/kmol}$$
$$T_2 = 22°\text{C} = 295\,\text{K} \;\rightarrow\; \bar{h}_2 = 8580\,\text{kJ/kmol}$$

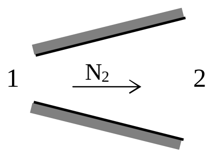
Analysis (a) There is only one inlet and one exit, and thus $\dot m_1 = \dot m_2 = \dot m$. We take diffuser as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

$$\dot E_{in} = \dot E_{out}$$

$$\dot m(h_1 + \mathbf{V}_1^2/2) = \dot m(h_2 + \mathbf{V}_2^2/2) \quad (\text{since } \dot Q \cong \dot W \cong \Delta\text{pe} \cong 0)$$

$$0 = h_2 - h_1 + \frac{\mathbf{V}_2^2 - \mathbf{V}_1^2}{2} = \frac{\bar{h}_2 - \bar{h}_1}{M} + \frac{\mathbf{V}_2^2 - \mathbf{V}_1^2}{2},$$



Substituting,

$$0 = \frac{(8580 - 8141)\,\text{kJ/kmol}}{28\,\text{kJ/kmol}} + \frac{\mathbf{V}_2^2 - (200\text{m/s})^2}{2}\left(\frac{1\text{kJ/kg}}{1000\text{m}^2/\text{s}^2}\right)$$

It yields $\mathbf{V_2 = 93.0\ m/s}$

(b) The ratio of the inlet to exit area is determined from the conservation of mass relation,

$$\frac{1}{v_2}A_2\mathbf{V}_2 = \frac{1}{v_1}A_1\mathbf{V}_1 \longrightarrow \frac{A_1}{A_2} = \frac{v_1}{v_2}\frac{\mathbf{V}_2}{\mathbf{V}_1} = \left(\frac{RT_1/P_1}{RT_2/P_2}\right)\frac{\mathbf{V}_2}{\mathbf{V}_1}$$

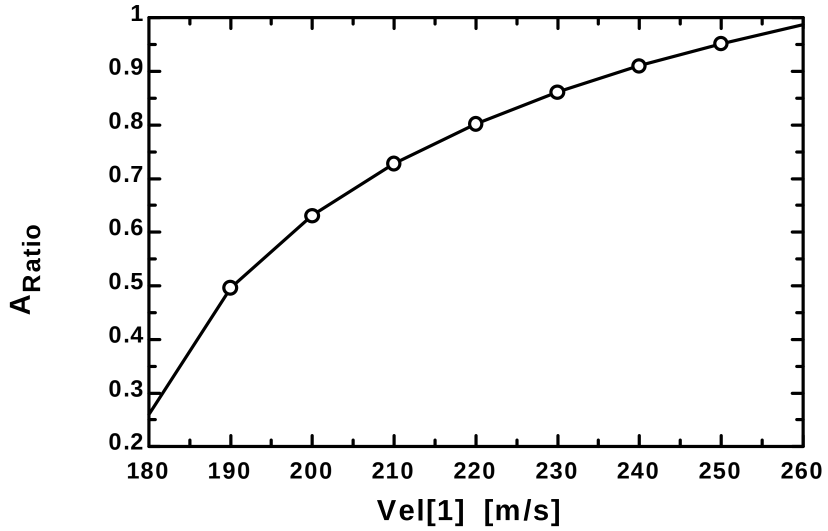
or,

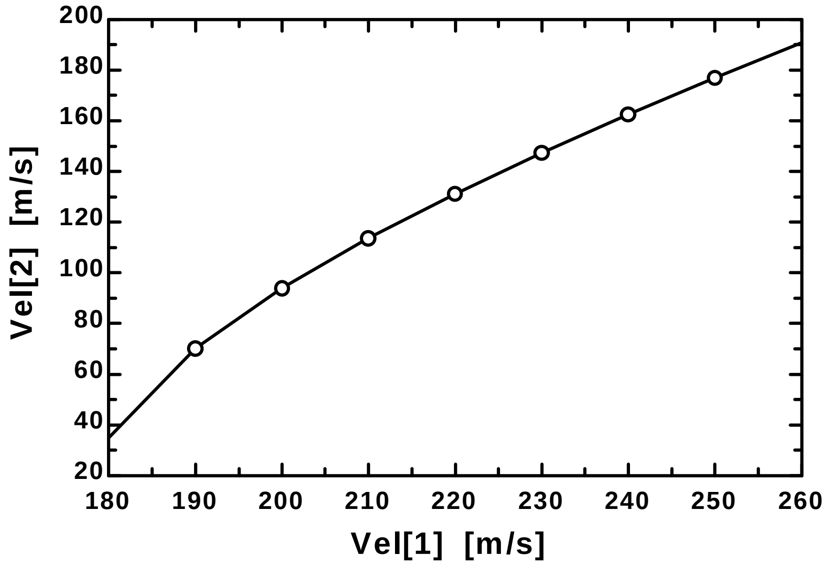
$$\frac{A_1}{A_2} = \left(\frac{T_1/P_1}{T_2/P_2}\right)\frac{\mathbf{V}_2}{\mathbf{V}_1} = \frac{(280\text{ K/60 kPa})(93.0\text{ m/s})}{(295\text{ K/85 kPa})(200\text{ m/s})} = \mathbf{0.625}$$

5-73 Problem 5-72 is reconsidered. The effect of the inlet velocity on the exit velocity and the ratio of the inlet-to-exit area as the inlet velocity varies from 180 m/s to 260 m/s is to be investigated. The final results are to be plotted against the inlet velocity.

```
Function HCal(WorkFluid$, Tx, Px)
"Function to calculate the enthalpy of an ideal gas or real gas"
   If   'N2' = WorkFluid$  then
        HCal:=ENTHALPY(WorkFluid$,T=Tx) "Ideal gas equ."
   else
       HCal:=ENTHALPY(WorkFluid$,T=Tx, P=Px)"Real gas equ."
   endif
end  HCal

"System: control volume for the nozzle"

"Property relation: Nitrogen is an ideal gas"

"Process: Steady state, steady flow, adiabatic, no work"

"Knowns"
WorkFluid$ = 'N2'
T[1] = 7  "[C]"
P[1] = 60   "[kPa]"
{Vel[1] = 200  "[m/s]"}
P[2] = 85  "[kPa]"
T[2] = 22 "[C]"

"Property Data - since the Enthalpy function has different parameters
for ideal gas and real fluids, a function was used to determine h."
h[1]=HCal(WorkFluid$,T[1],P[1])"[kJ/kg]"
h[2]=HCal(WorkFluid$,T[2],P[2])"[kJ/kg]"

"The Volume function has the same form for an ideal gas as for a real fluid."
v[1]=volume(workFluid$,T=T[1],p=P[1])"[m^3/kg]"
v[2]=volume(WorkFluid$,T=T[2],p=P[2])"[m^3/kg]"

"From the definition of mass flow rate, m_dot = A*Vel/v and conservation of mass the area
ratio A_Ratio = A_1/A_2 is:"
A_Ratio*Vel[1]/v[1] =Vel[2]/v[2]

"Conservation of Energy - SSSF energy balance"
h[1]+Vel[1]^2/(2*1000) = h[2]+Vel[2]^2/(2*1000)
```

A_{Ratio}	Vel_1 [m/s]	Vel_2 [m/s]
0.2603	180	34.84
0.4961	190	70.1
0.6312	200	93.88
0.7276	210	113.6
0.8019	220	131.2
0.8615	230	147.4
0.9106	240	162.5
0.9518	250	177
0.9869	260	190.8



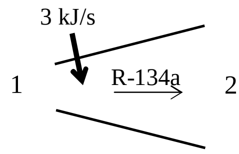


5-74 R-134a is decelerated in a diffuser from a velocity of 140 m/s. The exit velocity of R-134a and the mass flow rate of the R-134a are to be determined.

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

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

$$\left.\begin{array}{l} P_1 = 700\text{kPa} \\ sat.vapor \end{array}\right\} \begin{array}{l} v_1 = 0.0292\text{m}^3/\text{kg} \\ h_1 = 261.85\text{kJ/kg} \end{array}$$



and

$$\left.\begin{array}{l} P_2 = 800\text{kPa} \\ T_2 = 40^{\text{o}}\text{C} \end{array}\right\} \begin{array}{l} v_2 = 0.02691\text{m}^3/\text{kg} \\ h_2 = 273.66\text{kJ/kg} \end{array}$$

Analysis (a) There is only one inlet and one exit, and thus $\dot m_1 = \dot m_2 = \dot m$. Then the exit velocity of R-134a is determined from the steady-flow mass balance to be

$$\frac{1}{v_2} A_2 \mathbf{V}_2 = \frac{1}{v_1} A_1 \mathbf{V}_1 \longrightarrow \mathbf{V}_2 = \frac{v_2}{v_1}\frac{A_1}{A_2}\mathbf{V}_1 = \frac{1}{1.8}\frac{0.02691\text{m}^3/\text{kg}}{0.02920\text{m}^3/\text{kg}}(140\text{m/s}) = \mathbf{71.7m/s}$$

(b) We take diffuser as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

$$\dot E_{in} = \dot E_{out}$$

$$\dot Q_{\text{in}} + \dot m(h_1 + \mathbf{V}_1^2/2) = \dot m(h_2 + \mathbf{V}_2^2/2) \quad (\text{since } \dot W \cong \Delta\text{pe} \cong 0)$$

$$\dot Q_{\text{in}} = \dot m\left(h_2 - h_1 + \frac{\mathbf{V}_2^2 - \mathbf{V}_1^2}{2}\right)$$

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

$$3\text{kJ/s} = \dot m\left(273.66 - 261.85 + \frac{(71.7\text{m/s})^2 - (140\text{m/s})^2}{2}\left(\frac{1\text{kJ/kg}}{1000\text{m}^2/\text{s}^2}\right)\right)$$

It yields $\dot m = \mathbf{0.655\ kg/s}$

Turbines and Compressors

5-75C Yes.

5-76C The volume flow rate at the compressor inlet will be greater than that at the compressor exit.

5-77C Yes. Because energy (in the form of shaft work) is being added to the air.

5-78C No.

5-79 Steam expands in a turbine. The change in kinetic energy, the power output, and the turbine inlet area are to be determined.

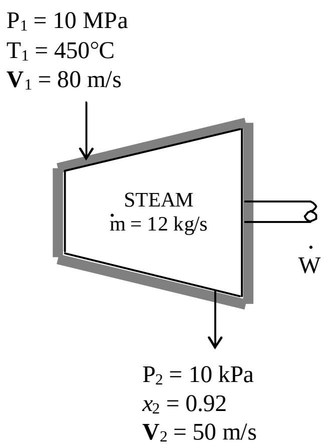
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

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

$$\left.\begin{array}{l} P_1 = 10\text{MPa} \\ T_1 = 450^\circ\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.02975\text{m}^3/\text{kg} \\ h_1 = 3240.9\text{kJ/kg} \end{array}$$

and

$$\left.\begin{array}{l} P_2 = 10\text{ kPa} \\ x_2 = 0.92 \end{array}\right\} h_2 = h_f + x_2 h_{fg} = 191.83 + 0.92 \times 2392.8 = 2393.2\text{kJ/kg}$$



Analysis (a) The change in kinetic energy is determined from

$$\Delta ke = \frac{\mathbf{V}_2^2 - \mathbf{V}_1^2}{2} = \frac{(50\text{m/s})^2 - (80\text{m/s})^2}{2}\left(\frac{1\text{kJ/kg}}{1000\text{m}^2/\text{s}^2}\right) = \mathbf{-1.95\text{kJ/kg}}$$

(*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

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

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

$$\dot{m}(h_1 + \mathbf{V}_1^2/2) = \dot{W}_{\text{out}} + \dot{m}(h_2 + \mathbf{V}_2^2/2) \quad (\text{since } \dot{Q} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{out}} = -\dot{m}\left(h_2 - h_1 + \frac{\mathbf{V}_2^2 - \mathbf{V}_1^2}{2}\right)$$

Then the power output of the turbine is determined by substitution to be

$$\dot{W}_{\text{out}} = -(12\text{ kg/s})(2393.2 - 3240.9 - 1.95)\text{kJ/kg} = \mathbf{10.2\ MW}$$

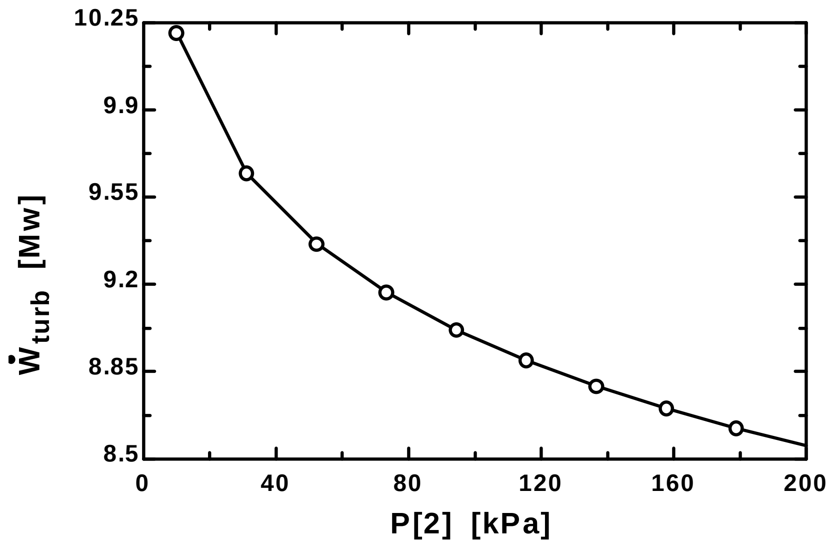
(*c*) The inlet area of the turbine is determined from the mass flow rate relation,

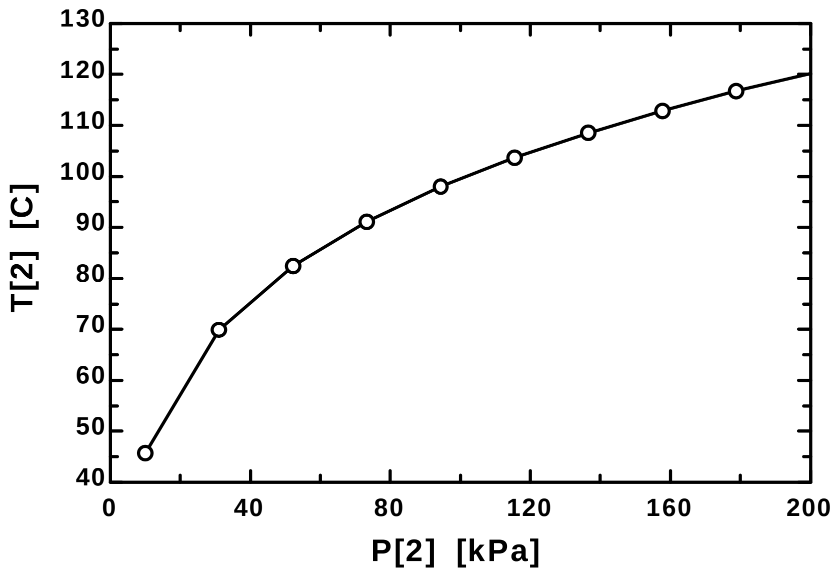
$$\dot{m} = \frac{1}{v_1}A_1\mathbf{V}_1 \longrightarrow A_1 = \frac{\dot{m}v_1}{\mathbf{V}_1} = \frac{(12\text{ kg/s})(0.02975\text{ m}^3/\text{kg})}{80\text{ m/s}} = \mathbf{0.00446\ m^2}$$

5-80 Problem 5-79 is reconsidered. The effect of the turbine exit pressure on the power output of the turbine as the exit pressure varies from 10 kPa to 200 kPa is to be investigated. The power output is to be plotted against the exit pressure.

```
"Knowns "
T[1] = 450  "[C]"
P[1] = 10000   "[kPa]"
Vel[1] = 80  "[m/s]"
P[2] = 10  "[kPa]"
X_2=0.92
Vel[2] = 50  "[m/s]"
m_dot[1]=12"[kg/s]"

"Property Data"
h[1]=enthalpy(Steam,T=T[1],P=P[1])"[kJ/kg]"
h[2]=enthalpy(Steam,P=P[2],x=x_2)"[kJ/kg]"

T[2]=temperature(Steam,P=P[2],x=x_2)"[C]"

v[1]=volume(Steam,T=T[1],p=P[1])"[m^3/kg]"
v[2]=volume(Steam,P=P[2],x=x_2)"[m^3/kg]"

"Conservation of mass: "
m_dot[1]= m_dot[2]

"Mass flow rate"
m_dot[1]=A[1]*Vel[1]/v[1] "[kg/s]"
m_dot[2]= A[2]*Vel[2]/v[2] "[kg/s]"

"Conservation of Energy - Steady Flow energy balance"
m_dot[1]*(h[1]+Vel[1]^2/(2*1000)) =
m_dot[2]*(h[2]+Vel[2]^2/(2*1000))+W_dot_turb*convert(MW,kJ/s)

DELTAke=Vel[2]^2/(2*1000)-Vel[1]^2/(2*1000)"[kJ/kg]"
```

P_2 [kPa]	W_{turb} [MW]	T_2 [C]
10	10.21	45.79
31.11	9.645	69.92
52.22	9.362	82.4
73.33	9.167	91.16
94.44	9.018	98.02
115.6	8.895	103.7
136.7	8.792	108.6
157.8	8.703	112.9
178.9	8.624	116.7
200	8.553	120.2


10.25
9.9
9.55
9.2
8.85
8.5
$\dot{W}_{turb}$ [Mw]
0
40
80
120
160
200
P[2] [kPa]


130
120
110
100
90
80
70
60
50
40
T[2] [C]
0
40
80
120
160
200
P[2] [kPa]

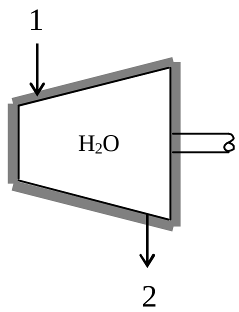
5-81 Steam expands in a turbine. The mass flow rate of steam for a power output of 5 MW is to be determined.

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.

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

$$\left.\begin{array}{l} P_1 = 10\text{MPa} \\ T_1 = 400^{\circ}\text{C} \end{array}\right\} h_1 = 3096.5\text{kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 20\text{kPa} \\ x_2 = 0.90 \end{array}\right\} h_2 = h_f + x_2 h_{fg} = 251.40 + 0.90 \times 2358.3 = 2373.9\text{kJ/kg}$$



Analysis 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

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

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

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

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

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

$$5000 \text{ kJ/s} = -\dot{m}(2373.9 - 3096.5) \text{ kJ/kg}$$

$$\dot{m} = \mathbf{6.919 \ \ kg/s}$$

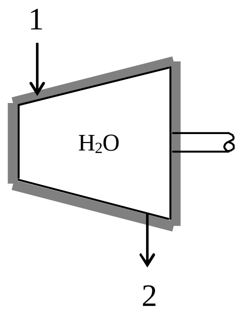
5-82E Steam expands in a turbine. The rate of heat loss from the steam for a power output of 4 MW is to be determined.

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

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

$$\left.\begin{array}{l} P_1 = 1000\,\text{psia} \\ T_1 = 900^{\circ}\text{F} \end{array}\right\} h_1 = 1448.1\,\text{Btu/lbm}$$

$$\left.\begin{array}{l} P_2 = 5\,\text{psia} \\ \textit{sat.vapor} \end{array}\right\} h_2 = 1131.0\,\text{Btu/lbm}$$



Analysis 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

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

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

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

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

Substituting,

$$\dot{Q}_{\text{out}} = -(45000/3600\,\text{lbm/s})(45000/3600\,\text{lbm/s})\text{Btu/lbm} - 4000\,\text{kJ/s}\left(\frac{1\,\text{Btu}}{1.055\,\text{kJ}}\right)$$

$$= \mathbf{172.3\ Btu/s}$$

5-83 Steam expands in a turbine. The exit temperature of the steam for a power output of 2 MW is to be determined.

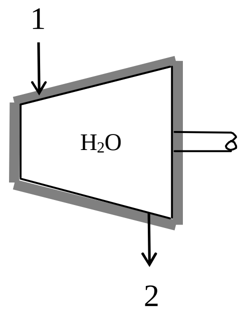
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

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

$$\left.\begin{array}{l} P_1 = 10\text{MPa} \\ T_1 = 500^\circ\text{C} \end{array}\right\} h_1 = 3373.7\text{kJ/kg}$$

Analysis 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

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



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

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

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

Substituting,

$$2000\text{kJ/s} = (3\text{kg/s})(3373.7 - h_2)\text{kJ/kg}$$

$$h_2 = 2707\text{kJ/kg}$$

Then the exit temperature becomes

$$\left.\begin{array}{l} P_2 = 20\text{kPa} \\ h_2 = 2707\text{kJ/kg} \end{array}\right\} T_2 = \mathbf{110.8^\circ C}$$

5-84 Argon gas expands in a turbine. The exit temperature of the argon for a power output of 250 kW is 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.

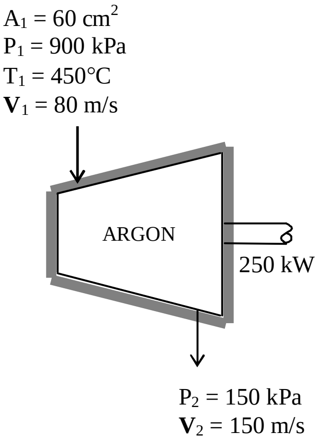
Properties The gas constant of Ar is R = 0.2081 kPa.m^3/kg.K. The constant pressure specific heat of Ar is C_p = 0.5203 kJ/kg·°C (Tables A-2a)

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The inlet specific volume of argon and its mass flow rate are

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.2081\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(723\,\text{K})}{900\,\text{kPa}} = 0.167\,\text{m}^3/\text{kg}$$

Thus,

$$\dot{m} = \frac{1}{v_1}A_1V_1 = \frac{1}{0.167\,\text{m}^3/\text{kg}}(0.006\,\text{m}^2)(80\,\text{m/s}) = 2.874\,\text{kg/s}$$



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

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

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

$$\dot{m}(h_1 + \mathbf{V}_1^2/2) = \dot{W}_{out} + \dot{m}(h_2 + \mathbf{V}_2^2/2) \quad (\text{since } \dot{Q} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{out} = -\dot{m}\left(h_2 - h_1 + \frac{\mathbf{V}_2^2 - \mathbf{V}_1^2}{2}\right)$$

Substituting,

$$250\,\text{kJ/s} = -(2.874\,\text{kg/s})\left[(0.5203\,\text{kJ/kg}\cdot^\circ\text{C})(T_2 - 450^\circ\text{C}) + \frac{(150\,\text{m/s})^2 - (80\,\text{m/s})^2}{2}\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right)\right]$$

It yields T_2 = **267.3°C**

5-85E Air expands in a turbine. The mass flow rate of air and the power output of the turbine are to be determined.

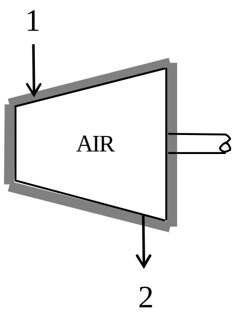
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Air is an ideal gas with constant specific heats.

Properties The gas constant of air is R = 0.3704 psia.ft^3/lbm.R. The constant pressure specific heat of air at the average temperature of (900 + 300)/2 = 600°F is C_p = 0.25 Btu/lbm·°F (Tables A-2a)

Analysis (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The inlet specific volume of air and its mass flow rate are

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.3704\,\text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(1360\,\text{R})}{150\,\text{psia}} = 3.358\,\text{ft}^3/\text{lbm}$$

$$\dot{m} = \frac{1}{v_1}A_1V_1 = \frac{1}{3.358\,\text{ft}^3/\text{lbm}}(0.1\,\text{ft}^2)(350\,\text{ft/s}) = \mathbf{10.42\,lbm/s}$$



(*b*) 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

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

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

$$\dot{m}(h_1 + \mathbf{V}_1^2/2) = \dot{W}_{\text{out}} + \dot{m}(h_2 + \mathbf{V}_2^2/2) \quad (\text{since } \dot{Q} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{out}} = -\dot{m}\left(h_2 - h_1 + \frac{\mathbf{V}_2^2 - \mathbf{V}_1^2}{2}\right) = -\dot{m}\left(C_p(T_2 - T_1) + \frac{\mathbf{V}_2^2 - \mathbf{V}_1^2}{2}\right)$$

Substituting,

$$\dot{W}_{\text{out}} = -(10.42\text{ lbm/s})\left[(0.250\text{ Btu/lbm}\cdot^{\circ}\text{F})(300 - 900)^{\circ}\text{F} + \frac{(700\text{ ft/s})^2 - (350\text{ ft/s})^2}{2}\left(\frac{1\text{ Btu/lbm}}{25{,}037\text{ ft}^2/\text{s}^2}\right)\right]$$

$$= 1486.5\text{ Btu/s} = \mathbf{1568\ \ kW}$$

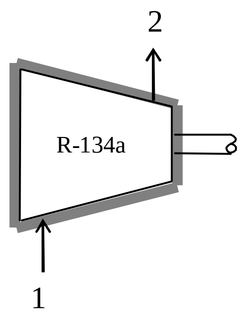
5-86 Refrigerant-134a is compressed steadily by a compressor. The power input to the compressor and the volume flow rate of the refrigerant at the compressor inlet are to be determined.

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

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

$$\left.\begin{array}{l} T_1 = -20^\circ\text{C} \\ sat.vapor \end{array}\right\} \begin{array}{l} v_1 = 0.1464\ \text{m}^3/\text{kg} \\ h_1 = 235.31\ \text{kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 0.7\ \text{MPa} \\ T_2 = 70^\circ\text{C} \end{array}\right\} h_2 = 307.01\ \text{kJ/kg}$$



Analysis (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the 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 in the rate form as

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

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

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

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

Substituting,

$$\dot{W}_{in} = (1.2\ \text{kg/s})(307.01 - 235.31)\text{kJ/kg}$$
$$= \mathbf{86.04\ kJ/s}$$

(*b*) The volume flow rate of the refrigerant at the compressor inlet is

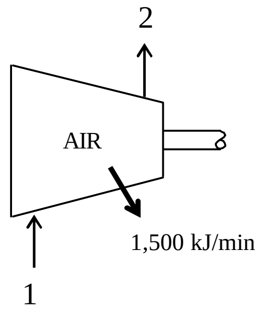
$$\dot{V}_1 = \dot{m}v_1 = (1.2\ \text{kg/s})(0.1464\ \text{m}^3/\text{kg}) = \mathbf{0.176\ m^3/s}$$

5-87 Air is compressed by a compressor. The mass flow rate of air through the compressor is 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** Air is an ideal gas with variable specific heats.

Properties The inlet and exit enthalpies of air are (Table A-17)

$$T_1 = 25°\text{C} = 298 \text{ K} \quad \rightarrow \quad h_1 = h_{@\,298\text{ K}} = 298.2 \text{ kJ/kg}$$
$$T_2 = 347°\text{C} = 620 \text{ K} \quad \rightarrow \quad h_2 = h_{@\,620\text{ K}} = 628.07 \text{ kJ/kg}$$



Analysis We take the 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 in the rate form as

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

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

$$\dot{W}_{\text{in}} + \dot{m}(h_1 + \mathbf{V}_1^2/2) = \dot{Q}_{\text{out}} + \dot{m}(h_2 + \mathbf{V}_2^2/2) \quad (\text{since } \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{in}} - \dot{Q}_{\text{out}} = \dot{m}\left(h_2 - h_1 + \frac{\mathbf{V}_2^2 - \mathbf{V}_1^2}{2}\right)$$

Substituting, the mass flow rate is determined to be

$$250 \text{ kJ/s} - (1500/60 \text{ kJ/s}) = \dot{m}\left[628.07 - 298.2 + \frac{(90 \text{ m/s})^2 - 0}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)\right]$$

$$\dot{m} = \mathbf{0.674 \text{ kg/s}}$$

5-88E Air is compressed by a compressor. The mass flow rate of air through the compressor and the exit temperature of air 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** Air is an ideal gas with variable specific heats.

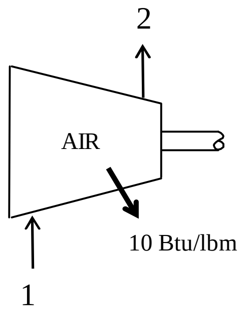
Properties The gas constant of air is R = 0.3704 psia.ft^3/lbm.R (Table A-1). The inlet enthalpy of air is (Table A-17E)

$$T_1 = 60°\text{F} = 520\text{ R} \qquad \rightarrow \qquad h_1 = h_{@\,520\text{ R}} = 124.27\text{ Btu/lbm}$$

Analysis (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The inlet specific volume of air and its mass flow rate are

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.3704\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(520\text{ R})}{14.7\text{ psia}} = 13.1\text{ ft}^3/\text{lbm}$$

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{5000\text{ ft}^3/\text{min}}{13.1\text{ ft}^3/\text{lbm}} = 381.7\text{ lbm/min} = \mathbf{6.36\text{ lbm/s}}$$



(*b*) We take the 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 in the rate form as

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

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

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

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

Substituting,

$$(700\text{ hp})\left(\frac{0.7068\text{ Btu/s}}{1\text{ hp}}\right) - (6.36\text{ lbm/s}) \times (10\text{ Btu/lbm}) = (6.36\text{ lbm/s})(h_2 - 124.27\text{ Btu/lbm})$$

$$h_2 = 192.06\text{ Btu/lbm}$$

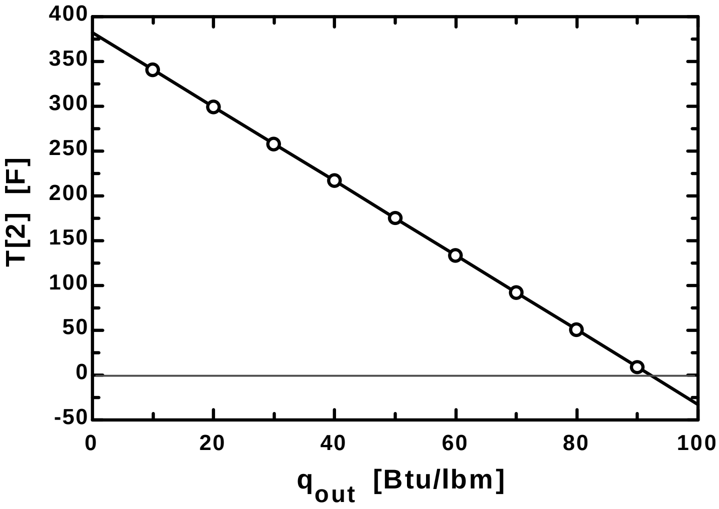
Then the exit temperature is determined from Table A-17E to be

$$T_2 = 801\text{ R} = \mathbf{341°F}$$

5-89E Problem 5-88E is reconsidered. The effect of the rate of cooling of the compressor on the exit temperature of air as the cooling rate varies from 0 to 100 Btu/lbm is to be investigated. The air exit temperature is to be plotted against the rate of cooling.

```
"Knowns "
T[1] = 60  "[F]"
P[1] = 14.7   "[psia]"
V_dot[1] = 5000  "[ft^3/min]"
P[2] = 150  "[psia]"
{q_out=10"[Btu/lbm]"}
W_dot_in=700"[hp]"

"Property Data"
h[1]=enthalpy(Air,T=T[1])"[Btu/lbm]"
h[2]=enthalpy(Air,T=T[2])"[Btu/lbm]"

TR_2=T[2]+460"[R]"

v[1]=volume(Air,T=T[1],p=P[1])"[ft^3/lbm]"
v[2]=volume(Air,T=T[2],p=P[2])"[ft^3/lbm]"

"Conservation of mass: "
m_dot[1]= m_dot[2]

"Mass flow rate"
m_dot[1]=V_dot[1]/v[1] *convert(ft^3/min,ft^3/s)"[lbm/s]"
m_dot[2]= V_dot[2]/v[2]*convert(ft^3/min,ft^3/s) "[lbm/s]"

"Conservation of Energy - Steady Flow energy balance"
W_dot_in*convert(hp,Btu/s)+m_dot[1]*(h[1]) = m_dot[1]*q_out+m_dot[1]*(h[2])
```

q_{out} [Btu/lbm]	T_2 [F]
0	382
10	340.9
20	299.7
30	258.3
40	216.9
50	175.4
60	133.8
70	92.26
80	50.67
90	9.053
100	-32.63



5-90 Helium is compressed by a compressor. For a mass flow rate of 90 kg/min, the power input required 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** Helium is an ideal gas with constant specific heats.

Properties The constant pressure specific heat of helium is C_p = 5.1926 kJ/kg·K (Table A-2a).

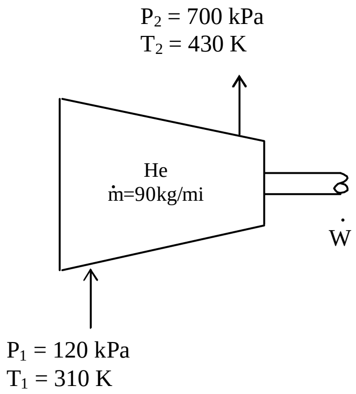
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

$$\dot{W}_{\text{in}} - \dot{Q}_{\text{out}} = \dot{m}(h_2 - h_1) = \dot{m}C_p(T_2 - T_1)$$



Thus,

$$\begin{aligned}\dot{W}_{in} &= \dot{Q}_{out} + \dot{m}C_p\left(T_2 - T_1\right)\\ &= (90/60\,\text{kg/s})(20\,\text{kJ/kg}) + (90/60\,\text{kg/s})(5.1926\,\text{kJ/kg}\cdot\text{K})(430 - 310)\text{K}\\ &= \mathbf{965\,kW}\end{aligned}$$

5-91 CO_2 is compressed by a compressor. The volume flow rate of CO_2 at the compressor inlet 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** Helium is an ideal gas with variable specific heats. **4** The device is adiabatic and thus heat transfer is negligible.

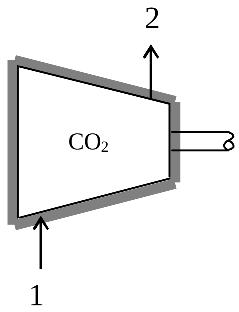
Properties The gas constant of CO_2 is R = 0.1889 kPa.m^3/kg.K, and its molar mass is M = 44 kg/kmol (Table A-1). The inlet and exit enthalpies of CO_2 are (Table A-20)

$$T_1 = 300\text{ K} \quad \rightarrow \quad \bar{h}_1 = 9{,}431\text{ kJ/kmol}$$
$$T_2 = 450\text{ K} \quad \rightarrow \quad \bar{h}_2 = 15{,}483\text{ kJ/kmol}$$

Analysis (a) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The inlet specific volume of air and its volume flow rate are

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.1889\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{K})}{100\text{kPa}} = 0.5667\text{m}^3/\text{kg}$$
$$\dot{V} = \dot{m}v_1 = (0.5\text{ kg/s})(0.5667\text{ m}^3/\text{kg}) = \mathbf{0.283\ m^3/s}$$

(b) We take the 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 in the rate form as



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

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

$$\dot{W}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$
$$\dot{W}_{in} = \dot{m}(h_2 - h_1) = \dot{m}(\bar{h}_2 - \bar{h}_1)/M$$

$$\dot{W}_{in} = \frac{(0.5\text{kg/s})(15{,}483 - 9{,}431\text{kJ/kmol})}{44\text{kg/kmol}} = \mathbf{68.8kW}$$

Throttling Valves

5-92C Because usually there is a large temperature drop associated with the throttling process.

5-93C Yes.

5-94C No. Because air is an ideal gas and h = h(T) for ideal gases. Thus if h remains constant, so does the temperature.

5-95C If it remains in the liquid phase, no. But if some of the liquid vaporizes during throttling, then yes.

5-96 Refrigerant-134a is throttled by a valve. The temperature drop of the refrigerant and specific volume after expansion 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 to or from the fluid is negligible. **4** There are no work interactions involved.

Properties The inlet enthalpy of R-134a is, from the refrigerant tables (Tables A-11 through 13),

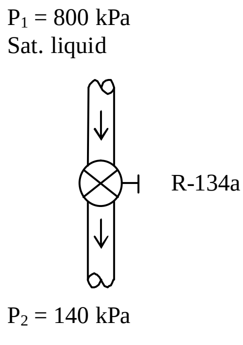
$$\left.\begin{array}{l} P_1 = 0.8 MPa \\ sat.liquid \end{array}\right\} \begin{array}{l} T_1 = T_{sat} = 31.33^{\circ}C \\ h_1 = h_f = 93.42 kJ/kg \end{array}$$

Analysis There is only one inlet and one exit, and thus $\dot m_1 = \dot m_2 = \dot m$. We take the throttling valve 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

$$\dot E_{in} - \dot E_{out} = \Delta \dot E_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \quad \rightarrow \quad \dot E_{in} = \dot E_{out} \quad \rightarrow \quad \dot m h_1 = \dot m h_2 \quad \rightarrow \quad h_1 = h_2$$

since $\dot Q \cong \dot W = \Delta ke \cong \Delta pe \cong 0$. Then,

$$\left.\begin{array}{l} P_2 = 0.14\text{MPa} \\ (h_2 = h_1) \end{array}\right\} \begin{array}{l} h_f = 25.77\text{kJ/kg}, \quad T_{sat} = -18.8^{\circ}\text{C} \\ h_g = 236.04\text{kJ/kg} \end{array}$$



Obviously $h_f < h_2 < h_g$, thus the refrigerant exists as a saturated mixture at the exit state and thus $T_2 = T_{sat}$ = -18.8°C. Then the temperature drop becomes

$$\Delta T = T_2 - T_1 = -18.8 - 31.33 = \mathbf{-50.13^{\circ}C}$$

The quality at this state is determined from

$$x_2 = \frac{h_2 - h_f}{h_{fg}} = \frac{93.42 - 25.77}{210.27 - 25.77} = 0.322$$

Thus,

$$v_2 = v_f + x_2 v_{fg} = 0.0007381 + 0.322 \times 0.13876 = \mathbf{0.0454 m^3/kg}$$

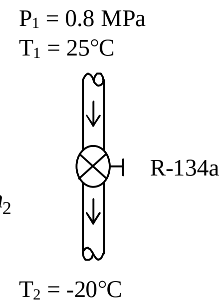
5-97 [*Also solved by EES on enclosed CD***]** Refrigerant-134a is throttled by a valve. The pressure and internal energy after expansion are to be determined.

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 to or from the fluid is negligible. **4** There are no work interactions involved.

Properties The inlet enthalpy of R-134a is, from the refrigerant tables (Tables A-11 through 13),

$$\left.\begin{array}{l} P_1 = 0.8\text{MPa} \\ T_1 = 25^\circ\text{C} \end{array}\right\} h_1 \cong h_{f @ 25^\circ\text{C}} = 84.33\text{kJ/kg}$$

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



$$\dot{E}_{in} - \dot{E}_{out} = \Delta\dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \quad \rightarrow \quad \dot{E}_{in} = \dot{E}_{out} \quad \rightarrow \quad \dot{m}h_1 = \dot{m}h_2 \quad \rightarrow \quad h_1 = h_2$$

since $\dot{Q} \cong \dot{W} = \Delta ke \cong \Delta pe \cong 0$. Then,

$$\left.\begin{array}{l} T_2 = -20^\circ\text{C} \\ (h_2 = h_1) \end{array}\right\} \begin{array}{ll} h_f = 24.26\text{kJ/kg}, & u_f = 24.17\text{kJ/kg} \\ h_g = 235.31\text{kJ/kg} & u_g = 215.84\text{kJ/kg} \end{array}$$

Obviously $h_f < h_2 < h_g$, thus the refrigerant exists as a saturated mixture at the exit state, and thus

$$P_2 = P_{\text{sat @ -20°C}} = \mathbf{0.13299\ MPa}$$

Also, $$x_2 = \frac{h_2 - h_f}{h_{fg}} = \frac{84.33 - 24.26}{211.05} = 0.285$$

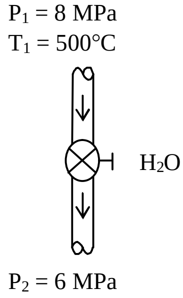
Thus, $$u_2 = u_f + x_2 u_{fg} = 24.17 + 0.285 \times (215.84 - 24.17) = \mathbf{78.8\ kJ/kg}$$

5-98 Steam is throttled by a well-insulated valve. The temperature drop of the steam after the expansion 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** Heat transfer to or from the fluid is negligible. **4** There are no work interactions involved.

Properties The inlet enthalpy of steam is (Tables A-6),

$$\left.\begin{array}{l} P_1 = 8\text{MPa} \\ T_1 = 500^\circ\text{C} \end{array}\right\} h_1 = 3398.3\text{kJ/kg}$$



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

$$\dot{E}_{in} - \dot{E}_{out} = \Delta\dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \quad \rightarrow \quad \dot{E}_{in} = \dot{E}_{out} \quad \rightarrow \quad \dot{m}h_1 = \dot{m}h_2 \quad \rightarrow \quad h_1 = h_2$$

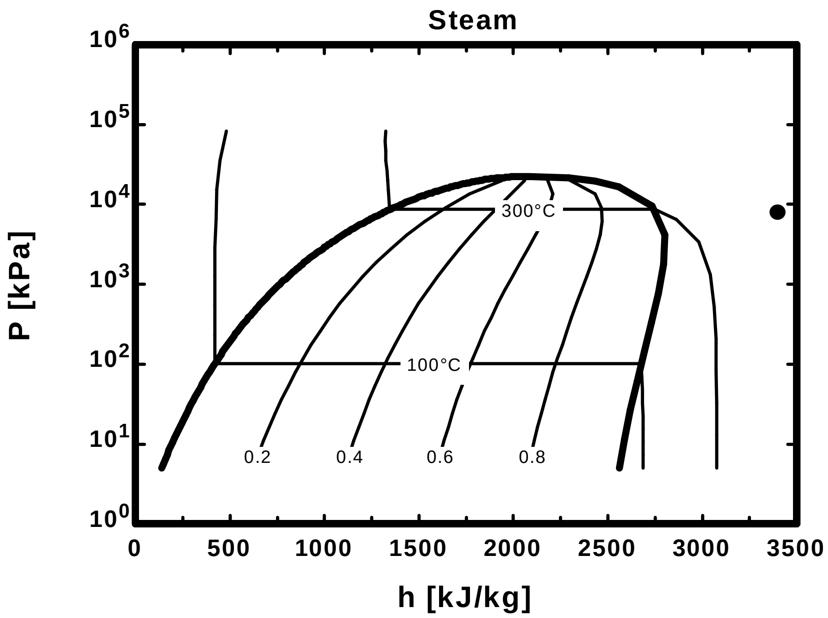
since $\dot{Q} \cong \dot{W} = \Delta ke \cong \Delta pe \cong 0$. Then the exit temperature of steam becomes

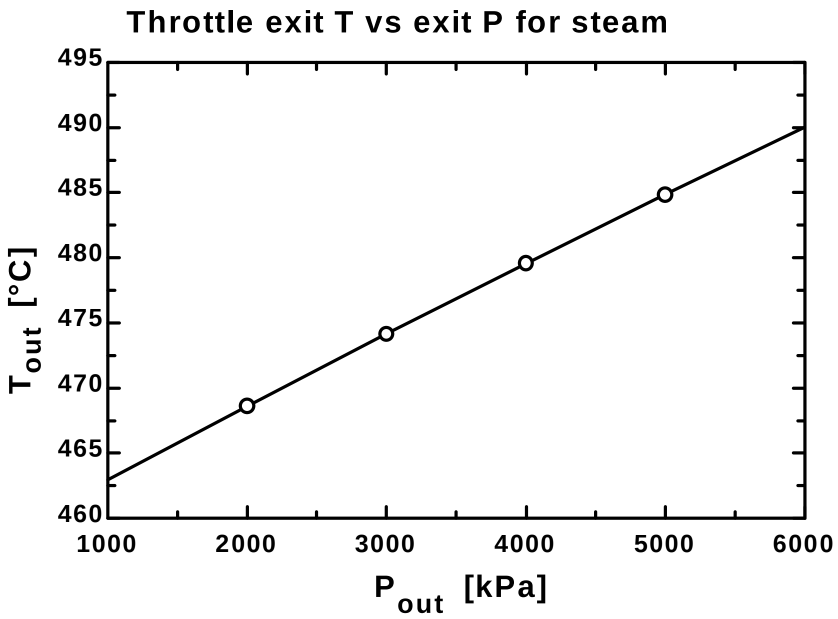
$$\left.\begin{array}{l} P_2 = 6\text{MPa} \\ (h_2 = h_1) \end{array}\right\} T_2 = \mathbf{490.1^\circ C}$$

5-99 Problems 5-98 is reconsidered. The effect of the exit pressure of steam on the exit temperature after throttling as the exit pressure varies from 6 MPa to 1 MPa is to be investigated. The exit temperature of steam is to be plotted against the exit pressure.

```
"Input information from Diagram Window"
{WorkingFluid$='Steam'        "WorkingFluid: can be  changed to ammonia or other fluids"
P_in=8000             "[kPa]"
T_in=500              "[C]"
P_out=6000            "[C]"}
$Warning off
"Analysis"
m_dot_in=m_dot_out        "steady-state mass balance"
m_dot_in=1            "mass flow rate is arbitrary"
m_dot_in*h_in+Q_dot-W_dot-m_dot_out*h_out=0        "steady-state energy balance"
Q_dot=0          "assume the throttle to operate adiabatically"
W_dot=0          "throttles do not have any means of producing power"
h_in=enthalpy(WorkingFluid$,T=T_in,P=P_in)    "property table lookup"
T_out=temperature(WorkingFluid$,P=P_out,h=h_out)    "property table lookup"
x_out=quality(WorkingFluid$,P=P_out,h=h_out)        "x_out is the quality at the outlet"

P[1]=P_in;  P[2]=P_out;  h[1]=h_in;  h[2]=h_out  "use arrays to place points on property plot"
```

P_{out} [kPa]	T_{out} [C]
1000	463
2000	468.6
3000	474.2
4000	479.6
5000	484.9
6000	490

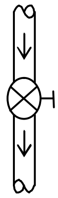



Throttle exit T vs exit P for steam
495
490
485
480
475
470
465
460
1000
2000
3000
4000
5000
6000
Tout [°C]
Pout [kPa]

5-100E High-pressure air is throttled to atmospheric pressure. The temperature of air after the expansion 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** Heat transfer to or from the fluid is negligible. **4** There are no work interactions involved. **5** Air is an ideal gas.

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



$$\dot{E}_{in} - \dot{E}_{out} = \Delta\dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \quad \rightarrow \quad \dot{E}_{in} = \dot{E}_{out} \quad \rightarrow \quad \dot{m}h_1 = \dot{m}h_2 \quad \rightarrow \quad h_1 = h_2$$

since $\dot{Q} \cong \dot{W} = \Delta ke \cong \Delta pe \cong 0$. For an ideal gas, $h = h(T)$.

Therefore,

$$T_2 = T_1 = \mathbf{90°F}$$

Mixing Chambers and Heat Exchangers

5-101C Yes, if the mixing chamber is losing heat to the surrounding medium.

5-102C Under the conditions of no heat and work interactions between the mixing chamber and the surrounding medium.

5-103C Under the conditions of no heat and work interactions between the heat exchanger and the surrounding medium.

5-104 A hot water stream is mixed with a cold water stream. For a specified mixture temperature, the mass flow rate of cold water is 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. **4** Fluid properties are constant. **5** There are no work interactions.

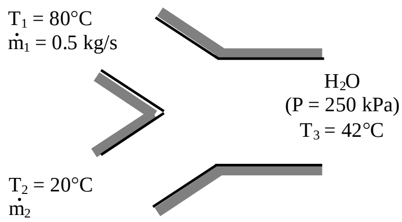
Properties Noting that $T < T_{sat\,@\,250\text{ kPa}} = 127.44°C$, the water in all three streams exists as a compressed liquid, which can be approximated as a saturated liquid at the given temperature. Thus,

$$\begin{aligned} h_1 &\cong h_{f@\,80°C} = 334.91 \text{ kJ/kg} \\ h_2 &\cong h_{f@\,20°C} = 83.96 \text{ kJ/kg} \\ h_3 &\cong h_{f@\,42°C} = 175.92 \text{ kJ/kg} \end{aligned}$$

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

Energy balance:



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

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

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

Combining the two relations and solving for $\dot{m}_2$ gives

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = (\dot{m}_1 + \dot{m}_2) h_3$$

$$\dot{m}_2 = \frac{h_1 - h_3}{h_3 - h_2} \dot{m}_1$$

Substituting, the mass flow rate of cold water stream is determined to be

$$\dot{m}_2 = \frac{(334.91 - 175.92)\text{kJ/kg}}{(175.92 - 83.96)\text{kJ/kg}} (0.5 \text{ kg/s}) = \mathbf{0.864 \text{ kg/s}}$$

5-105 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 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** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

Properties Noting that $T < T_{sat\ @\ 300\ kPa}$ = 133.55°C, the cold water stream and the mixture exist as a compressed liquid, which can be approximated as a saturated liquid at the given temperature. Thus,

$$h_1 \cong h_{f\ @\ 20°C} = 83.96 \text{ kJ/kg}$$
$$h_3 \cong h_{f\ @\ 60°C} = 251.13 \text{ kJ/kg}$$

and

$$\left.\begin{array}{l} P_2 = 300 \text{kPa} \\ T_2 = 300^{\circ}\text{C} \end{array}\right\} h_2 = 3069.3 \text{kJ/kg}$$

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

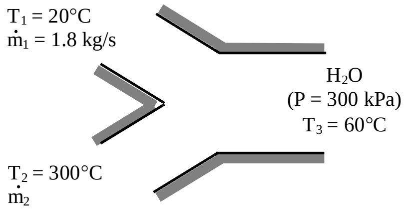
Mass balance: $\dot{m}_{in} - \dot{m}_{out} = \Delta \dot{m}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \quad \rightarrow \quad \dot{m}_{in} = \dot{m}_{out} \quad \rightarrow \quad \dot{m}_1 + \dot{m}_2 = \dot{m}_3$

Energy balance:

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

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

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



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

Solving for $\dot{m}_2$: $\dot{m}_2 = \dfrac{h_1 - h_3}{h_3 - h_2} \dot{m}_1$

Substituting,

$$\dot{m}_2 = \frac{83.96 - 251.13}{251.13 - 3069.3}(1.8 \text{ kg/s}) = \mathbf{0.107 \text{ kg/s}}$$

5-106 Feedwater is heated in a chamber by mixing it with superheated steam. If the mixture is saturated liquid, the ratio of the mass flow rates of the feedwater and the superheated vapor 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** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

Properties Noting that $T < T_{\text{sat @ 800 kPa}}$ = 170.43°C, the cold water stream and the mixture exist as a compressed liquid, which can be approximated as a saturated liquid at the given temperature. Thus,

$$h_1 \cong h_{f\,@\,50°\text{C}} = 209.33 \text{ kJ/kg}$$
$$h_3 \cong h_{f\,@\,800\text{ kPa}} = 721.11 \text{ kJ/kg}$$

and

$$\left.\begin{array}{l} P_2 = 800\text{kPa} \\ T_2 = 200^\circ\text{C} \end{array}\right\} h_2 = 2839.3\text{kJ/kg}$$

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

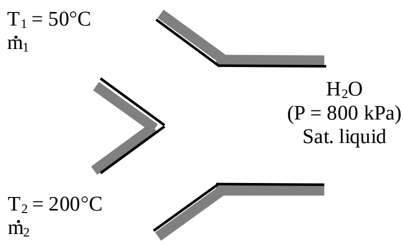
Mass balance: $$\dot{m}_{in} - \dot{m}_{out} = \Delta \dot{m}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \quad \rightarrow \quad \dot{m}_{in} = \dot{m}_{out} \quad \rightarrow \quad \dot{m}_1 + \dot{m}_2 = \dot{m}_3$$

Energy balance:

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

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

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



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

Dividing by $\dot{m}_2$ yields $$y h_1 + h_2 = (y+1) h_3$$

Solving for y: $$y = \frac{h_3 - h_2}{h_1 - h_3}$$

where $y = \dot{m}_1 / \dot{m}_2$ is the desired mass flow rate ratio. Substituting,

$$y = \frac{721.11 - 2839.3}{209.33 - 721.11} = \mathbf{4.14}$$

5-107E Liquid water is heated in a chamber by mixing it with saturated water vapor. If both streams enter at the same rate, the temperature and quality (if saturated) of the exit stream 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** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

Properties From steam tables (Tables A-5 through A-6),

$$h_1 \cong h_{f\,@\,50°F} = 18.06 \text{ Btu/lbm}$$
$$h_2 = h_{g\,@\,50\text{ psia}} = 1174.4 \text{ Btu/lbm}$$

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

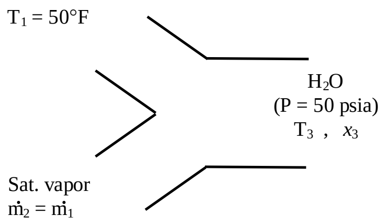
Mass balance: $\dot m_{in} - \dot m_{out} = \Delta \dot m_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \rightarrow \dot m_{in} = \dot m_{out} \rightarrow \dot m_1 + \dot m_2 = \dot m_3 = 2\dot m \rightarrow \dot m_1 = \dot m_2 = \dot m$

Energy balance:

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

$$\dot E_{in} = \dot E_{out}$$

$$\dot m_1 h_1 + \dot m_2 h_2 = \dot m_3 h_3 \quad (\text{since } \dot Q \cong \dot W \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$



Combining the two gives $\dot m h_1 + \dot m h_2 = 2\dot m h_3$ or $h_3 = (h_1 + h_2)/2$

Substituting,

$$h_3 = (18.06 + 1174.4)/2 = 596.23 \text{ Btu/lbm}$$

At 50 psia, $h_f = 250.24$ Btu/lbm and $h_g = 1174.4$ Btu/lbm. Thus the exit stream is a saturated mixture since
$h_f < h_3 < h_g$. Therefore,

$$T_3 = T_{sat\,@\,50\text{ psia}} = \mathbf{281.03°F}$$

and

$$x_3 = \frac{h_3 - h_f}{h_{fg}} = \frac{596.23 - 250.24}{924.2} = \mathbf{0.374}$$

5-108 Two streams of refrigerant-134a are mixed in a chamber. If the cold stream enters at twice the rate of the hot stream, the temperature and quality (if saturated) of the exit stream 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. **4** The device is adiabatic and thus heat transfer is negligible.

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

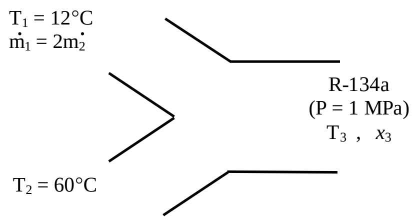
$$h_1 \cong h_{f @ 12°C} = 66.18 \text{ kJ/kg}$$
$$h_2 = h_{@ 1 \text{ MPa}, \ 60°C} = 291.36 \text{ kJ/kg}$$

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

Mass balance: $\dot{m}_{in} - \dot{m}_{out} = \Delta \dot{m}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \rightarrow \dot{m}_{in} = \dot{m}_{out} \rightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3 = 3\dot{m}_2$ since $\dot{m}_1 = 2\dot{m}_2$

Energy balance:

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



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

Combining the two gives $2\dot{m}_2 h_1 + \dot{m}_2 h_2 = 3\dot{m}_2 h_3$ *or* $h_3 = (2h_1 + h_2)/3$

Substituting,

$$h_3 = (2 \times 66.18 + 291.36)/3 = 141.24 \text{ kJ/kg}$$

At 1 MPa, $h_f = 105.29$ kJ/kg and $h_g = 267.97$ kJ/kg. Thus the exit stream is a saturated mixture since $h_f < h_3 < h_g$. Therefore,

$$T_3 = T_{\text{sat @ 1 MPa}} = \mathbf{39.39°C}$$

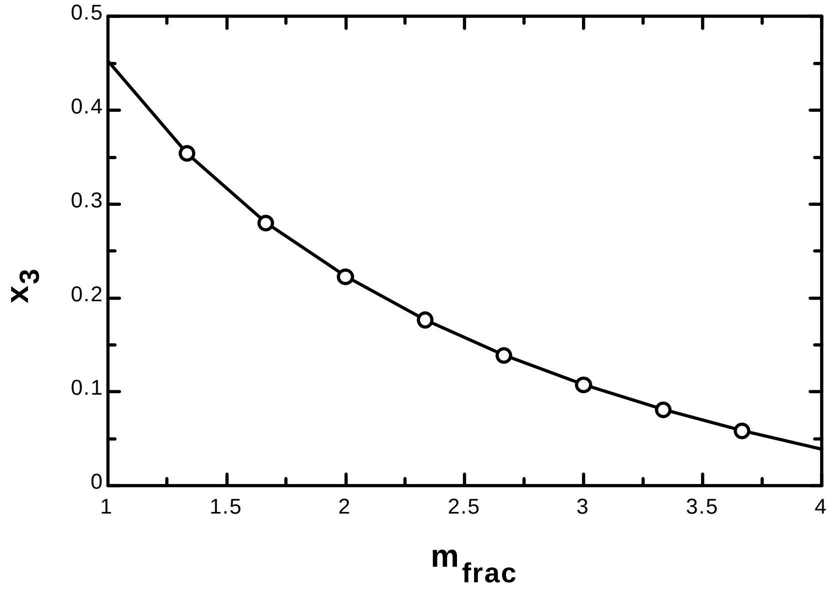
and

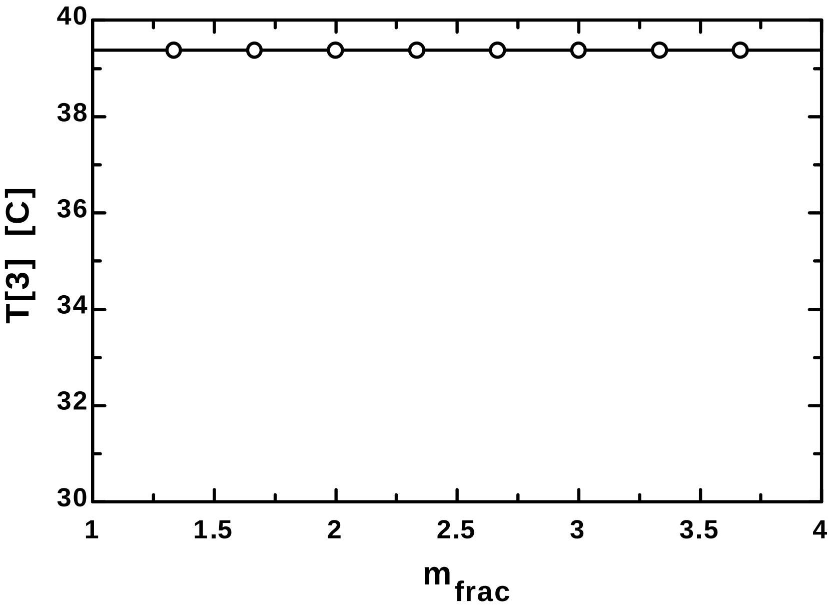
$$x_3 = \frac{h_3 - h_f}{h_{fg}} = \frac{141.24 - 105.29}{162.68} = \mathbf{0.221}$$

5-109 Problem 5-108 is reconsidered. The effect of the mass flow rate of the cold stream of R-134a on the temperature and the quality of the exit stream as the ratio of the mass flow rate of the cold stream to that of the hot stream varies from 1 to 4 is to be investigated. The mixture temperature and quality are to be plotted against the cold-to-hot mass flow rate ratio.

```
"Input Data"
m_frac = 2  "m_frac =m_dot_cold/m_dot_hot= m_dot_1/m_dot_2"
T[1]=12"[C]"
P[1]=1000"[kPa]"
T[2]=60"[C]"
P[2]=1000"[kPa]"
m_dot_1=m_frac*m_dot_2
P[3]=1000"[kPa]"
m_dot_1=1

"Conservation of mass for the R134a: Sum of m_dot_in=m_dot_out"

m_dot_1+ m_dot_2 =m_dot_3

"Conservation of Energy for steady-flow: neglect changes in KE and PE"
"We assume no heat transfer and no work occur across the control surface."
E_dot_in - E_dot_out = DELTAE_dot_cv
DELTAE_dot_cv=0       "Steady-flow requirement"
E_dot_in=m_dot_1*h[1] + m_dot_2*h[2]
E_dot_out=m_dot_3*h[3]

"Property data are given by:"
h[1] =enthalpy(R134a,T=T[1],P=P[1])     "R134a data"
h[2] =enthalpy(R134a,T=T[2],P=P[2])
T[3] =temperature(R134a,P=P[3],h=h[3])
x_3=QUALITY(R134a,h=h[3],P=P[3])
```

m_{frac}	T_3 [C]	x_3
1	39.38	0.4523
1.333	39.38	0.3538
1.667	39.38	0.28
2	39.38	0.2225
2.333	39.38	0.1766
2.667	39.38	0.139
3	39.38	0.1077
3.333	39.38	0.08117
3.667	39.38	0.05845
4	39.38	0.03876


0.5
0.4
0.3
0.2
0.1
0
1
1.5
2
2.5
3
3.5
4
x3
mfrac


40
38
36
34
32
30
1
1.5
2
2.5
3
3.5
4
T[3] [C]
mfrac

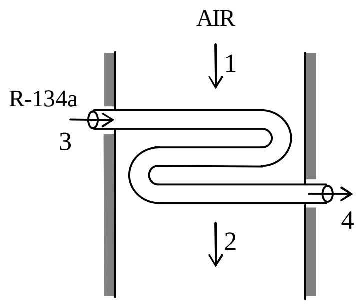
5-110 Refrigerant-134a is to be cooled by air in the condenser. For a specified volume flow rate of air, the mass flow rate of the refrigerant is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** Heat loss from the device to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **5** 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 is C_p = 1.005 kJ/kg·°C (Table A-2). The enthalpies of the R-134a at the inlet and the exit states are (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_3 = 1\ \text{MPa} \\ T_3 = 80^\circ\text{C} \end{array}\right\} h_3 = 313.20\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 1\ \text{MPa} \\ T_4 = 30^\circ\text{C} \end{array}\right\} h_4 \cong h_{f@30^\circ C} = 91.49\ \text{kJ/kg}$$



Analysis The inlet specific volume and the mass flow rate of air are

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{K})}{100\text{kPa}} = 0.861\text{m}^3/\text{kg}$$

and

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{800\text{m}^3/\text{min}}{0.861\text{m}^3/\text{kg}} = 929.2\text{kg/min}$$

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*):

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

Energy balance (for the entire heat exchanger):

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

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

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

Combining the two, $\dot{m}_a(h_2 - h_1) = \dot{m}_R(h_3 - h_4)$

Solving for $\dot{m}_R$: $\dot{m}_R = \dfrac{h_2 - h_1}{h_3 - h_4}\dot{m}_a \cong \dfrac{C_p(T_2 - T_1)}{h_3 - h_4}\dot{m}_a$

Substituting,

$$\dot{m}_R = \frac{(1.005\ \text{kJ/kg}\cdot^\circ\text{C})(60-27)^\circ\text{C}}{(313.20 - 91.49)\ \text{kJ/kg}}(929.2\ \text{kg/min}) = \mathbf{139.0\ \ kg/min}$$

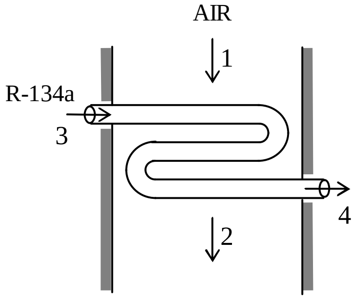
5-111E Refrigerant-134a is vaporized by air in the evaporator of an air-conditioner. For specified flow rates, the exit temperature of the air and the rate of heat transfer from the air are to be determined.

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. **4** Heat loss from the device to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **5** Air is an ideal gas with constant specific heats at room temperature.

Properties The gas constant of air is 0.3704 psia.ft^3/lbm.R (Table A-1E). The constant pressure specific heat of air is C_p = 0.240 Btu/lbm·°F (Table A-2E). The enthalpies of the R-134a at the inlet and the exit states are (Tables A-11E through A-13E)

$$\left.\begin{array}{l} P_3 = 20 \text{ psia} \\ x_3 = 0.3 \end{array}\right\} h_3 = h_f + x_3 h_{fg} = 10.89 + 0.3 \times 90.50 = 38.04 \text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_4 = 20 \text{ psia} \\ \text{sat.vapor} \end{array}\right\} h_4 = h_{g@20 \text{ psia}} = 101.39 \text{ Btu/lbm}$$



Analysis The inlet specific volume and the mass flow rate of air are

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(550\text{R})}{14.7\text{psia}} = 13.86 \text{ ft}^3/\text{lbm}$$

and

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{200 \text{ ft}^3/\text{min}}{13.86 \text{ ft}^3/\text{lbm}} = 14.43 \text{ lbm/min}$$

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*):

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

Energy balance (for the entire heat exchanger):

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

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

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

Combining the two, $\dot{m}_R(h_3 - h_4) = \dot{m}_a(h_2 - h_1) = \dot{m}_a C_p (T_2 - T_1)$

Solving for T_2: $T_2 = T_1 + \dfrac{\dot{m}_R(h_3 - h_4)}{\dot{m}_a C_p}$

Substituting, $T_2 = 90^\circ\text{F} + \dfrac{(4 \text{ lbm/min})(38.04 - 101.39)\text{Btu/lbm}}{(14.43 \text{ Btu/min})(0.24 \text{ Btu/lbm}\cdot{}^\circ\text{F})} = \mathbf{16.8^\circ F}$

(*b*) The rate of heat transfer from the air to the refrigerant is determined from the steady-flow energy balance applied to the air only. It yields

$$-\dot{Q}_{air,out} = \dot{m}_a (h_2 - h_1) = \dot{m}_a C_p (T_2 - T_1)$$

$$\dot{Q}_{air,out} = -(14.43\text{lbm/min})(0.24\text{Btu/lbm}\cdot{}^\circ\text{F})(16.8 - 90)^\circ\text{F}$$

$$= \mathbf{253.5 \ Btu/min}$$

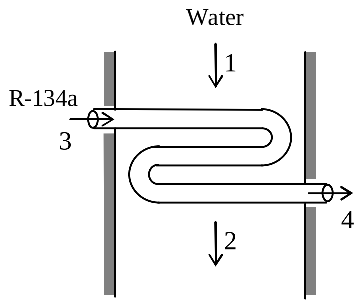
5-112 Refrigerant-134a is condensed in a water-cooled condenser. The mass flow rate of the cooling water required is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** Heat loss from the device to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid.

Properties The enthalpies of R-134a at the inlet and the exit states are (Tables A-5 and A-6)

$$\left.\begin{array}{l} P_3 = 800\text{kPa} \\ T_3 = 70^0\text{C} \end{array}\right\} h_3 = 305.50\text{kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 800\text{kPa} \\ sat.liquid \end{array}\right\} h_4 = h_{f\,@800\text{kPa}} = 93.42\text{kJ/kg}$$



Water exists as compressed liquid at both states, and thus

$$h_1 \cong h_{f\,@\,15°C} = 62.99 \text{ kJ/kg}$$
$$h_2 \cong h_{f\,@\,30°C} = 125.79 \text{ kJ/kg}$$

Analysis We take the 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):

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

Energy balance (for the heat exchanger):

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

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

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

Combining the two, $\dot{m}_w(h_2 - h_1) = \dot{m}_R(h_3 - h_4)$

Solving for $\dot{m}_w$: $\dot{m}_w = \dfrac{h_3 - h_4}{h_2 - h_1}\dot{m}_R$

Substituting,

$$\dot{m}_w = \frac{(305.50 - 93.42) \text{ kJ/kg}}{(125.79 - 62.99) \text{ kJ/kg}}(8 \text{ kg/min}) = \mathbf{27.0 \text{ kg/min}}$$

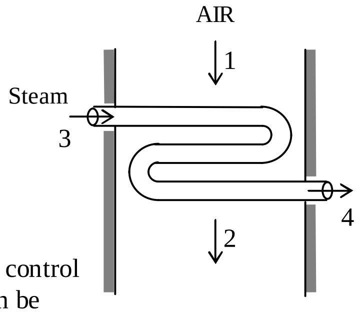
5-113E [*Also solved by EES on enclosed CD*] Air is heated in a steam heating system. For specified flow rates, the volume flow rate of air at the inlet is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** Heat loss from the device to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **5** Air is an ideal gas with constant specific heats at room temperature.

Properties The gas constant of air is 0.3704 psia.ft^3/lbm.R (Table A-1E). The constant pressure specific heat of air is C_p = 0.240 Btu/lbm·°F (Table A-2E). The enthalpies of steam at the inlet and the exit states are (Tables A-4E through A-6E)

$$\left.\begin{array}{l} P_3 = 30\text{psia} \\ T_3 = 400^\circ\text{F} \end{array}\right\} h_3 = 1237.8\text{Btu/lbm}$$

$$\left.\begin{array}{l} P_4 = 25\text{psia} \\ T_4 = 212^\circ F \end{array}\right\} h_4 \cong h_{f@212^\circ\text{F}} = 180.16\text{Btu/lbm}$$



Analysis 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*):

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

Energy balance (for the entire heat exchanger):

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

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

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

Combining the two, $\dot{m}_a(h_2 - h_1) = \dot{m}_s(h_3 - h_4)$

Solving for $\dot{m}_a$: $\dot{m}_a = \dfrac{h_3 - h_4}{h_2 - h_1}\dot{m}_s \cong \dfrac{h_3 - h_4}{C_p(T_2 - T_1)}\dot{m}_s$

Substituting,

$$\dot{m}_a = \frac{(1237.8 - 180.16)\text{ Btu/lbm}}{(0.240\text{ Btu/lbm}\cdot^\circ\text{F})(130 - 80)^\circ\text{F}}(15\text{ lbm/min}) = 1322\text{ lbm/min} = 22.03\text{ lbm/s}$$

Also, $v_1 = \dfrac{RT_1}{P_1} = \dfrac{(0.3704\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(540\text{ R})}{14.7\text{ psia}} = 13.61\text{ ft}^3/\text{lbm}$

Then the volume flow rate of air at the inlet becomes

$$\dot{V}_1 = \dot{m}_a v_1 = (22.03\text{ lbm/s})(13.61\text{ ft}^3/\text{lbm}) = \mathbf{299.8\text{ ft}^3/\text{s}}$$

5-114 Steam is condensed by cooling water in the condenser of a power plant. If the temperature rise of the cooling water is not to exceed 10°C, the minimum mass flow rate of the cooling water required 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** There are no work interactions. **4** Heat loss from the device to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **5** Liquid water is an incompressible substance with constant specific heats at room temperature.

Properties The cooling water exists as compressed liquid at both states, and its specific heat at room temperature is C = 4.18 kJ/kg·°C (Table A-3). The enthalpies of the steam at the inlet and the exit states are (Tables A-5 and A-6)

$$\left.\begin{array}{l} P_3 = 20\text{kPa} \\ x_3 = 0.95 \end{array}\right\} h_3 = h_f + x_3 h_{fg} = 251.40 + 0.95 \times 2358.3 = 2491.8 \text{kJ/kg}$$

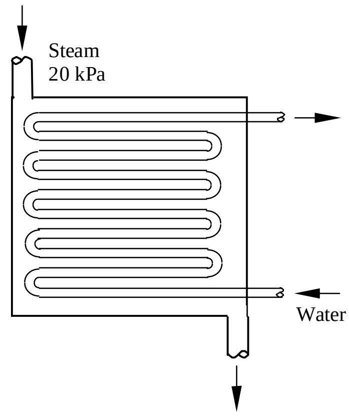
$$\left.\begin{array}{l} P_4 = 20\text{kPa} \\ sat.liquid \end{array}\right\} h_4 \cong h_{f@20kPa} = 251.40 \text{kJ/kg}$$

Analysis We take the 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):

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

Energy balance (for the heat exchanger):



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

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

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

Combining the two, $\dot{m}_w (h_2 - h_1) = \dot{m}_s (h_3 - h_4)$

Solving for $\dot{m}_w$: $\dot{m}_w = \dfrac{h_3 - h_4}{h_2 - h_1} \dot{m}_s \cong \dfrac{h_3 - h_4}{C_p (T_2 - T_1)} \dot{m}_s$

Substituting,

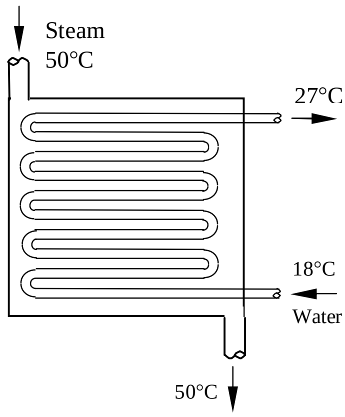
$$\dot{m}_w = \frac{(2491.8 - 251.4)\text{kJ/kg}}{(4.18 \text{kJ/kg}\cdot{}^\circ\text{C})(10^\circ\text{C})} (20{,}000/3600 \text{kg/s}) = \mathbf{298\ kg/s} = 17{,}866 \text{ kg/min}$$

5-115 Steam is condensed by cooling water in the condenser of a power plant. The rate of condensation of steam is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The heat of vaporization of water at 50°C is h_{fg} = 2382.7 kJ/kg and specific heat of cold water is C_p = 4.18 kJ/kg.°C (Tables A-3 and A-4).

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



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

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

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

$$\dot{Q}_{in} = \dot{m}C_p(T_2 - T_1)$$

Then the heat transfer rate to the cooling water in the condenser becomes

$$\begin{aligned}\dot{Q} &= [\dot{m}C_p(T_{out} - T_{in})]_{\text{cooling water}} \\ &= (101\text{ kg/s})(4.18\text{ kJ/kg.°C})(27°\text{C} - 18°\text{C}) \\ &= 3800\text{ kJ/s}\end{aligned}$$

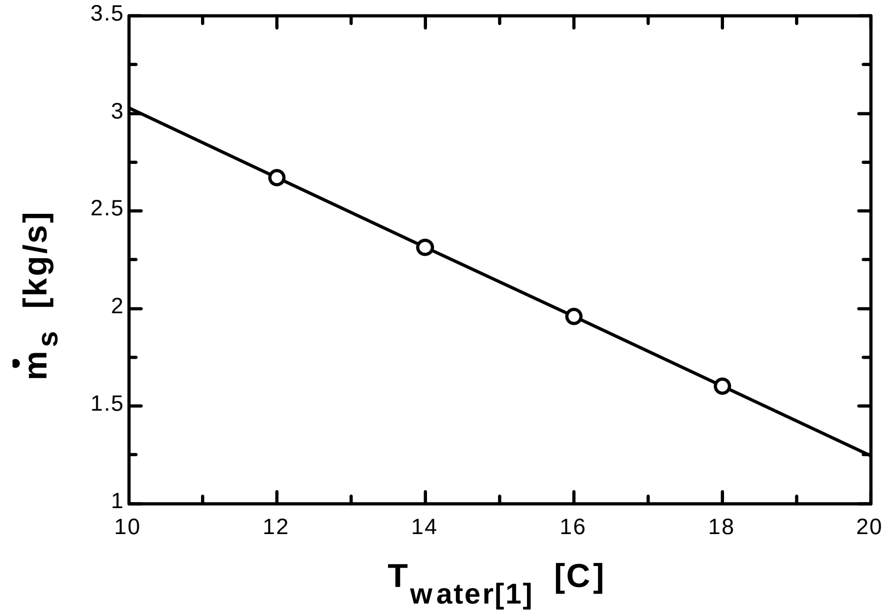
The rate of condensation of steam is determined to be

$$\dot{Q} = (\dot{m}h_{fg})_{\text{steam}} \longrightarrow \dot{m}_{\text{steam}} = \frac{\dot{Q}}{h_{fg}} = \frac{3800\text{ kJ/s}}{2382.7\text{kJ/kg}} = \mathbf{1.59\ kg/s}$$

5-116 Problem 5-115 is reconsidered. The effect of the inlet temperature of cooling water on the rate of condensation of steam as the inlet temperature varies from 10°C to 20°C at constant exit temperature is to be investigated. The rate of condensation of steam is to be plotted against the inlet temperature of the cooling water.

```
"Input Data"
T_s[1]=50"[C]"
T_s[2]=50"[C]"
m_dot_water=101 "[kg/s]"
T_water[1]=18"[C]"
T_water[2]=27"[C]"
C_P_water = 4.20 "[kJ/kg-°C]"

"Conservation of mass for the steam: m_dot_s_in=m_dot_s_out=m_dot_s"
"Conservation of mass for the water: m_dot_water_in=lm_dot_water_out=m_dot_water"

"Conservation of Energy for steady-flow: neglect changes in KE and PE"
"We assume no heat transfer and no work occur across the control surface."
E_dot_in - E_dot_out = DELTAE_dot_cv
DELTAE_dot_cv=0 "[kW]" "Steady-flow requirement"
E_dot_in=m_dot_s*h_s[1] + m_dot_water*h_water[1] "[kW]"
E_dot_out=m_dot_s*h_s[2] + m_dot_water*h_water[2] "[kW]"

"Property data are given by:"
h_s[1] =enthalpy(steam,T=T_s[1],x=1)"[kJ/kg]" "steam data"
h_s[2] =enthalpy(steam,T=T_s[2],x=0)"[kJ/kg]"
h_water[1] =C_P_water*T_water[1]"[kJ/kg]" "water data"
h_water[2] =C_P_water*T_water[2]"[kJ/kg]"

h_fg_s=h_s[1]-h_s[2] "kJ/kg]"  "h_fg is found from the EES functions rather than using h_fg
= 2305 kJ/kg"
```

m_s [kg/s]	$T_{water,1}$ [C]
3.028	10
2.671	12
2.315	14
1.959	16
1.603	18
1.247	20


3.5
3
2.5
2
1.5
1
10
12
14
16
18
20
ṁs [kg/s]
Twater[1] [C]

5-117 Water is heated in a heat exchanger by geothermal water. The rate of heat transfer to the water and the exit temperature of the geothermal water is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The specific heats of water and geothermal fluid are given to be 4.18 and 4.31 kJ/kg.°C, respectively.

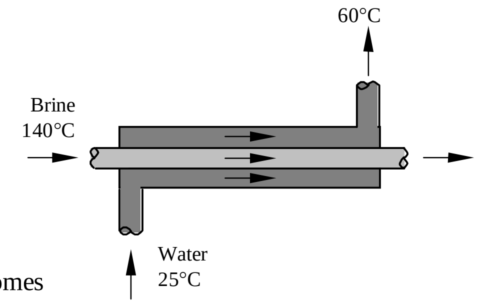
Analysis We take the cold water tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

$$\dot{Q}_{in} = \dot{m}C_p(T_2 - T_1)$$



Then the rate of heat transfer to the cold water in the heat exchanger becomes

$$\dot{Q} = [\dot{m}C_p(T_{out} - T_{in})]_{\text{water}} = (0.2\ \text{kg/s})(4.18\ \text{kJ/kg.°C})(60°\text{C} - 25°\text{C}) = \mathbf{29.26\ kW}$$

Noting that heat transfer to the cold water is equal to the heat loss from the geothermal water, the outlet temperature of the geothermal water is determined from

$$\dot{Q} = [\dot{m}C_p(T_{in} - T_{out})]_{\text{geot.water}} \longrightarrow T_{out} = T_{in} - \frac{\dot{Q}}{\dot{m}C_p}$$

$$= 140°\text{C} - \frac{29.26\ \text{kW}}{(0.3\ \text{kg/s})(4.31\ \text{kJ/kg.°C})} = \mathbf{117.4°C}$$

5-118 Ethylene glycol is cooled by water in a heat exchanger. The rate of heat transfer in the heat exchanger and the mass flow rate of water are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The specific heats of water and ethylene glycol are given to be 4.18 and 2.56 kJ/kg.°C, respectively.

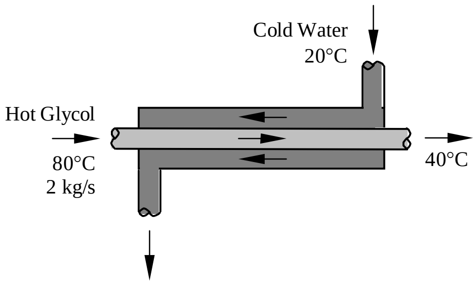
Analysis (a) We take the ethylene glycol tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

$$\dot{Q}_{out} = \dot{m}C_p(T_1 - T_2)$$



Then the rate of heat transfer becomes

$$\dot{Q} = [\dot{m}C_p(T_{in} - T_{out})]_{\text{glycol}} = (2 \text{ kg/s})(2.56 \text{ kJ/kg.°C})(80°\text{C} - 40°\text{C}) = \mathbf{204.8\ kW}$$

(b) The rate of heat transfer from water must be equal to the rate of heat transfer to the glycol. Then,

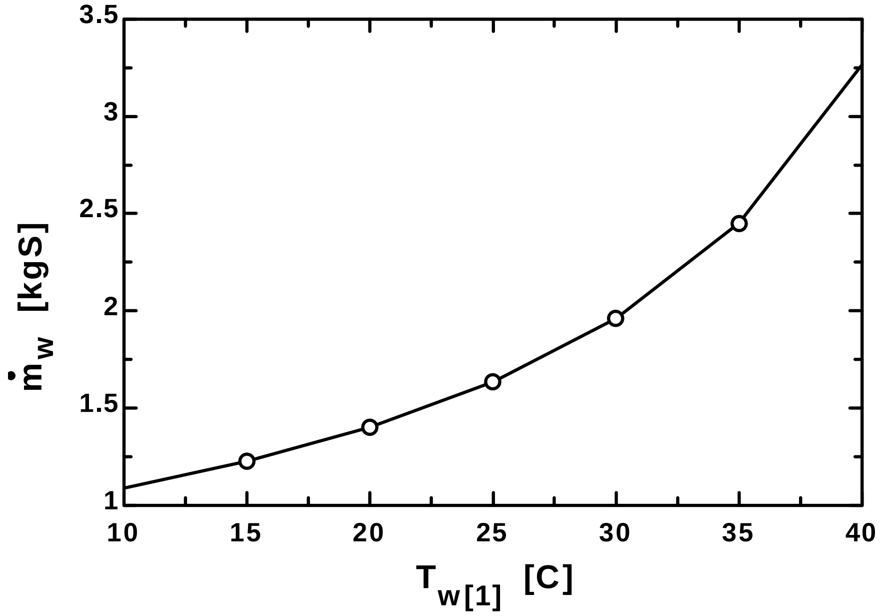
$$\dot{Q} = [\dot{m}C_p(T_{out} - T_{in})]_{\text{water}} \longrightarrow \dot{m}_{\text{water}} = \frac{\dot{Q}}{C_p(T_{out} - T_{in})}$$

$$= \frac{204.8 \text{ kJ/s}}{(4.18 \text{ kJ/kg.°C})(55°\text{C} - 20°\text{C})} = \mathbf{1.4\ kg/s}$$

5-119 Problem 5-118 is reconsidered. The effect of the inlet temperature of cooling water on the mass flow rate of water as the inlet temperature varies from 10°C to 40°C at constant exit temperature) is to be investigated. The mass flow rate of water is to be plotted against the inlet temperature.

```
"Input Data"
{T_w[1]=20"[C]"}
T_w[2]=55"[C]"  "w: water"
m_dot_eg=2"[kg/s]" "eg: ethylene glycol"
T_eg[1]=80"[C]"
T_eg[2]=40"[C]"
C_p_w=4.18"[kJ/kg-K]"
C_p_eg=2.56"[kJ/kg-K]"

"Conservation of mass for the water: m_dot_w_in=m_dot_w_out=m_dot_w"
"Conservation of mass for the ethylene glycol: m_dot_eg_in=m_dot_eg_out=m_dot_eg"

"Conservation of Energy for steady-flow: neglect changes in KE and PE in each mass steam"
"We assume no heat transfer and no work occur across the control surface."
E_dot_in - E_dot_out = DELTAE_dot_cv
DELTAE_dot_cv=0  "[kW]"     "Steady-flow requirement"
E_dot_in=m_dot_w*h_w[1] + m_dot_eg*h_eg[1] "[kW]"
E_dot_out=m_dot_w*h_w[2] + m_dot_eg*h_eg[2] "[kW]"

Q_exchanged =m_dot_eg*h_eg[1] - m_dot_eg*h_eg[2] "[kW]"

"Property data are given by:"
h_w[1] =C_p_w*T_w[1] " liquid approximation applied for water and ethylene glycol"
h_w[2] =C_p_w*T_w[2]

h_eg[1] =C_p_eg*T_eg[1]
h_eg[2] =C_p_eg*T_eg[2]
```

m_w [kg/s]	$T_{w,1}$ [C]
1.089	10
1.225	15
1.4	20
1.633	25
1.96	30
2.45	35
3.266	40



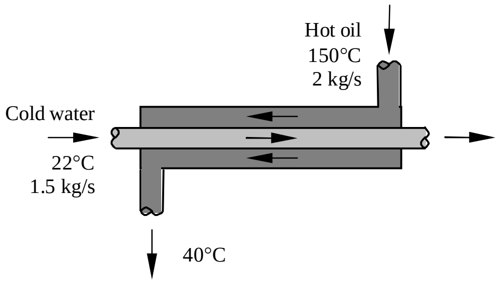
3.5
3
2.5
2
1.5
1
10
15
20
25
30
35
40
ṁw [kgS]
Tw[1] [C]

5-120 Oil is to be cooled by water in a thin-walled heat exchanger. The rate of heat transfer in the heat exchanger and the exit temperature of water is to be determined. √

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The specific heats of water and oil are given to be 4.18 and 2.20 kJ/kg.°C, respectively.

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



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

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

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

$$\dot{Q}_{out} = \dot{m}C_p(T_1 - T_2)$$

Then the rate of heat transfer from the oil becomes

$$\dot{Q} = [\dot{m}C_p(T_{in} - T_{out})]_{\text{oil}} = (2 \text{ kg/s})(2.2 \text{ kJ/kg.°C})(150°\text{C} - 40°\text{C}) = \mathbf{484\ kW}$$

Noting that the heat lost by the oil is gained by the water, the outlet temperature of the water is determined from

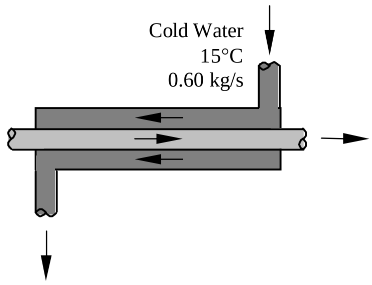
$$\dot{Q} = [\dot{m}C_p(T_{out} - T_{in})]_{\text{water}} \longrightarrow T_{\text{out}} = T_{\text{in}} + \frac{\dot{Q}}{\dot{m}_{\text{water}}C_p} = 22°\text{C} + \frac{484 \text{ kJ/s}}{(1.5 \text{ kg/s})(4.18 \text{ kJ/kg.°C})} = \mathbf{99.2°C}$$

5-121 Cold water is heated by hot water in a heat exchanger. The rate of heat transfer and the exit temperature of hot water are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The specific heats of cold and hot water are given to be 4.18 and 4.19 kJ/kg.°C, respectively.

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



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

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

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

$$\dot{Q}_{in} = \dot{m}C_p(T_2 - T_1)$$

Then the rate of heat transfer to the cold water in this heat exchanger becomes

$$\dot{Q} = [\dot{m}C_p(T_{out} - T_{in})]_{\text{cold water}} = (0.60\text{ kg/s})(4.18\text{ kJ/kg.°C})(45°\text{C} - 15°\text{C}) = \mathbf{75.24\ kW}$$

Noting that heat gain by the cold water is equal to the heat loss by the hot water, the outlet temperature of the hot water is determined to be

$$\dot{Q} = [\dot{m}C_p(T_{in} - T_{out})]_{\text{hot water}} \longrightarrow T_{out} = T_{in} - \frac{\dot{Q}}{\dot{m}C_p}$$

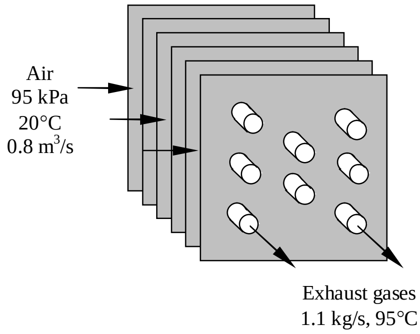
$$= 100°\text{C} - \frac{75.24\text{ kW}}{(3\text{ kg/s})(4.19\text{ kJ/kg.°C})} = \mathbf{94.0°C}$$

5-122 Air is preheated by hot exhaust gases in a cross-flow heat exchanger. The rate of heat transfer and the outlet temperature of the air are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The specific heats of air and combustion gases are given to be 1.005 and 1.10 kJ/kg.°C, respectively.

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



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

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

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

$$\dot{Q}_{out} = \dot{m}C_p(T_1 - T_2)$$

Then the rate of heat transfer from the exhaust gases becomes

$$\dot{Q} = [\dot{m}C_p(T_{in} - T_{out})]_{\text{gas.}} = (1.1 \text{ kg/s})(1.1 \text{ kJ/kg.°C})(180\text{°C} - 95\text{°C}) = \mathbf{102.85\ kW}$$

The mass flow rate of air is

$$\dot{m} = \frac{P\dot{V}}{RT} = \frac{(95 \text{ kPa})(0.8 \text{ m}^3/\text{s})}{(0.287 \text{ kPa.m}^3/\text{kg.K}) \times 293 \text{ K}} = 0.904 \text{ kg/s}$$

Noting that heat loss by the exhaust gases is equal to the heat gain by the air, the outlet temperature of the air becomes

$$\dot{Q} = \dot{m}C_p(T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{\dot{m}C_p}$$

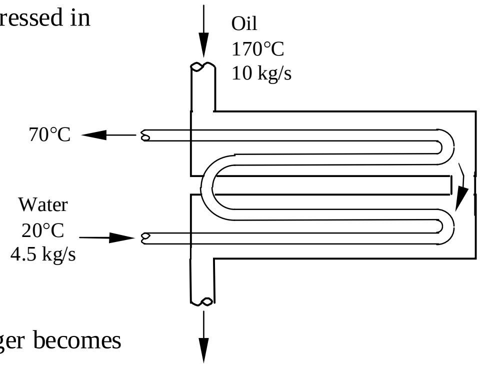
$$= 20\text{°C} + \frac{102.85 \text{ kW}}{(0.904 \text{ kg/s})(1.005 \text{ kJ/kg.°C})} = \mathbf{133.2°C}$$

5-123 Water is heated by hot oil in a heat exchanger. The rate of heat transfer in the heat exchanger and the outlet temperature of oil are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The specific heats of water and oil are given to be 4.18 and 2.3 kJ/kg.°C, respectively.

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



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

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

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

$$\dot{Q}_{in} = \dot{m}C_p(T_2 - T_1)$$

Then the rate of heat transfer to the cold water in this heat exchanger becomes

$$\dot{Q} = [\dot{m}C_p(T_{out} - T_{in})]_{\text{water}} = (4.5 \text{ kg/s})(4.18 \text{ kJ/kg.°C})(70\text{°C} - 20\text{°C}) = \mathbf{940.5 \text{ kW}}$$

Noting that heat gain by the water is equal to the heat loss by the oil, the outlet temperature of the hot water is determined from

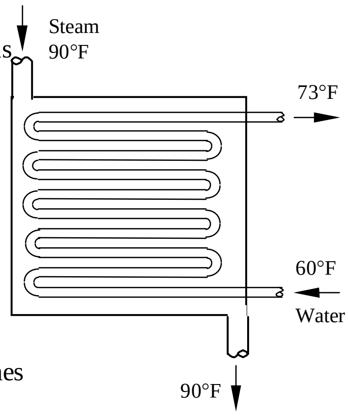
$$\dot{Q} = [\dot{m}C_p(T_{in} - T_{out})]_{\text{oil}} \longrightarrow T_{out} = T_{in} - \frac{\dot{Q}}{\dot{m}C_p} = 170\text{°C} - \frac{940.5 \text{ kW}}{(10 \text{ kg/s})(2.3 \text{ kJ/kg.°C})} = \mathbf{129.1°C}$$

5-124E Steam is condensed by cooling water in a condenser. The rate of heat transfer in the heat exchanger and the rate of condensation of steam are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The specific heat of water is 1.0 Btu/lbm.°F. The enthalpy of vaporization of water at 90°F is 1042.7 Btu/lbm (Table A-4E).

Analysis We take the tube-side of the heat exchanger where cold water is flowing as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as



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

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

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

$$\dot{Q}_{in} = \dot{m}C_p(T_2 - T_1)$$

Then the rate of heat transfer to the cold water in this heat exchanger becomes

$$\dot{Q} = [\dot{m}C_p(T_{out} - T_{in})]_{\text{water}} = (115.3 \text{ lbm/s})(1.0 \text{ Btu/lbm.°F})(73\text{°F} - 60\text{°F}) = \mathbf{1499\ Btu/s}$$

Noting that heat gain by the water is equal to the heat loss by the condensing steam, the rate of condensation of the steam in the heat exchanger is determined from

$$\dot{Q} = (\dot{m}h_{fg})_{steam} \longrightarrow \dot{m}_{steam} = \frac{\dot{Q}}{h_{fg}} = \frac{1499 \text{ Btu/s}}{1042.7 \text{ Btu/lbm}} = \mathbf{1.44\ lbm/s}$$

Pipe and duct Flow

5-125 A desktop computer is to be cooled safely by a fan in hot environments and high elevations. The air flow rate of the fan and the diameter of the casing are to be determined.

Assumptions **1** Steady operation under worst conditions is considered. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energy changes are negligible.

Properties The specific heat of air at the average temperature of T_{ave} = (45+60)/2 = 52.5°C = 325.5 K is C_p = 1.0065 kJ/kg.°C (Table A-2b)

Analysis The fan selected must be able to meet the cooling requirements of the computer at worst conditions. Therefore, we assume air to enter the computer at 66.63 kPa and 45°C, and leave at 60°C.

We take the air space in the computer as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

$$\dot{Q}_{in} = \dot{m}C_p(T_2 - T_1)$$

Then the required mass flow rate of air to absorb heat at a rate of 60 W is determined to be

$$\dot{Q} = \dot{m}C_p(T_{out} - T_{in}) \to \dot{m} = \frac{\dot{Q}}{C_p(T_{out} - T_{in})} = \frac{60\text{ W}}{(1006.5\text{ J/kg.°C})(60 - 45)\text{°C}}$$
$$= 0.00397\text{ kg/s} = 0.238\text{ kg/min}$$

The density of air entering the fan at the exit and its volume flow rate are

$$\rho = \frac{P}{RT} = \frac{66.63\text{ kPa}}{(0.287\text{ kPa.m}^3\text{/kg.K})(60 + 273)\text{K}} = 0.6972\text{ kg/m}^3$$

$$\dot{V} = \frac{\dot{m}}{\rho} = \frac{0.238\text{ kg/min}}{0.6972\text{ kg/m}^3} = \mathbf{0.341\ m^3/min}$$

For an average exit velocity of 110 m/min, the diameter of the casing of the fan is determined from

$$\dot{V} = A_c\mathbf{V} = \frac{\pi D^2}{4}\mathbf{V} \to D = \sqrt{\frac{4\dot{V}}{\pi\mathbf{V}}} = \sqrt{\frac{(4)(0.341\text{ m}^3\text{/min})}{\pi(110\text{ m/min})}} = 0.063\text{ m} = \mathbf{6.3\ cm}$$

5-126 A desktop computer is to be cooled safely by a fan in hot environments and high elevations. The air flow rate of the fan and the diameter of the casing are to be determined.

Assumptions **1** Steady operation under worst conditions is considered. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energy changes are negligible.

Properties The specific heat of air at the average temperature of T_{ave} = (45+60)/2 = 52.5°C is C_p = 1.0065 kJ/kg.°C (Table A-3).

Analysis The fan selected must be able to meet the cooling requirements of the computer at worst conditions. Therefore, we assume air to enter the computer at 66.63 kPa and 45°C, and leave at 60°C.

We take the air space in the computer as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

$$\dot{Q}_{in} = \dot{m}C_p(T_2 - T_1)$$

Then the required mass flow rate of air to absorb heat at a rate of 100 W is determined to be

$$\dot{Q} = \dot{m}C_p(T_{out} - T_{in}) \longrightarrow \dot{m} = \frac{\dot{Q}}{C_p(T_{out} - T_{in})} = \frac{100\ \text{W}}{(1006.5\ \text{J/kg.°C})(60 - 45)\text{°C}}$$
$$= 0.006624\ \text{kg/s} = 0.397\ \text{kg/min}$$

The density of air entering the fan at the exit and its volume flow rate are

$$\rho = \frac{P}{RT} = \frac{66.63\ \text{kPa}}{(0.287\ \text{kPa.m}^3/\text{kg.K})(60 + 273)\text{K}} = 0.6972\ \text{kg/m}^3$$

$$\dot{V} = \frac{\dot{m}}{\rho} = \frac{0.397\ \text{kg/min}}{0.6972\ \text{kg/m}^3} = \mathbf{0.57\ m^3/min}$$

For an average exit velocity of 110 m/min, the diameter of the casing of the fan is determined from

$$\dot{V} = A_c \mathbf{V} = \frac{\pi D^2}{4}\mathbf{V} \longrightarrow D = \sqrt{\frac{4\dot{V}}{\pi \mathbf{V}}} = \sqrt{\frac{(4)(0.57\ \text{m}^3/\text{min})}{\pi(110\ \text{m/min})}} = 0.081\ \text{m} = \mathbf{8.1\ cm}$$

5-127E Electronic devices mounted on a cold plate are cooled by water. The amount of heat generated by the electronic devices is to be determined.

Assumptions **1** Steady operating conditions exist. **2** About 15 percent of the heat generated is dissipated from the components to the surroundings by convection and radiation. **3** Kinetic and potential energy changes are negligible.

Properties The properties of water at room temperature are ρ = 62.1 lbm/ft^3 and C_p = 1.00 Btu/lbm.°F (Table A-3).

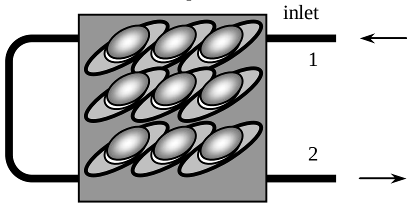
Analysis We take the tubes of the cold plate to be the system, which is a control volume.
The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

$$\dot{Q}_{in} = \dot{m}C_p(T_2 - T_1)$$



Then mass flow rate of water and the rate of heat removal by the water are determined to be

$$\dot{m} = \rho A\mathbf{V} = \rho \frac{\pi D^2}{4}\mathbf{V} = (62.1\,\text{lbm/ft}^3)\frac{\pi(0.25/12\,\text{ft})^2}{4}(60\,\text{ft/min}) = 1.270\,\text{lbm/min} = 76.2\,\text{lbm/h}$$

$$\dot{Q} = \dot{m}C_p(T_{out} - T_{in}) = (76.2\,\text{lbm/h})(1.00\,\text{Btu/lbm.°F})(105 - 95)\text{°F} = 762\,\text{Btu/h}$$

which is 85 percent of the heat generated by the electronic devices. Then the total amount of heat generated by the electronic devices becomes

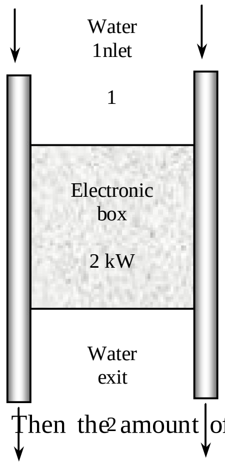
$$\dot{Q} = \frac{762\text{ Btu/h}}{0.85} = \mathbf{896\ Btu/h = 263\ W}$$

5-128 A sealed electronic box is to be cooled by tap water flowing through channels on two of its sides. The mass flow rate of water and the amount of water used per year are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Entire heat generated is dissipated by water. **3** Water is an incompressible substance with constant specific heats at room temperature. **4** Kinetic and potential energy changes are negligible.

Properties The specific heat of water at room temperature is C_p = 4.18 kJ/kg.°C (Table A-3).

Analysis We take the water channels on the sides 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



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

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

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

$$\dot{Q}_{in} = \dot{m}C_p(T_2 - T_1)$$

Then the mass flow rate of tap water flowing through the electronic box becomes

$$\dot{Q} = \dot{m}C_p\Delta T \longrightarrow \dot{m} = \frac{\dot{Q}}{C_p\Delta T} = \frac{2 \text{ kJ/s}}{(4.18 \text{ kJ/kg.°C})(4\text{°C})} = \mathbf{0.1196 \text{ kg/s}}$$

Therefore, 0.11962 kg of water is needed per second to cool this electronic box. Then the amount of cooling water used per year becomes

$$m = \dot{m}\Delta t = (0.1196 \text{ kg/s})(365 \text{ days/yr} \times 24 \text{ h/day} \times 3600 \text{ s/h})$$

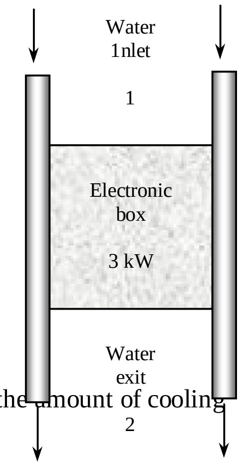
$$= 3{,}772{,}000 \text{ kg/yr} = \mathbf{3{,}772 \text{ tons/yr}}$$

5-129 A sealed electronic box is to be cooled by tap water flowing through channels on two of its sides. The mass flow rate of water and the amount of water used per year are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Entire heat generated is dissipated by water. **3** Water is an incompressible substance with constant specific heats at room temperature. **4** Kinetic and potential energy changes are negligible

Properties The specific heat of water at room temperature is C_p = 4.18 kJ/kg.°C (Table A-3).

Analysis We take the water channels on the sides 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



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

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

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

$$\dot{Q}_{in} = \dot{m}C_p(T_2 - T_1)$$

Then the mass flow rate of tap water flowing through the electronic box becomes

$$\dot{Q} = \dot{m}C_p \Delta T \longrightarrow \dot{m} = \frac{\dot{Q}}{C_p \Delta T} = \frac{3 \text{ kJ/s}}{(4.18 \text{ kJ/kg.°C})(4\text{°C})} = \mathbf{0.1794 \text{ kg/s}}$$

Therefore, 0.1794 kg of water is needed per second to cool this electronic box. Then the amount of cooling water used per year becomes

$$m = \dot{m}\Delta t = (0.1794 \text{ kg/s})(365 \text{ days/yr} \times 24 \text{ h/day} \times 3600 \text{ s/h})$$
$$= 5{,}658{,}400 \text{ kg/yr} = \mathbf{5{,}658 \text{ tons/yr}}$$

5-130 A long roll of large 1-Mn manganese steel plate is to be quenched in an oil bath at a specified rate. The rate at which heat needs to be removed from the oil to keep its temperature constant is to be determined.

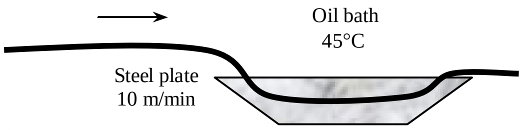
Assumptions **1** Steady operating conditions exist. **2** The thermal properties of the roll are constant. **3** Kinetic and potential energy changes are negligible

Properties The properties of the steel plate are given to be ρ = 7854 kg/m^3 and C_p = 0.454 kJ/kg.°C.

Analysis The mass flow rate of the sheet metal through the oil bath is

$$\dot{m} = \rho\dot{V} = \rho wtV = (7854\ \text{kg/m}^3)(2\ \text{m})(0.005\ \text{m})(10\ \text{m/min}) = 785.4\ \text{kg/min}$$

We take the volume occupied by the sheet metal in the oil bath to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed I n the rate form as



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

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

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

$$\dot{Q}_{out} = \dot{m}C_p(T_1 - T_2)$$

Then the rate of heat transfer from the sheet metal to the oil bath becomes

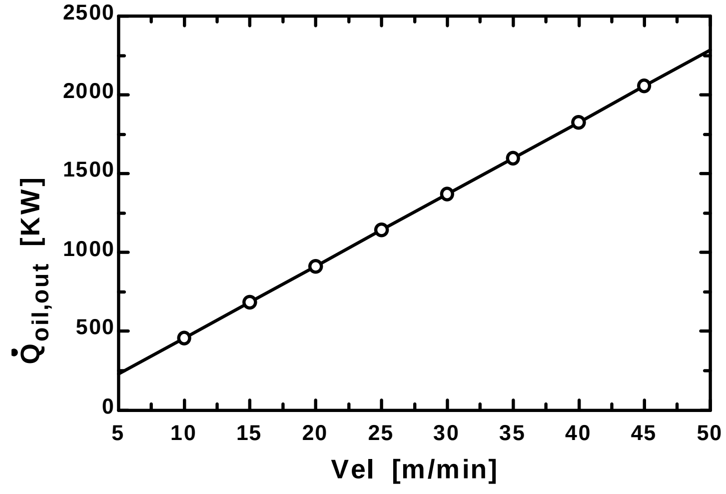
$$\begin{aligned}\dot{Q}_{out} &= \dot{m}C_p[T_{in} - T_{out}]_{\text{metal}}\\ &= (785.4\ \text{kg/min})(0.434\ \text{kJ/kg.°C})(820 - 51.1)\text{°C}\\ &= 262{,}090\ \text{kJ/min} = \mathbf{4368\ kW}\end{aligned}$$

This is the rate of heat transfer from the metal sheet to the oil, which is equal to the rate of heat removal from the oil since the oil temperature is maintained constant.

5-131 Problem 5-130 is reconsidered. The effect of the moving velocity of the steel plate on the rate of heat transfer from the oil bath as the velocity varies from 5 to 50 m/min is to be investigated. Tate of heat transfer is to be plotted against the plate velocity.

```
"Knowns"
Vel = 10 "[m/min]"
T_bath = 45"[C]"
T_1 = 820"[C]"
T_2 = 51.1 "[C]"
rho = 785 "[kg/m^3]"
C_P = 0.454 "[kJ/kg-C]"
width = 2"[m]"
thick = 0.5"[cm]"

"Analysis:
 The mass flow rate of the sheet metal through the oil bath is:"
Vol_dot = width*thick*convert(cm,m)*Vel/convert(min,s)"[m^3/s]"
m_dot = rho*Vol_dot"[kg/s]"

"We take the volume occupied by the sheet metal in the oil bath to be the system,
which is a control volume. The energy balance for this steady-flow system--the metal can be
expressed  in the rate form as:"

E_dot_metal_in = E_dot_metal_out
E_dot_metal_in=m_dot*h_1
E_dot_metal_out=m_dot*h_2+Q_dot_metal_out

h_1 = C_P*T_1"[kJ/kg]"
h_2 = C_P*T_2"[kJ/kg]"

Q_dot_oil_out = Q_dot_metal_out"[KW]"
```

Q_{oilout} [kW]	Vel [m/min]
228.4	5
456.7	10
685.1	15
913.4	20
1142	25
1370	30
1598	35
1827	40
2055	45
2284	50



5-132 [*Also solved by EES on enclosed CD***]** The components of an electronic device located in a horizontal duct of rectangular cross section are cooled by forced air. The heat transfer from the outer surfaces of the duct is to be determined.

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

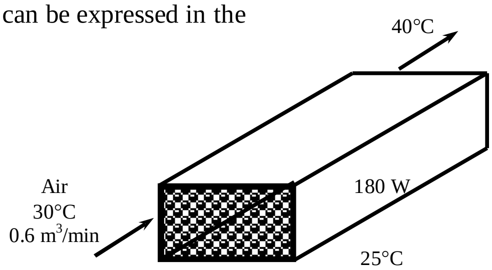
Properties The gas constant of air is R = 0.287 kJ/kg.°C (Table A-1). The specific heat of air at room temperature is C_p = 1.005 kJ/kg.°C (Table A-2).

Analysis The density of air entering the duct and the mass flow rate are

$$\rho = \frac{P}{RT} = \frac{101.325 \text{ kPa}}{(0.287 \text{ kPa.m}^3/\text{kg.K})(30+273)\text{K}} = 1.165 \text{ kg/m}^3$$

$$\dot{m} = \rho\dot{V} = (1.165 \text{ kg/m}^3)(0.6 \text{ m}^3/\text{min}) = \mathbf{0.700 \text{ kg/min}}$$

We take the channel, excluding the electronic components, 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



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

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

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

$$\dot{Q}_{in} = \dot{m}C_p(T_2 - T_1)$$

Then the rate of heat transfer to the air passing through the duct becomes

$$\dot{Q}_{air} = [\dot{m}C_p(T_{out} - T_{in})]_{\text{air}} = (0.700/60 \text{ kg/s})(1.005 \text{ kJ/kg.°C})(40-30)\text{°C} = 0.117 \text{ kW} = 117 \text{ W}$$

The rest of the 180 W heat generated must be dissipated through the outer surfaces of the duct by natural convection and radiation,

$$\dot{Q}_{\text{external}} = \dot{Q}_{\text{total}} - \dot{Q}_{\text{internal}} = 180 - 117 = \mathbf{63 \text{ W}}$$

5-133 The components of an electronic device located in a horizontal duct of circular cross section is cooled by forced air. The heat transfer from the outer surfaces of the duct is to be determined.

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

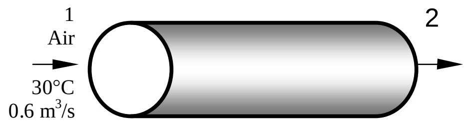
Properties The gas constant of air is R = 0.287 kJ/kg.°C (Table A-1). The specific heat of air at room temperature is C_p = 1.005 kJ/kg.°C (Table A-2).

Analysis The density of air entering the duct and the mass flow rate are

$$\rho = \frac{P}{RT} = \frac{101.325 \text{ kPa}}{(0.287 \text{ kPa.m}^3/\text{kg.K})(30+273)\text{K}} = 1.165 \text{ kg/m}^3$$

$$\dot{m} = \rho\dot{V} = (1.165 \text{ kg/m}^3)(0.6 \text{ m}^3/\text{min}) = 0.700 \text{ kg/min}$$

We take the channel, excluding the electronic components, 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



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

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

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

$$\dot{Q}_{in} = \dot{m}C_p(T_2 - T_1)$$

Then the rate of heat transfer to the air passing through the duct becomes

$$\dot{Q}_{air} = [\dot{m}C_p(T_{out} - T_{in})]_{\text{air}} = (0.700/60 \text{ kg/s})(1.005 \text{ kJ/kg.°C})(40-30)\text{°C} = 0.117 \text{ kW} = 117 \text{ W}$$

The rest of the 180 W heat generated must be dissipated through the outer surfaces of the duct by natural convection and radiation,

$$\dot{Q}_{\text{external}} = \dot{Q}_{\text{total}} - \dot{Q}_{\text{internal}} = 180 - 117 = \mathbf{63 \text{ W}}$$

5-134E Water is heated in a parabolic solar collector. The required length of parabolic collector is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat loss from the tube is negligible so that the entire solar energy incident on the tube is transferred to the water. **3** Kinetic and potential energy changes are negligible

Properties The specific heat of water at room temperature is C_p = 1.00 Btu/lbm.°F (Table A-2E).

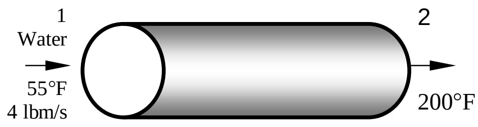
Analysis We take the thin aluminum tube to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

$$\dot{Q}_{in} = \dot{m}_{\text{water}} C_p (T_2 - T_1)$$



Then the total rate of heat transfer to the water flowing through the tube becomes

$$\dot{Q}_{\text{total}} = \dot{m}C_p (T_e - T_i) = (4 \text{ lbm/s})(1.00 \text{ Btu/lbm.°F})(200-55)\text{°F} = 580 \text{ Btu/s} = 2{,}088{,}000 \text{ Btu/h}$$

The length of the tube required is

$$L = \frac{\dot{Q}_{\text{total}}}{\dot{Q}} = \frac{2{,}088{,}000 \text{ Btu/h}}{350 \text{ Btu/h.ft}} = \mathbf{5966 \text{ ft}}$$

5-135 Air enters a hollow-core printed circuit board. The exit temperature of the air is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant specific heats at room temperature. **3** The local atmospheric pressure is 1 atm. **4** Kinetic and potential energy changes are negligible.

Properties The gas constant of air is R = 0.287 kJ/kg.°C (Table A-1). The specific heat of air at room temperature is C_p = 1.005 kJ/kg.°C (Table A-2).

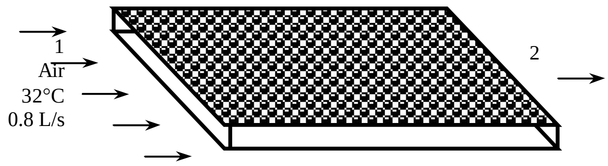
Analysis The density of air entering the duct and the mass flow rate are

$$\rho = \frac{P}{RT} = \frac{101.325 \text{ kPa}}{(0.287 \text{ kPa.m}^3/\text{kg.K})(32+273)\text{K}} = 1.16 \text{ kg/m}^3$$

$$\dot{m} = \rho\dot{V} = (1.16 \text{ kg/m}^3)(0.0008 \text{ m}^3/\text{s}) = 0.000928 \text{ kg/s}$$

We take the hollow core 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

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



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

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

$$\dot{Q}_{in} = \dot{m}C_p(T_2 - T_1)$$

Then the exit temperature of air leaving the hollow core becomes

$$\dot{Q}_{in} = \dot{m}C_p(T_2 - T_1) \rightarrow T_2 = T_1 + \frac{\dot{Q}_{in}}{\dot{m}C_p} = 32\,°\text{C} + \frac{20 \text{ J/s}}{(0.000928 \text{ kg/s})(1005 \text{ J/kg.°C})} = \mathbf{53.4°C}$$

5-136 A computer is cooled by a fan blowing air through the case of the computer. The required flow rate of the air and the fraction of the temperature rise of air that is due to heat generated by the fan are to be determined.

Assumptions **1** Steady flow conditions exist. **2** Air is an ideal gas with constant specific heats. **3** The pressure of air is 1 atm. **4** Kinetic and potential energy changes are negligible

Properties The specific heat of air at room temperature is C_p = 1.005 kJ/kg.°C (Table A-2).

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

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

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

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

$$\dot{Q}_{in} + \dot{W}_{in} = \dot{m}C_p(T_2 - T_1)$$

Noting that the fan power is 25 W and the 8 PCBs transfer a total of 80 W of heat to air, the mass flow rate of air is determined to be

$$\dot{Q}_{in} + \dot{W}_{in} = \dot{m}C_p(T_e - T_i) \rightarrow \dot{m} = \frac{\dot{Q}_{in} + \dot{W}_{in}}{C_p(T_e - T_i)} = \frac{25 + (8 \times 10)\text{ W}}{(1005\text{ J/kg.°C})(10\text{°C})} = \mathbf{0.0104\ kg/s}$$

(*b*) The fraction of temperature rise of air that is due to the heat generated by the fan and its motor can be determined from

$$\dot{Q} = \dot{m}C_p\Delta T \rightarrow \Delta T = \frac{\dot{Q}}{\dot{m}C_p} = \frac{25\text{ W}}{(0.0104\text{ kg/s})(1005\text{ J/kg.°C})} = 2.4\text{°C}$$

$$f = \frac{2.4\text{°C}}{10\text{°C}} = 0.24 = \mathbf{24\%}$$

5-137 Hot water enters a pipe whose outer surface is exposed to cold air in a basement. The rate of heat loss from the water is to be determined.

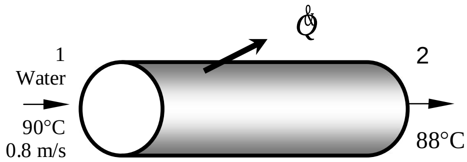
Assumptions **1** Steady flow conditions exist. **2** Water is an incompressible substance with constant specific heats. **3** The changes in kinetic and potential energies are negligible.

Properties The properties of water at the average temperature of (90+88)/2 = 89°C are $\rho = 965\,\text{kg/m}^3$ and C_p = 4.21 kJ/kg.°C (Table A-3).

Analysis The mass flow rate of water is

$$\dot{m} = \rho A_c V = (965\,\text{kg/m}^3)\left(\frac{\pi(0.04\,\text{m})^2}{4}\right)(0.8\text{ m/s}) = 0.970\,\text{kg/s}$$

We take the section of the pipe in the basement 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



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

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

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

$$\dot{Q}_{out} = \dot{m}C_p(T_1 - T_2)$$

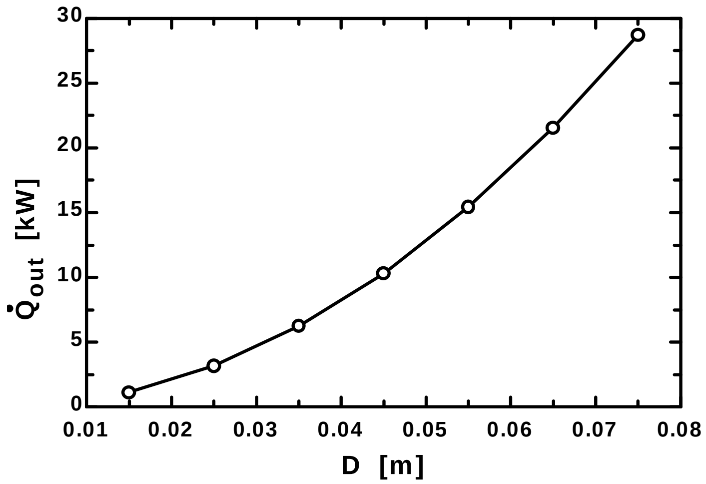
Then the rate of heat transfer from the hot water to the surrounding air becomes

$$\begin{aligned}\dot{Q}_{out} &= \dot{m}C_p[T_{in} - T_{out}]_{\text{water}}\\ &= (0.970\text{ kg/s})(4.21\text{ kJ/kg.°C})(90-88)\text{°C}\\ &= \mathbf{8.17\ kW}\end{aligned}$$

5-138 Problem 5-137 is reconsidered. The effect of the inner pipe diameter on the rate of heat loss as the pipe diameter varies from 1.5 cm to 7.5 cm is to be investigated. The rate of heat loss is to be plotted against the diameter.

```
"Knowns:"
{D = 0.04 "[m]"}
rho = 965 "[kg/m^3]"
Vel = 0.8"[m/s]"
T_1 = 90"[C]"
T_2 = 88"[C]"
C_P = 4.21"[kJ/kg-C]"

"Analysis:"
 "The mass flow rate of water is:"
Area = pi*D^2/4"[m^2]"
m_dot = rho*Area*Vel"[kg/s]"

"We take the section of the pipe in the basement 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"

E_dot_in - E_dot_out = DELTAE_dot_sys

DELTAE_dot_sys = 0"[kW]" "Steady-flow assumption"

E_dot_in = m_dot*h_in"[kW]"
E_dot_out =Q_dot_out+m_dot*h_out"[kW]"

h_in = C_P * T_1"[kJ/kg]"
h_out = C_P * T_2"[kJ/kg]"
```

D [m]	Q_{out} [kW]
0.015	1.149
0.025	3.191
0.035	6.254
0.045	10.34
0.055	15.44
0.065	21.57
0.075	28.72

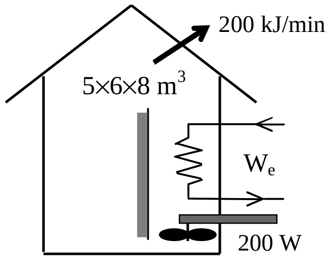


5-139 A room is to be heated by an electric resistance heater placed in a duct in the room. The power rating of the electric heater and the temperature rise of air as it passes through the heater are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible. **4** The heating duct is adiabatic, and thus heat transfer through it is negligible. **5** No air leaks in and out of the room.

Properties The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1). The specific heats of air at room temperature are C_p = 1.005 and C_v = 0.718 kJ/kg·K (Table A-2).

Analysis (*a*) The total mass of air in the room is



$$V = 5\times 6\times 8\,\text{m}^3 = 240\,\text{m}^3$$

$$m = \frac{P_1 V}{RT_1} = \frac{(98\text{ kPa})(240\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(288\text{ K})} = 284.6\text{ kg}$$

We first take the *entire room* as our system, which is a closed system since no mass leaks in or out. The power rating of the electric heater is determined by applying the conservation of energy relation to this constant volume closed system:

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

$$W_{e,in} + W_{fan,in} - Q_{out} = \Delta U \quad (\text{since } \Delta\text{KE} = \Delta\text{PE} = 0)$$

$$\Delta t\left(\dot{W}_{e,in} + \dot{W}_{fan,in} - \dot{Q}_{out}\right) = mC_{v,ave}\left(T_2 - T_1\right)$$

Solving for the electrical work input gives

$$\begin{aligned}\dot{W}_{e,in} &= \dot{Q}_{out} - \dot{W}_{fan,in} + mC_v(T_2 - T_1)/\Delta t \\ &= (200/60\text{ kJ/s}) - (0.2\text{ kJ/s}) + (284.6\text{ kg})(0.718\text{kJ/kg}\cdot{}^\circ\text{C})(25-15)^\circ\text{C}/(15\times 60\text{s}) \\ &= \mathbf{5.40\ \ kW}\end{aligned}$$

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

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

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

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

$$\dot{W}_{e,in} + \dot{W}_{fan,in} = \dot{m}(h_2 - h_1) = \dot{m}C_p(T_2 - T_1)$$

Thus,

$$\Delta T = T_2 - T_1 = \frac{\dot{W}_{e,in} + \dot{W}_{fan,in}}{\dot{m}C_p} = \frac{(5.40 + 0.2)\text{kJ/s}}{(50/60\text{kg/s})(1.005\text{kJ/kg}\cdot\text{K})} = \mathbf{6.7^\circ C}$$

5-140 A house is heated by an electric resistance heater placed in a duct. The power rating of the electric heater is to be determined.

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

Properties The constant pressure specific heat of air at room temperature is C_p = 1.005 kJ/kg·K (Table A-2)

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

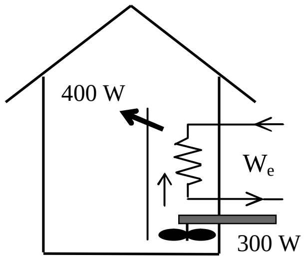
$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

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

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

$$\dot{W}_{e,in} + \dot{W}_{fan,in} = \dot{Q}_{out} + \dot{m}(h_2 - h_1)$$

$$= \dot{Q}_{out} + \dot{m}C_p(T_2 - T_1)$$



Substituting, the power rating of the heating element is determined to be

$$\dot{W}_{e,in} = \dot{Q}_{out} + \dot{m}C_p\Delta T - \dot{W}_{fan,in}$$

$$= (0.4 \text{ kJ/s}) + (0.6 \text{ kg/s})(1.005 \text{ kJ/kg}\cdot{}^\circ\text{C})(5^\circ\text{C}) - 0.3 \text{ kW}$$

$$= \mathbf{3.12\ \ kW}$$

5-141 A hair dryer consumes 1200 W of electric power when running. The inlet volume flow rate and the exit velocity of air are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible. **4** The power consumed by the fan and the heat losses are negligible.

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)

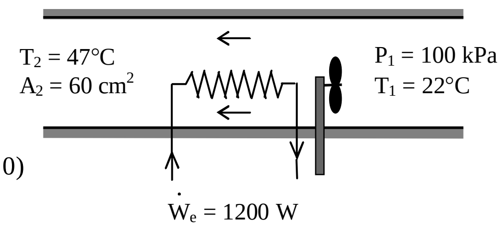
Analysis We take the *hair dryer* as the system, which is a control volume since mass crosses the boundary. There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

$$\dot{W}_{e,in} = \dot{m}(h_2 - h_1) = \dot{m}C_p(T_2 - T_1)$$



Substituting, the mass and volume flow rates of air are determined to be

$$\dot{m} = \frac{\dot{W}_{e,in}}{C_p(T_2 - T_1)} = \frac{1.2 \text{ kJ/s}}{(1.005 \text{kJ/kg}\cdot{}^\circ\text{C})(47 - 22)^\circ\text{C}} = 0.04776 \text{kg/s}$$

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(295 \text{ K})}{(100 \text{ kPa})} = 0.8467 \text{ m}^3/\text{kg}$$

$$\dot{V}_1 = \dot{m}v_1 = (0.04776 \text{ kg/s})(0.8467 \text{ m}^3/\text{kg}) = \mathbf{0.0404 \, m^3/s}$$

(*b*) The exit velocity of air is determined from the mass balance $\dot{m}_1 = \dot{m}_2 = \dot{m}$ to be

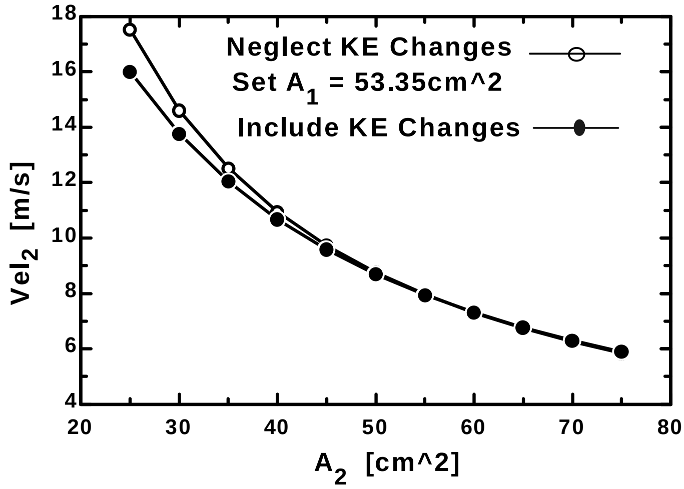
$$v_2 = \frac{RT_2}{P_2} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(320\text{K})}{(100 \text{ kPa})} = 0.9184 \text{ m}^3/\text{kg}$$

$$\dot{m} = \frac{1}{v_2}A_2\mathbf{V}_2 \longrightarrow \mathbf{V}_2 = \frac{\dot{m}v_2}{A_2} = \frac{(0.04776 \text{ kg/s})(0.9184 \text{ m}^3/\text{kg})}{60\times10^{-4} \text{ m}^2} = \mathbf{7.31 \ m/s}$$

5-142 Problem 5-141 is reconsidered. The effect of the exit cross-sectional area of the hair drier on the exit velocity as the exit area varies from 25 cm^2 to 75 cm^2 is to be investigated. The exit velocity is to be plotted against the exit cross-sectional area,

```
"Knowns:"
R=0.287 "[kPa-m^3/kg-K]"
P= 100"[kPa]"
T_1 = 22"[C]"
T_2= 47"[C]"
{A_2 = 60"[cm^2]"}
A_1 = 53.35"[cm^2]"
W_dot_ele=1200"[W]"

"Analysis:
 We take the hair dryer as the system, which is a control volume since mass
crosses the boundary. There is only one inlet and one exit.  Thus, the
energy balance for this steady-flow system can be expressed in the rate form
as:"
E_dot_in = E_dot_out
E_dot_in = W_dot_ele*convert(W,kW) +
m_dot_1*h_1+Vel_1^2/2*convert(m^2/s^2,kJ/kg)"[kW]"
E_dot_out = m_dot_2*h_2+Vel_2^2/2*convert(m^2/s^2,kJ/kg)"[kW]"
h_2  = enthalpy(air, T = T_2)"[kJ/kg]"
h_1= enthalpy(air, T = T_1)"[kJ/kg]"

"The volume flow rates of air are determined to be:"
V_dot_1 = m_dot_1*v_1"[m^3/s]"
P*v_1=R*(T_1+273)
V_dot_2 = m_dot_2*v_2"[m^3/s]"
P*v_2=R*(T_2+273)"[m^3/s]"
m_dot_1 = m_dot_2
Vel_1=V_dot_1/(A_1*convert(cm^2,m^2))"[m/s]"

"(b) The exit velocity of air is determined from the mass balance  to be"
Vel_2=V_dot_2/(A_2*convert(cm^2,m^2))"[m/s]"
```

A_2 [cm^2]	Vel_2 [m/s]
25	16
30	13.75
35	12.03
40	10.68
45	9.583
50	8.688
55	7.941
60	7.31
65	6.77
70	6.303
75	5.896



5-143 The ducts of a heating system pass through an unheated area. The rate of heat loss from the air in the ducts is to be determined.

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

Properties The constant pressure specific heat of air at room temperature is C_p = 1.005 kJ/kg·K (Table A-2)

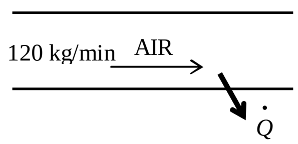
Analysis We take the *heating duct* as the system, which is a control volume since mass crosses the boundary. There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

$$\dot{Q}_{out} = \dot{m}(h_1 - h_2) = \dot{m}C_p(T_1 - T_2)$$



Substituting,

$$\dot{Q}_{out} = (120 \text{ kg/min})(1.005 \text{ kJ/kg}\cdot{}^{\circ}\text{C})(4^{\circ}\text{C}) = \mathbf{482\ kJ/min}$$

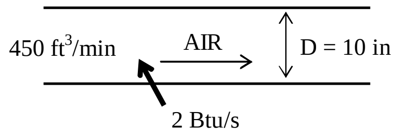
5-144E The ducts of an air-conditioning system pass through an unconditioned area. The inlet velocity and the exit temperature of air are to be determined.

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

Properties The gas constant of air is 0.3704 psia.ft^3/lbm.R (Table A-1E). The constant pressure specific heat of air at room temperature is C_p = 0.240 Btu/lbm.R (Table A-2E)

Analysis (*a*) The inlet velocity of air through the duct is

$$\mathbf{V}_1 = \frac{\dot{V}_1}{A_1} = \frac{\dot{V}_1}{\pi r^2} = \frac{450\text{ ft}^3/\text{min}}{\pi (5/12\text{ ft})^2} = \mathbf{825\ ft/min}$$



Then the mass flow rate of air becomes

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.3704\text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(510\text{R})}{(15\text{psia})} = 12.6\text{ft}^3/\text{lbm}$$

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{450\text{ft}^3/\text{min}}{12.6\text{ft}^3/\text{lbm}} = 35.7\text{lbm/min} = 0.595\text{lbm/s}$$

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

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

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

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

$$\dot{Q}_{in} = \dot{m}(h_2 - h_1) = \dot{m}C_p(T_2 - T_1)$$

Then the exit temperature of air becomes

$$T_2 = T_1 + \frac{\dot{Q}_{in}}{\dot{m}C_p} = 50^\circ\text{F} + \frac{2\text{ Btu/s}}{(0.595\text{ lbm/s})(0.24\text{ Btu/lbm}\cdot^\circ\text{F})} = \mathbf{64.0^\circ F}$$

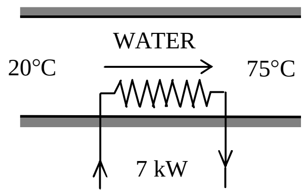
5-145 Water is heated by a 7-kW resistance heater as it flows through an insulated tube. The mass flow rate of water is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Water is an incompressible substance with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible. **4** The tube is adiabatic and thus heat losses are negligible.

Properties The specific heat of water at room temperature is C = 4.18 kJ/kg·°C (Table A-3).

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

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



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

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

$$\dot{W}_{e,in} = \dot{m}(h_2 - h_1) = \dot{m}[C(T_2 - T_1) + v\Delta P^{\nearrow 0}] = \dot{m}C(T_2 - T_1)$$

Substituting, the mass flow rates of water is determined to be

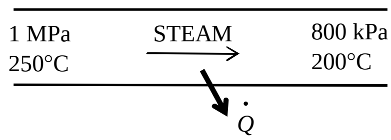
$$\dot{m} = \frac{\dot{W}_{e,in}}{C(T_2 - T_1)} = \frac{7 \text{ kJ/s}}{(4.184 \text{ kJ/kg}\cdot^\circ\text{C})(75 - 20)^\circ\text{C}} = \mathbf{0.0304 \text{ kg/s}}$$

5-146 Steam pipes pass through an unheated area, and the temperature of steam drops as a result of heat losses. The mass flow rate of steam and the rate of heat loss from 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. **4** There are no work interactions involved.

Properties From the steam tables (Table A-6),

$$\left.\begin{array}{l} P_1 = 1\text{MPa} \\ T_1 = 250^{\circ}\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.2327\text{m}^3/\text{kg} \\ h_1 = 2942.6\text{kJ/kg} \end{array}$$



and

$$\left.\begin{array}{l} P_2 = 800\text{kPa} \\ T_2 = 200^{\circ}\text{C} \end{array}\right\} h_2 = 2839.3\text{kJ/kg}$$

Analysis (*a*) The mass flow rate of steam is determined directly from

$$\dot{m} = \frac{1}{v_1} A_1 \mathbf{V}_1 = \frac{1}{0.2327\text{m}^3/\text{kg}} \left[\pi(0.06\text{m})^2\right](2\text{m/s}) = \mathbf{0.0972\ kg/s}$$

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

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

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

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

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

Substituting, the rate of heat loss is determined to be

$$\dot{Q}_{loss} = (0.0972\text{ kg/s})(2942.6 - 2839.3)\text{ kJ/kg} = \mathbf{10.04\ \ kJ/s}$$

Energy Balance for Charging and Discharging Processes

5-147 An evacuated bottle is surrounded by atmospheric air. A valve is opened, and air is allowed to fill the bottle. The amount of heat transfer through the wall of the bottle when thermal and mechanical equilibrium is established is to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Air is an ideal gas with variable specific heats. **3** Kinetic and potential energies are negligible. **4** There are no work interactions involved. **5** The direction of heat transfer is to the air in the bottle (will be verified).

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

Analysis We take the bottle as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{in} - m_{out} = \Delta m_{\text{system}} \quad \rightarrow \quad m_i = m_2 \quad (\text{since } m_{out} = m_{initial} = 0)$

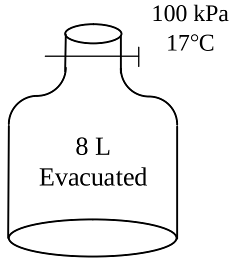
Energy balance:

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

$$Q_{in} + m_i h_i = m_2 u_2 \quad (\text{since } W \cong E_{out} = E_{initial} = ke \cong pe \cong 0)$$

Combining the two balances:

$$Q_{in} = m_2 (u_2 - h_i)$$



where

$$m_2 = \frac{P_2 V}{RT_2} = \frac{(100 \text{ kPa})(0.008 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(290 \text{ K})} = 0.0096 \text{ kg}$$

$$T_i = T_2 = 290 \text{ K} \xrightarrow{\text{Table A-17}} \begin{array}{l} h_i = 290.16 \text{ kJ/kg} \\ u_2 = 206.91 \text{ kJ/kg} \end{array}$$

Substituting,

$$Q_{\text{in}} = (0.0096 \text{ kg})(206.91 - 290.16) \text{ kJ/kg} = -0.8 \text{ kJ} \quad \rightarrow \quad Q_{\text{out}} = \mathbf{0.8 \text{ kJ}}$$

Discussion The negative sign for heat transfer indicates that the assumed direction is wrong. Therefore, we reverse the direction.

5-148 An insulated rigid tank is evacuated. A valve is opened, and air is allowed to fill the tank until mechanical equilibrium is established. The final temperature in the tank is to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energies are negligible. **4** There are no work interactions involved. **5** The device is adiabatic and thus heat transfer is negligible.

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

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{in} - m_{out} = \Delta m_{\text{system}} \quad \rightarrow \quad m_i = m_2 \quad (\text{since } m_{out} = m_{initial} = 0)$

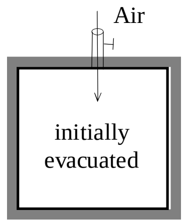
Energy balance:

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

$$m_i h_i = m_2 u_2 \quad (\text{since } Q \cong W \cong E_{out} = E_{initial} = ke \cong pe \cong 0)$$

Combining the two balances:

$$u_2 = h_i \quad \rightarrow \quad C_v T_2 = C_p T_i \quad \rightarrow \quad T_2 = (C_p / C_v) T_i = k T_i$$



Substituting,

$$T_2 = 1.4 \times 290\text{K} = 406\text{K} = \mathbf{133^{o}\,C}$$

5-149 A rigid tank initially contains air at atmospheric conditions. The tank is connected to a supply line, and air is allowed to enter the tank until mechanical equilibrium is established. The mass of air that entered and the amount of heat transfer are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Air is an ideal gas with variable specific heats. **3** Kinetic and potential energies are negligible. **4** There are no work interactions involved. **5** The direction of heat transfer is to the tank (will be verified).

Properties The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1). The properties of air are (Table A-17)

$$
\begin{aligned}
T_i &= 295 \text{ K} \longrightarrow h_i = 295.17 \text{ kJ/kg} \\
T_1 &= 295 \text{ K} \longrightarrow u_1 = 210.49 \text{ kJ/kg} \\
T_2 &= 350 \text{ K} \longrightarrow u_2 = 250.02 \text{ kJ/kg}
\end{aligned}
$$

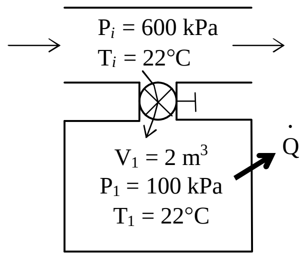
Analysis (*a*) We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{in} - m_{out} = \Delta m_{\text{system}} \quad \rightarrow \quad m_i = m_2 - m_1$

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

$$Q_{in} + m_i h_i = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

The initial and the final masses in the tank are



$$m_1 = \frac{P_1 V}{RT_1} = \frac{(100 \text{ kPa})(2 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(295 \text{ K})} = 2.362 \text{ kg}$$

$$m_2 = \frac{P_2 V}{RT_2} = \frac{(600\text{kPa})(2\text{m}^3)}{(0.287\text{kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(350\text{K})} = 11.946 \text{ kg}$$

Then from the mass balance,

$$m_i = m_2 - m_1 = 11.946 - 2.362 = \mathbf{9.584 \ kg}$$

(*b*) The heat transfer during this process is determined from

$$
\begin{aligned}
Q_{in} &= -m_i h_i + m_2 u_2 - m_1 u_1 \\
&= -(9.584 \text{ kg})(295.17 \text{ kJ/kg}) + (11.946 \text{ kg})(250.02 \text{ kJ/kg}) - (2.362 \text{ kg})(210.49 \text{ kJ/kg}) \\
&= -339 \text{ kJ} \quad \rightarrow \quad Q_{out} = \mathbf{339 \ kJ}
\end{aligned}
$$

Discussion The negative sign for heat transfer indicates that the assumed direction is wrong. Therefore, we reversed the direction.

5-150 A rigid tank initially contains saturated R-134a vapor. The tank is connected to a supply line, and R-134a is allowed to enter the tank. The final temperature in the tank, the mass of R-134a that entered, and the heat transfer are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

Properties The properties of refrigerant are (Tables A-11 through A-13)

$$\left.\begin{array}{l} T_1 = 8^\circ\text{C} \\ x_1 = 0.6 \end{array}\right\} \begin{array}{l} v_1 = v_f + x_1 v_{fg} = 0.0007884 + 0.6 \times (0.0525 - 0.0007884) = 0.03182\text{m}^3/\text{kg} \\ u_1 = u_f + x_1 u_{fg} = 60.43 + 0.6 \times (231.46 - 60.43) = 163.05\text{kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 800\text{kPa} \\ sat.vapor \end{array}\right\} \begin{array}{l} v_2 = v_{g@800\text{kPa}} = 0.0255\text{m}^3/\text{kg} \\ u_2 = u_{g@800\text{kPa}} = 243.78\text{kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_i = 1.0\text{MPa} \\ T_i = 120^\circ C \end{array}\right\} h_i = 356.52\text{kJ/kg}$$

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{in} - m_{out} = \Delta m_{\text{system}} \quad \rightarrow \quad m_i = m_2 - m_1$

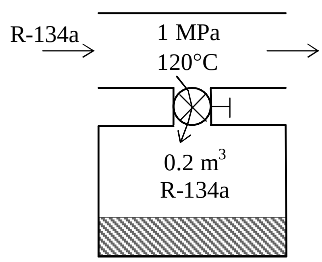
Energy balance:

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

$$Q_{in} + m_i h_i = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

(*a*) The tank contains saturated vapor at the final state at 800 kPa, and thus the final temperature is the saturation temperature at this pressure,

$$T_2 = T_{sat\ @\ 800\ kPa} = \mathbf{31.33^\circ C}$$



(*b*) The initial and the final masses in the tank are

$$m_1 = \frac{V}{v_1} = \frac{0.2\ m^3}{0.03182\ m^3/kg} = 6.29\ kg$$

$$m_2 = \frac{V}{v_2} = \frac{0.2\ m^3}{0.0255\ m^3/kg} = 7.84\ kg$$

Then from the mass balance

$$m_i = m_2 - m_1 = 7.84 - 6.29 = \mathbf{1.55\ kg}$$

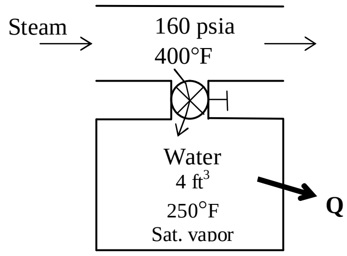
(*c*) The heat transfer during this process is determined from the energy balance to be

$$\begin{aligned} Q_{in} &= -m_i h_i + m_2 u_2 - m_1 u_1 \\ &= -(1.55\text{ kg})(356.52\text{ kJ/kg}) + (7.84\text{ kg})(243.78\text{ kJ/kg}) - (6.29\text{ kg})(163.05\text{ kJ/kg}) \\ &= \mathbf{333\ \ kJ} \end{aligned}$$

5-151E A rigid tank initially contains saturated water vapor. The tank is connected to a supply line, and water vapor is allowed to enter the tank until one-half of the tank is filled with liquid water. The final pressure in the tank, the mass of steam that entered, and the heat transfer are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

Properties The properties of water are (Tables A-4E through A-6E)



$$\left.\begin{array}{l} T_1 = 250^\circ\text{F} \\ sat.vapor \end{array}\right\} \begin{array}{l} v_1 = v_{g@250^\circ\text{F}} = 13.826\,\text{ft}^3/\text{lbm} \\ u_1 = u_{g@250^\circ\text{F}} = 1087.9\,\text{Btu/lbm} \end{array}$$

$$\left.\begin{array}{l} T_2 = 250^\circ F \\ sat.mixture \end{array}\right\} \begin{array}{l} v_f = 0.017001, \quad v_g = 13.826\,\text{ft}^3/\text{lbm} \\ u_f = 218.49, \quad u_g = 1087.9\,\text{Btu/lbm} \end{array}$$

$$\left.\begin{array}{l} P_i = 160\,\text{psia} \\ T_i = 400^\circ\text{F} \end{array}\right\} h_i = 1217.8\,\text{Btu/lbm}$$

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $\quad m_{in} - m_{out} = \Delta m_{\text{system}} \quad \rightarrow \quad m_i = m_2 - m_1$

Energy balance:

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

$$Q_{in} + m_i h_i = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

(a) The tank contains saturated mixture at the final state at 250°F, and thus the exit pressure is the saturation pressure at this temperature,

$$P_2 = P_{sat@250^\circ F} = \mathbf{29.82\ \ psia}$$

(b) The initial and the final masses in the tank are

$$m_1 = \frac{V}{v_1} = \frac{4\ \text{ft}^3}{13.826\ \text{ft}^3/\text{lbm}} = 0.289\ \text{lbm}$$

$$m_2 = m_f + m_g = \frac{V_f}{v_f} + \frac{V_g}{v_g} = \frac{2\ \text{ft}^3}{0.017001\,\text{ft}^3/\text{lbm}} + \frac{2\ \text{ft}^3}{13.826\ \text{ft}^3/\text{lbm}} = 117.64 + 0.14 = 117.78\ \text{lbm}$$

Then from the mass balance: $m_i = m_2 - m_1 = 117.78 - 0.289 = \mathbf{117.49\ lbm}$

(c) The heat transfer during this process is determined from the energy balance to be

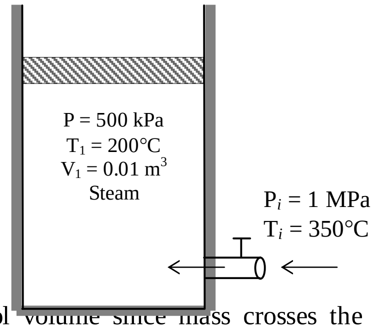
$$\begin{aligned} Q_{in} &= -m_i h_i + m_2 u_2 - m_1 u_1 \\ &= -(117.49\ \text{lbm})(1217.8\ \text{Btu/lbm}) + 25{,}855\ \text{Btu} - (0.289\ \text{lbm})(1087.9\ \text{Btu/lbm}) \\ &= -117{,}539\ \text{Btu} \quad \rightarrow \quad Q_{out} = \mathbf{117{,}539\ Btu} \end{aligned}$$

since $\quad U_2 = m_2 u_2 = m_f u_f + m_g u_g = 117.64 \times 218.49 + 0.14 \times 1087.9 = 25{,}855\ \text{Btu}$

Discussion A negative result for heat transfer indicates that the assumed direction is wrong, and should be reversed.

5-152 A cylinder initially contains superheated steam. The cylinder is connected to a supply line, and is superheated steam is allowed to enter the cylinder until the volume doubles at constant pressure. The final temperature in the cylinder and the mass of the steam that entered are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** The expansion process is quasi-equilibrium. **3** Kinetic and potential energies are negligible. **3** There are no work interactions involved other than boundary work. **4** The device is insulated and thus heat transfer is negligible.



Properties The properties of steam are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 500\text{kPa} \\ T_1 = 200^\circ\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.4249\text{m}^3/\text{kg} \\ u_1 = 2642.9\text{kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_i = 1\text{MPa} \\ T_i = 350^\circ\text{C} \end{array}\right\} h_i = 3157.7\text{kJ/kg}$$

Analysis (*a*) We take the cylinder as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{in} - m_{out} = \Delta m_{\text{system}} \quad \to \quad m_i = m_2 - m_1$

Energy balance:

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

$$m_i h_i = W_{b,out} + m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong ke \cong pe \cong 0)$$

Combining the two relations gives $0 = W_{b,out} - (m_2 - m_1)h_i + m_2 u_2 - m_1 u_1$

The boundary work done during this process is

$$W_{b,out} = \int_1^2 \text{PdV} = \text{P}(\text{V}_2 - \text{V}_1) = (500\text{ kPa})(0.02 - 0.01)\text{m}^3\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) = 5\text{ kJ}$$

The initial and the final masses in the cylinder are

$$m_1 = \frac{V_1}{v_1} = \frac{0.01\text{ m}^3}{0.4249\text{ m}^3/\text{kg}} = 0.0235\text{ kg}$$

$$m_2 = \frac{V_2}{v_2} = \frac{0.02\text{ m}^3}{v_2}$$

Substituting,

$$= 5 - \left(\frac{0.02}{v_2} - 0.0235\right)(3157.7) + \frac{0.02}{v_2}u_2 - (0.0235)(2642.9)$$

Then by trial and error, $T_2 =$ **262.6°C** and $v_2 = 0.4865\text{ m}^3/\text{kg}$

(*b*) The final mass in the cylinder is

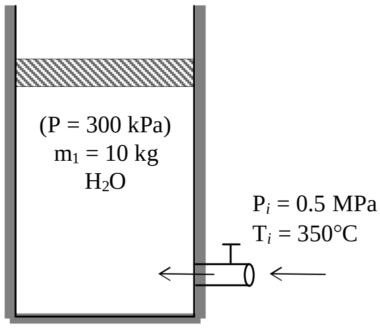
$$m_2 = \frac{V_2}{v_2} = \frac{0.02\text{ m}^3}{0.4865\text{ m}^3/\text{kg}} = 0.0411\text{ kg}$$

Then,

$m_i = m_2 - m_1 = 0.0411 - 0.0235 =$ **0.0176 kg**

5-153 A cylinder initially contains saturated liquid-vapor mixture of water. The cylinder is connected to a supply line, and the steam is allowed to enter the cylinder until all the liquid is vaporized. The final temperature in the cylinder and the mass of the steam that entered are to be determined. √

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** The expansion process is quasi-equilibrium. **3** Kinetic and potential energies are negligible. **3** There are no work interactions involved other than boundary work. **4** The device is insulated and thus heat transfer is negligible.



Properties The properties of steam are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 300\text{kPa} \\ x_1 = 0.8 \end{array}\right\} h_1 = h_f + x_1 h_{fg} = 561.47 + 0.8 \times 2163.8 = 2292.51 \text{kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 300\text{kPa} \\ sat.vapor \end{array}\right\} h_2 = h_{g@300\text{kPa}} = 2725.3 \text{kJ/kg}$$

$$\left.\begin{array}{l} P_i = 0.5\text{MPa} \\ T_i = 350^{\circ}\text{C} \end{array}\right\} h_i = 3167.7 \text{kJ/kg}$$

Analysis (*a*) The cylinder contains saturated vapor at the final state at a pressure of 300 kPa, thus the final temperature in the cylinder must be

$$T_2 = T_{\text{sat @ 300 kPa}} = \mathbf{133.6°C}$$

(*b*) We take the cylinder as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{in} - m_{out} = \Delta m_{\text{system}} \quad \rightarrow \quad m_i = m_2 - m_1$

Energy balance:

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

$$m_i h_i = W_{b,out} + m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong ke \cong pe \cong 0)$$

Combining the two relations gives $0 = W_{b,out} - (m_2 - m_1)h_i + m_2 u_2 - m_1 u_1$

or, $0 = -(m_2 - m_1)h_i + m_2 h_2 - m_1 h_1$

since the boundary work and ΔU combine into ΔH for constant pressure expansion and compression processes. Solving for m_2 and substituting,

$$m_2 = \frac{h_i - h_1}{h_i - h_2} m_1 = \frac{(3167.7 - 2292.51)\text{kJ/kg}}{(3167.7 - 2725.3)\text{kJ/kg}}(10\text{kg}) = 19.78\text{kg}$$

Thus,

$$m_i = m_2 - m_1 = 19.78 - 10 = \mathbf{9.78\ kg}$$

5-154 A rigid tank initially contains saturated R-134a vapor. The tank is connected to a supply line, and R-134a is allowed to enter the tank. The mass of the R-134a that entered and the heat transfer are to be determined.

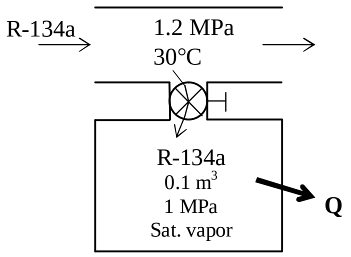
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

Properties The properties of refrigerant are (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_1 = 1\text{ MPa} \\ sat.vapor \end{array}\right\} \begin{array}{l} v_1 = v_{g@1\text{ MPa}} = 0.0202\text{ m}^3\text{/kg} \\ u_1 = u_{g@1\text{ MPa}} = 247.77\text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1.2\text{ MPa} \\ sat.liquid \end{array}\right\} \begin{array}{l} v_2 = v_{f@1.2\text{ MPa}} = 0.0008928\text{ m}^3\text{/kg} \\ u_2 = u_{f@1.2\text{ MPa}} = 114.69\text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_i = 1.2\text{ MPa} \\ T_i = 30^\circ\text{C} \end{array}\right\} h_i = h_{f@30^\circ\text{C}} = 91.49\text{ kJ/kg}$$



Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $\quad m_{in} - m_{out} = \Delta m_{\text{system}} \quad \rightarrow \quad m_i = m_2 - m_1$

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

$$Q_{in} + m_i h_i = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

(*a*) The initial and the final masses in the tank are

$$m_1 = \frac{V_1}{v_1} = \frac{0.1\text{ m}^3}{0.0202\text{ m}^3/\text{kg}} = 4.95\text{ kg}$$

$$m_2 = \frac{V_2}{v_2} = \frac{0.1\text{ m}^3}{0.0008928\text{ m}^3/\text{kg}} = 112.01\text{ kg}$$

Then from the mass balance

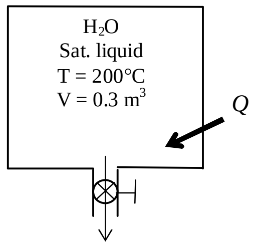
$$m_i = m_2 - m_1 = 112.01 - 4.95 = \mathbf{107.06\ kg}$$

(*c*) The heat transfer during this process is determined from the energy balance to be

$$\begin{aligned} Q_{in} &= -m_i h_i + m_2 u_2 - m_1 u_1 \\ &= -(107.06\text{ kg})(91.49\text{ kJ/kg}) + (112.01\text{ kg})(114.69\text{ kJ/kg}) - (4.95\text{ kg})(247.77\text{ kJ/kg}) \\ &= \mathbf{1825\ kJ} \end{aligned}$$

5-155 A rigid tank initially contains saturated liquid water. A valve at the bottom of the tank is opened, and half of the mass in liquid form is withdrawn from the tank. The temperature in the tank is maintained constant. The amount of heat transfer is to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid leaving the device remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).



Properties The properties of water are (Tables A-4 through A-6)

$$\left.\begin{array}{l} T_1 = 200^\circ\text{C} \\ sat.liquid \end{array}\right\} \begin{array}{l} v_1 = v_{f@200^\circ\text{C}} = 0.001157\ \text{m}^3/\text{kg} \\ u_1 = u_{f@200^\circ\text{C}} = 850.65\ \text{kJ/kg} \end{array}$$

$$\left.\begin{array}{l} T_e = 200^\circ\text{C} \\ sat.liquid \end{array}\right\} h_e = h_{f@200^\circ\text{C}} = 852.45\ \text{kJ/kg}$$

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{in} - m_{out} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$

Energy balance:

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

$$Q_{in} = m_e h_e + m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

The initial and the final masses in the tank are

$$m_1 = \frac{V_1}{v_1} = \frac{0.3\text{m}^3}{0.001157\text{m}^3/\text{kg}} = 259.3\text{kg}$$

$$m_2 = \frac{1}{2} m_1 = \frac{1}{2}(259.3\text{kg}) = 129.65\text{kg}$$

Then from the mass balance,

$$m_e = m_1 - m_2 = 259.3 - 129.65 = 129.65\ \text{kg}$$

Now we determine the final internal energy,

$$v_2 = \frac{V}{m_2} = \frac{0.3\text{m}^3}{129.65\text{kg}} = 0.002314\text{m}^3/\text{kg}$$

$$x_2 = \frac{v_2 - v_f}{v_{fg}} = \frac{0.002314 - 0.001157}{0.13736 - 0.001157} = 0.00849$$

$$\left.\begin{array}{l} T_2 = 200^\circ\text{C} \\ x_2 = 0.00849 \end{array}\right\} u_2 = u_f + x_2 u_{fg} = 850.65 + (0.00849)(1744.7) = 865.46\ \text{kJ/kg}$$

Then the heat transfer during this process is determined from the energy balance by substitution to be

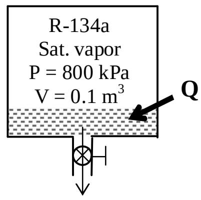
$$Q = (129.65\ \text{kg})(852.45\ \text{kJ/kg}) + (129.65\ \text{kg})(865.46\ \text{kJ/kg}) - (259.3\ \text{kg})(850.65\ \text{kJ/kg}) = \mathbf{2153\ kJ}$$

5-156 A rigid tank initially contains saturated liquid-vapor mixture of refrigerant-134a. A valve at the bottom of the tank is opened, and liquid is withdrawn from the tank at constant pressure until no liquid remains inside. The amount of heat transfer is to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid leaving the device remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

Properties The properties of R-134a are (Tables A-11 through A-13)

$$P_1 = 800\text{kPa} \rightarrow v_f = 0.0008454\text{m}^3/\text{kg},\ v_g = 0.0255\text{m}^3/\text{kg}$$
$$u_f = 92.75\text{kJ/kg},\ \ u_g = 243.78\text{kJ/kg}$$



$$\left.\begin{array}{l} P_2 = 800\text{kPa} \\ sat.vapor \end{array}\right\} \begin{array}{l} v_2 = v_{g@800\text{kPa}} = 0.0255\text{m}^3/\text{kg} \\ u_2 = u_{g@800\text{kPa}} = 243.78\text{kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_e = 800\text{kPa} \\ sat.liquid \end{array}\right\} h_e = h_{f@800\text{kPa}} = 93.42\text{kJ/kg}$$

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $\quad m_{in} - m_{out} = \Delta m_{\text{system}} \quad \rightarrow \quad m_e = m_1 - m_2$

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

$$Q_{in} = m_e h_e + m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

The initial mass, initial internal energy, and final mass in the tank are

$$m_1 = m_f + m_g = \frac{V_f}{v_f} + \frac{V_g}{v_g} = \frac{0.1 \times 0.4\text{m}^3}{0.0008454\text{m}^3/\text{kg}} + \frac{0.1 \times 0.6\text{m}^3}{0.0255\text{m}^3/\text{kg}} = 47.32 + 2.35 = 49.67\text{kg}$$

$$U_1 = m_1 u_1 = m_f u_f + m_g u_g = (47.32)(92.75) + (2.35)(243.78) = 4962\text{kJ}$$

$$m_2 = \frac{V}{v_2} = \frac{0.1\text{m}^3}{0.0255\text{m}^3/\text{kg}} = 3.92\text{kg}$$

Then from the mass and energy balances,

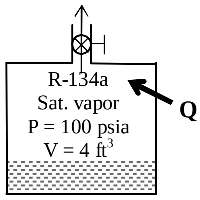
$$m_e = m_1 - m_2 = 49.67 - 3.92 = 45.75\text{ kg}$$

$$Q_{in} = (45.75\text{ kg})(93.42\text{ kJ/kg}) + (3.92\text{ kg})(243.78\text{ kJ/kg}) - 4962\text{ kJ} = \mathbf{267.6\ kJ}$$

5-157E A rigid tank initially contains saturated liquid-vapor mixture of R-134a. A valve at the top of the tank is opened, and vapor is allowed to escape at constant pressure until all the liquid in the tank disappears. The amount of heat transfer is to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid leaving the device remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved.

Properties The properties of R-134a are (Tables A-11E through A-13E)



$$P_1 = 100\text{psia}, \rightarrow v_f = 0.01332\text{ft}^3/\text{lbm},\ v_g = 0.4747\text{ft}^3/\text{lbm}$$
$$u_f = 36.75\text{Btu/lbm},\ u_g = 103.68\text{Btu/lbm}$$

$$\left.\begin{array}{l} P_2 = 100\text{psia} \\ sat.vapor \end{array}\right\} \begin{array}{l} v_2 = v_{g@100\text{psia}} = 0.4747\text{ft}^3/\text{lbm} \\ u_2 = u_{g@100\text{psia}} = 103.68\text{Btu/lbm} \end{array}$$

$$\left.\begin{array}{l} P_e = 100\text{psia} \\ sat.vapor \end{array}\right\} h_e = h_{g@100\text{psia}} = 112.46\text{Btu/lbm}$$

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $\quad m_{in} - m_{out} = \Delta m_{\text{system}} \quad \rightarrow \quad m_e = m_1 - m_2$

Energy balance:

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

$$Q_{in} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

The initial mass, initial internal energy, and final mass in the tank are

$$m_1 = m_f + m_g = \frac{V_f}{v_f} + \frac{V_g}{v_g} = \frac{4 \times 0.2\text{ft}^3}{0.01332\text{ft}^3/\text{lbm}} + \frac{4 \times 0.8\text{ft}^3}{0.4747\text{ft}^3/\text{lbm}} = 60.06 + 6.74 = 66.8\text{lbm}$$

$$U_1 = m_1 u_1 = m_f u_f + m_g u_g = (60.06)(36.75) + (6.74)(103.68) = 2906\text{Btu}$$

$$m_2 = \frac{V}{v_2} = \frac{4\text{ft}^3}{0.4747\text{ft}^3/\text{lbm}} = 8.426\text{lbm}$$

Then from the mass and energy balances,

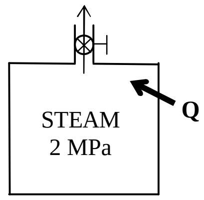
$$m_e = m_1 - m_2 = 66.8 - 8.426 = 58.374 \text{ lbm}$$

$$\begin{aligned} Q_{in} &= m_e h_e + m_2 u_2 - m_1 u_1 \\ &= (58.374 \text{ lbm})(112.46 \text{ Btu/lbm}) + (8.426 \text{ lbm})(103.68 \text{ Btu/lbm}) - 2906 \text{ Btu} \\ &= \mathbf{4532\ Btu} \end{aligned}$$

5-158 A rigid tank initially contains superheated steam. A valve at the top of the tank is opened, and vapor is allowed to escape at constant pressure until the temperature rises to 500°C. The amount of heat transfer is to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process by using constant average properties for the steam leaving the tank. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

Properties The properties of water are (Tables A-4 through A-6)



$$\left.\begin{array}{l} P_1 = 2\text{MPa} \\ T_1 = 300^{\circ}\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.12547\text{m}^3/\text{kg} \\ u_1 = 2772.6\text{kJ/kg}, \quad h_1 = 3023.5\text{kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 2\text{MPa} \\ T_2 = 500^{\circ}\text{C} \end{array}\right\} \begin{array}{l} v_2 = 0.17568\text{m}^3/\text{kg} \\ u_2 = 3116.2\text{kJ/kg}, \quad h_2 = 3467.6\text{kJ/kg} \end{array}$$

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $$m_{in} - m_{out} = \Delta m_{\text{system}} \quad \rightarrow \quad m_e = m_1 - m_2$$

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

$$Q_{in} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

The state and thus the enthalpy of the steam leaving the tank is changing during this process. But for simplicity, we assume constant properties for the exiting steam at the average values. Thus,

$$h_e \cong \frac{h_1 + h_2}{2} = \frac{3023.5 + 3467.6 \text{ kJ/kg}}{2} = 3245.55 \text{ kJ/kg}$$

The initial and the final masses in the tank are

$$m_1 = \frac{V_1}{v_1} = \frac{0.2 \text{ m}^3}{0.12547 \text{ m}^3/\text{kg}} = 1.594 \text{ kg}$$

$$m_2 = \frac{V_2}{v_2} = \frac{0.2 \text{ m}^3}{0.17568 \text{ m}^3/\text{kg}} = 1.138 \text{ kg}$$

Then from the mass and energy balance relations,

$$m_e = m_1 - m_2 = 1.594 - 1.138 = 0.456 \text{ kg}$$

$$\begin{aligned} Q_{in} &= m_e h_e + m_2 u_2 - m_1 u_1 \\ &= (0.456 \text{ kg})(3245.55 \text{ kJ/kg}) + (1.138 \text{ kg})(3116.2 \text{ kJ/kg}) - (1.594 \text{ kg})(2772.6 \text{ kJ/kg}) \\ &= \mathbf{606.7 \ kJ} \end{aligned}$$

5-159 A pressure cooker is initially half-filled with liquid water. If the pressure cooker is not to run out of liquid water for 1 h, the highest rate of heat transfer allowed is to be determined.

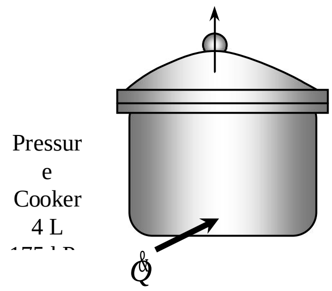
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid leaving the device remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved.

Properties The properties of water are (Tables A-4 through A-6)

$$P_1 = 175 kPa \;\rightarrow\; v_f = 0.001057 \text{m}^3/\text{kg},\; v_g = 1.0036 \text{m}^3/\text{kg}$$
$$u_f = 486.8 \text{kJ/kg},\; u_g = 2524.9 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 175 \text{kPa} \\ sat.vapor \end{array}\right\} \begin{array}{l} v_2 = v_{g@175\text{kPa}} = 1.0036 \text{m}^3/\text{kg} \\ u_2 = u_{g@175\text{kPa}} = 2524.9 \text{kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_e = 175 \text{kPa} \\ sat.vapor \end{array}\right\} h_e = h_{g@175\text{kPa}} = 2700.6 \text{kJ/kg}$$



Analysis We take the cooker as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{in} - m_{out} = \Delta m_{\text{system}} \;\rightarrow\; m_e = m_1 - m_2$

Energy balance:

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

$$Q_{in} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

The initial mass, initial internal energy, and final mass in the tank are

$$m_1 = m_f + m_g = \frac{V_f}{v_f} + \frac{V_g}{v_g} = \frac{0.002 \text{m}^3}{0.001057 \text{m}^3/\text{kg}} + \frac{0.002 \text{m}^3}{1.0036 \text{m}^3/\text{kg}} = 1.892 + 0.002 = 1.894 \text{kg}$$

$$U_1 = m_1 u_1 = m_f u_f + m_g u_g = (1.892)(486.8) + (0.002)(2524.9) = 926.1 \text{kJ}$$

$$m_2 = \frac{V}{v_2} = \frac{0.004 \text{m}^3}{1.0036 \text{m}^3/\text{kg}} = 0.004 \text{kg}$$

Then from the mass and energy balances,

$$m_e = m_1 - m_2 = 1.894 - 0.004 = 1.890 \text{ kg}$$

$$Q_{in} = m_e h_e + m_2 u_2 - m_1 u_1$$
$$= (1.890 \text{kg})(2700.6 \text{kJ/kg}) + (0.004 \text{kg})(2524.9 \text{kJ/kg}) - 926.1 \text{kJ} = 4188 \text{kJ}$$

Thus,

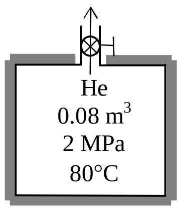
$$\dot{Q} = \frac{Q}{\Delta t} = \frac{4188 \text{ kJ}}{3600 \text{ s}} = \mathbf{1.163\ kW}$$

5-160 An insulated rigid tank initially contains helium gas at high pressure. A valve is opened, and half of the mass of helium is allowed to escape. The final temperature and pressure in the tank are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process by using constant average properties for the helium leaving the tank. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The tank is insulated and thus heat transfer is negligible. **5** Helium is an ideal gas with constant specific heats.

Properties The specific heat ratio of helium is k =1.667 (Table A-2).

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as



Mass balance:

$$m_{in} - m_{out} = \Delta m_{\text{system}} \quad \to \quad m_e = m_1 - m_2$$

$$m_2 = \tfrac{1}{2} m_1 \quad \text{(given)} \quad \longrightarrow \quad m_e = m_2 = \tfrac{1}{2} m_1$$

Energy balance:

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

$$-m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong Q \cong ke \cong pe \cong 0)$$

Note that the state and thus the enthalpy of helium leaving the tank is changing during this process. But for simplicity, we assume constant properties for the exiting steam at the average values.

Combining the mass and energy balances: $\quad 0 = \tfrac{1}{2} m_1 h_e + \tfrac{1}{2} m_1 u_2 - m_1 u_1$

Dividing by $2m_1$: $\quad 0 = h_e + u_2 - 2u_1 \;\; or \;\; 0 = C_p \dfrac{T_1 + T_2}{2} + C_v T_2 - 2C_v T_1$

Dividing by C_v: $\quad 0 = k(T_1 + T_2) + 2T_2 - 4T_1 \quad$ since $k = C_p / C_v$

Solving for T_2: $\quad T_2 = \dfrac{(4-k)}{(2+k)} T_1 = \dfrac{(4-1.667)}{(2+1.667)}(353\,\text{K}) = \mathbf{225\,K}$

The final pressure in the tank is

$$\frac{P_1 V}{P_2 V} = \frac{m_1 R T_1}{m_2 R T_2} \longrightarrow P_2 = \frac{m_2 T_2}{m_1 T_2} P_1 = \frac{1}{2}\frac{225}{353}(2000\,\text{kPa}) = \mathbf{637\,kPa}$$

5-161E An insulated rigid tank equipped with an electric heater initially contains pressurized air. A valve is opened, and air is allowed to escape at constant temperature until the pressure inside drops to 30 psia. The amount of electrical work transferred is to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the exit temperature (and enthalpy) of air remains constant. **2** Kinetic and potential energies are negligible. **3** The tank is insulated and thus heat transfer is negligible. **4** Air is an ideal gas with variable specific heats.

Properties The gas constant of air is R =0.3704 psia.ft^3/lbm.R (Table A-1E). The properties of air are (Table A-17E)

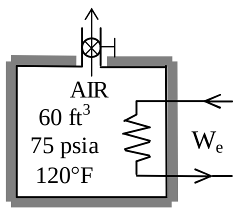
$$\begin{aligned} T_i &= 580 \text{ R} \longrightarrow h_i = 138.66 \text{ Btu/lbm} \\ T_1 &= 580 \text{ R} \longrightarrow u_1 = 98.90 \text{ Btu/lbm} \\ T_2 &= 580 \text{ R} \longrightarrow u_2 = 98.90 \text{ Btu/lbm} \end{aligned}$$

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $$m_{in} - m_{out} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$$

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

$$W_{e,in} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong ke \cong pe \cong 0)$$



The initial and the final masses of air in the tank are

$$m_1 = \frac{P_1 V}{RT_1} = \frac{(75 \text{psia})(60 \text{ft}^3)}{(0.3704 \text{psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(580 \text{R})} = 20.95 \text{lbm}$$

$$m_2 = \frac{P_2 V}{RT_2} = \frac{(30 \text{psia})(60 \text{ft}^3)}{(0.3704 \text{psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(580 \text{R})} = 8.38 \text{lbm}$$

Then from the mass and energy balances,

$$m_e = m_1 - m_2 = 20.95 - 8.38 = 12.57 \text{ lbm}$$

$$\begin{aligned} W_{e,in} &= m_e h_e + m_2 u_2 - m_1 u_1 \\ &= (12.57 \text{ lbm})(138.66 \text{ Btu/lbm}) + (8.38 \text{ lbm})(98.90 \text{ Btu/lbm}) - (20.95 \text{ lbm})(98.90 \text{ Btu/lbm}) \\ &= \mathbf{500 \text{ Btu}} \end{aligned}$$

5-162 A vertical cylinder initially contains air at room temperature. Now a valve is opened, and air is allowed to escape at constant pressure and temperature until the volume of the cylinder goes down by half. The amount air that left the cylinder and the amount of heat transfer are to be determined.

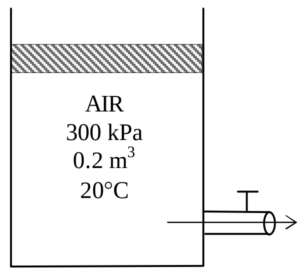
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the exit temperature (and enthalpy) of air remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions. **4** Air is an ideal gas with constant specific heats. **5** The direction of heat transfer is to the cylinder (will be verified).

Properties The gas constant of air is R =0.287 kPa.m^3/kg.K.

Analysis (*a*) We take the cylinder as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{in} - m_{out} = \Delta m_{\text{system}} \quad \rightarrow \quad m_e = m_1 - m_2$

Energy balance:



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

$$Q_{in} + W_{b,in} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } ke \cong pe \cong 0)$$

The initial and the final masses of air in the cylinder are

$$m_1 = \frac{P_1 V_1}{RT_1} = \frac{(300\text{kPa})(0.2\text{m}^3)}{(0.287\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\text{K})} = 0.714\text{kg}$$

$$m_2 = \frac{P_2 V_2}{RT_2} = \frac{(300\text{kPa})(0.1\text{m}^3)}{(0.287\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\text{K})} = 0.357\text{kg} = \frac{1}{2} m_1$$

Then from the mass balance,

$$m_e = m_1 - m_2 = 0.714 - 0.357 = \mathbf{0.357\ kg}$$

(*b*) This is a constant pressure process, and thus the W_b and the ΔU terms can be combined into Δh to yield

$$Q = m_e h_e + m_2 h_2 - m_1 h_1$$

Noting that the temperature of the air remains constant during this process, we have $h_i = h_1 = h_2 = h$.
Also, $m_e = m_2 = \frac{1}{2} m_1$. Thus,

$$Q = \left(\frac{1}{2} m_1 + \frac{1}{2} m_1 - m_1\right) h = \mathbf{0}$$

5-163 A balloon is initially filled with helium gas at atmospheric conditions. The tank is connected to a supply line, and helium is allowed to enter the balloon until the pressure rises from 100 to 150 kPa. The final temperature in the balloon is to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Helium is an ideal gas with constant specific heats. **3** The expansion process is quasi-equilibrium. **4** Kinetic and potential energies are negligible. **5** There are no work interactions involved other than boundary work. **6** Heat transfer is negligible.

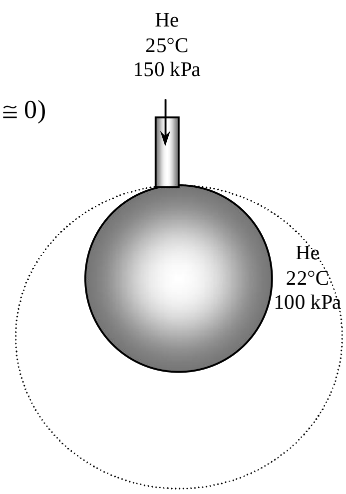
Properties The gas constant of helium is R = 2.0769 kJ/kg·K (Table A-1). The specific heats of helium are C_p = 5.1926 and C_v = 3.1156 kJ/kg·K (Table A-2a).

Analysis We take the cylinder as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{in} - m_{out} = \Delta m_{\text{system}} \quad \rightarrow \quad m_i = m_2 - m_1$

Energy balance:

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



$$m_i h_i = W_{b,out} + m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong ke \cong pe \cong 0)$$

$$m_1 = \frac{P_1 V_1}{R T_1} = \frac{(100\,\text{kPa})(65\,\text{m}^3)}{(2.0769\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(295\,\text{K})} = 10.61\,\text{kg}$$

$$\frac{P_1}{P_2} = \frac{V_1}{V_2} \longrightarrow V_2 = \frac{P_2}{P_1} V_1 = \frac{150\,\text{kPa}}{100\,\text{kPa}}(65\,\text{m}^3) = 97.5\,\text{m}^3$$

$$m_2 = \frac{P_2 V_2}{R T_2} = \frac{(150\,\text{kPa})(97.5\,\text{m}^3)}{(2.0769\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(T_2\,\text{K})} = \frac{7041.74}{T_2}\,\text{kg}$$

Then from the mass balance,

$$m_i = m_2 - m_1 = \frac{7041.74}{T_2} - 10.61\,\text{kg}$$

Noting that P varies linearly with V, the boundary work done during this process is

$$W_b = \frac{P_1 + P_2}{2}(V_2 - V_1) = \frac{(100+150)\,\text{kPa}}{2}(97.5-65)\,\text{m}^3 = 4062.5\,\text{kJ}$$

Using specific heats, the energy balance relation reduces to

$$W_{b,out} = m_i C_p T_i - m_2 C_v T_2 + m_1 C_v T_1$$

Substituting,

$$4062.5 = \left(\frac{7041.74}{T_2} - 10.61\right)(5.1926)(298) - \frac{7041.74}{T_2}(3.1156)T_2 + (10.61)(3.1156)(295)$$

It yields $\quad T_2$ = **333.6 K**

5-164 A balloon is initially filled with pressurized helium gas. Now a valve is opened, and helium is allowed to escape until the pressure inside drops to atmospheric pressure. The final temperature of helium in the balloon and the amount helium that has escaped are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process by assuming exit properties to be constant at average conditions. **2** Kinetic and potential energies are negligible. **3** There are no work interactions other than boundary work. **4** Helium is an ideal gas with constant specific heats. **5** Heat transfer is negligible.

Properties The gas constant of helium is R =2.0769 kPa.m^3/kg.K (Table A-1). The specific heats of helium are C_p = 5.1926 and C_v = 3.1156 kJ/kg·K (Table A-2).

Analysis The properties of helium leaving the balloon are changing during this process. But we will treat them as a constant at the average temperature. Thus $T_e \cong (T_1 + T_2)/2$. Also $h = C_pT$ and $u = C_vT$.

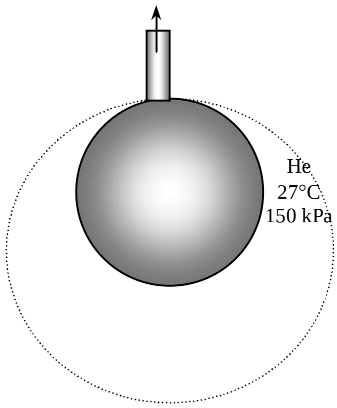
We take the balloon as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{in} - m_{out} = \Delta m_{\text{system}} \quad \rightarrow \quad m_e = m_1 - m_2$

Energy balance:

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

$$W_{b,in} - m_eh_e = m_2u_2 - m_1u_1 \quad (\text{since } Q \cong ke \cong pe \cong 0)$$



or

$$W_{b,in} = m_eC_p\frac{T_1 + T_2}{2} + m_2C_vT_2 - m_1C_vT_1$$

The initial and the final masses in the balloon are

$$m_1 = \frac{P_1V_1}{RT_1} = \frac{(150\text{kPa})(10\text{m}^3)}{(2.0769\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{K})} = 2.4074\text{kg}$$

$$m_2 = \frac{P_2V_2}{RT_2} = \frac{(100\text{kPa})(8.5\text{m}^3)}{(2.0769\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})T_2} = \frac{409.264}{T_2}$$

Then from the mass balance,

$$m_e = m_1 - m_2 = 2.4074 - \frac{409.264}{T_2}$$

Noting that the pressure changes linearly with volume, the boundary work done during this process is

$$W_b = \frac{P_1 + P_2}{2}(V_2 - V_1) = \frac{(150+100)\text{kPa}}{2}(8.5-10)\text{m}^3 = -187.5\text{kJ}$$

Combining mass and energy balances and substituting,

$$-(-187.5) = \left(2.4074 - \frac{409.264}{T_2}\right)(5.1926)\frac{300+T_2}{2} + \frac{409.264}{T_2}(3.1156)T_2 - (2.4074)(3.1156)(300)$$

It yields $\qquad T_2^2 - 56T_2 - 51{,}003.5 = 0 \quad \rightarrow \quad T_2 = \mathbf{256\ K}$

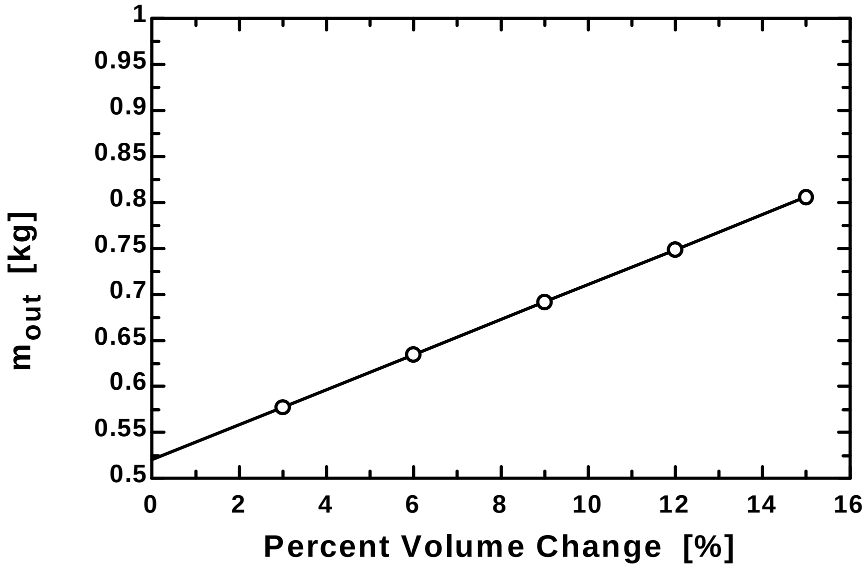
(*b*) The amount of helium that has escaped is

$$m_e = m_1 - m_2 = 2.4074 - \frac{409.264}{T_2} = 2.1074 - \frac{409.264}{256} = \mathbf{0.509\ \ kg}$$

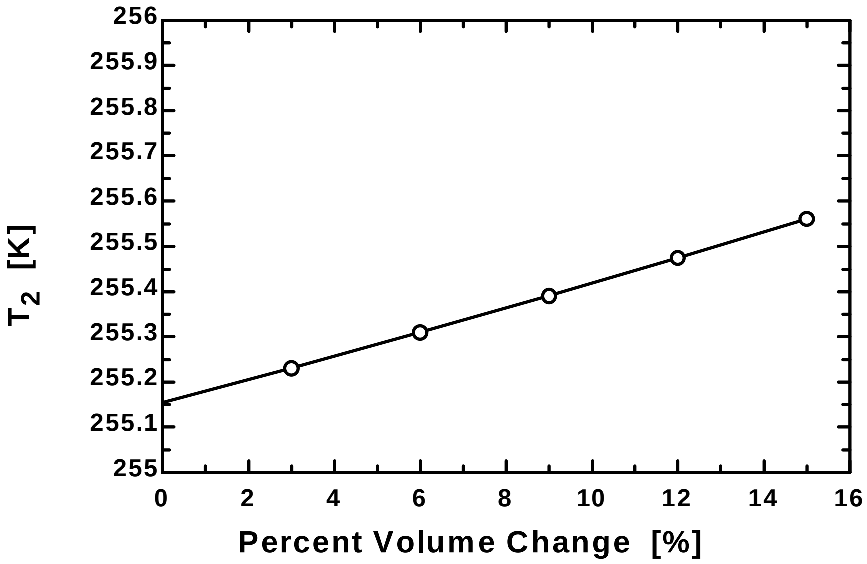
5-165 Problem 5-164 is reconsidered. The effect of the percent change of the volume of the balloon (in the range of 0 to 15%) on the final temperature in the balloon and the amount of mass that has escaped is to be investigated. The final temperature and the amount of discharged helium are to be plotted against the percent change in volume.

```
"Knowns:"
C_P = 5.1926"[kJ/kg-K ]"
C_V = 3.1156 "[kJ/kg-K ]"
R=2.0769 "[kPa-m^3/kg-K]"
P_1= 150"[kPa]"
P_2= 100"[kPa]"
T_1 = 300"[K]"
V_1 = 10"[m^3]"
{PCVolChange =15 "[%]"} "Percent Volume Change"
{V_2 = 8.5"[m^3]"}
T_out =  (T_1 + T_2)/2"[K]"

V_1*PCVolChange=(V_1-V_2)*100

"Analysis:
"Mass balance:"
m_in = 0"[kg]"
m_in - m_out = m_2 - m_1

"Energy balance:"
E_in - E_out = DELTAE_sys
E_in = W_b_in"[kJ]"
E_out = m_out*h_out"[kJ]"
h_out  = C_P*T_out"[kJ/kg]"

DELTAE_sys = m_2*u_2-m_1*u_1

u_1=C_V*T_1"[kJ/kg]"
u_2= C_V*T_2"[kJ/kg]"

"The volume flow rates of air are determined to be:"
P_1*V_1=m_1*R*T_1
P_2*V_2=m_2*R*T_2

"Boundary Work: Due to pressure changing linearly with volume.  Note the minus sign for
work in"
W_b_in = -(P_1+P_2)/2*(V_2-V_1)
```

m_{out} [kg]	PCVolChange [%]	T_2 [K]
0.5204	0	255.2
0.5775	3	255.2
0.6347	6	255.3
0.6918	9	255.4
0.7489	12	255.5
0.806	15	255.6



m_out [kg]
Percent Volume Change [%]



T_2 [K]
Percent Volume Change [%]

5-166 A vertical piston-cylinder device equipped with an external spring initially contains superheated steam. Now a valve is opened, and steam is allowed to escape until the volume of the cylinder goes down by half. The initial and final masses of steam in the cylinder and the amount of heat transferred are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process by assuming the properties of steam that escape to be constant at average conditions. **2** Kinetic and potential energies are negligible. **3** The spring is a linear spring. **4** The direction of heat transfer is to the cylinder (will be verified).

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

$$\left.\begin{array}{l} P_1 = 1\text{ MPa} \\ T_1 = 250^{\circ}\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.2327\text{m}^3/\text{kg} \\ u_1 = 2709.9\text{kJ/kg}, \quad h_1 = 2942.6\text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 800\text{ kPa} \\ sat.vapor \end{array}\right\} \begin{array}{l} v_2 = 0.2404\text{ m}^3/\text{kg} \\ u_2 = 2576.8\text{ kJ/kg}, \quad h_2 = 2769.1\text{ kJ/kg} \end{array}$$

Analysis (*a*) We take the cylinder as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{in} - m_{out} = \Delta m_{\text{system}} \quad \rightarrow \quad m_e = m_1 - m_2$

Energy balance:

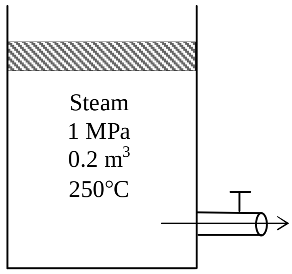
$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{in} + W_{b,in} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } ke \cong pe \cong 0)$$

The state and thus the enthalpy of the steam leaving the cylinder is changing during this process. But for simplicity, we assume constant properties for the exiting steam at the average values. Thus,

$$h_e \cong \frac{h_1 + h_2}{2} = \frac{2942.6 + 2769.1\text{ kJ/kg}}{2} = 2855.9\text{ kJ/kg}$$

The initial and the final masses in the tank are



$$m_1 = \frac{V_1}{v_1} = \frac{0.2\text{ m}^3}{0.2327\text{ m}^3/\text{kg}} = \mathbf{0.859\ kg}$$

$$m_2 = \frac{V_2}{v_2} = \frac{0.1\text{ m}^3}{0.2404\text{ m}^3/\text{kg}} = \mathbf{0.416\ kg}$$

Then from the mass balance,

$$m_e = m_1 - m_2 = 0.859 - 0.416 = 0.443\text{ kg}$$

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

$$W_{b,in} = \frac{P_1 + P_2}{2}(V_1 - V_2) = \frac{(1000 + 800)\text{kPa}}{2}(0.2 - 0.1)\text{m}^3 = 90\text{ kJ}$$

Then the heat transfer during this process becomes

$$\begin{aligned} Q_{in} &= -W_{b,in} + m_e h_e + m_2 u_2 - m_1 u_1 \\ &= -90\text{ kJ} + (0.443\text{kg})(2855.9\text{kJ/kg}) + (0.416\text{kg})(2576.8\text{kJ/kg}) - (0.859\text{kg})(2709.9\text{kJ/kg}) \\ &= -80.7\text{ kJ} \quad \rightarrow \quad Q_{out} = \mathbf{80.7kJ} \end{aligned}$$

5-167 A vertical piston-cylinder device initially contains steam at a constant pressure of 300 kPa. Now a valve is opened, and steam is allowed to escape at constant temperature and pressure until the volume reduces to one-third. The mass of steam that escaped and the amount of heat transfer are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the properties of steam that escape remain constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions other than boundary work. **4** The direction of heat transfer is to the cylinder (will be verified).

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

$$\left.\begin{array}{l} P_e = P_1 = P_2 = 300\text{kPa} \\ T_e = T_1 = T_2 = 250^\circ\text{C} \end{array}\right\} \begin{array}{l} h_e = 2967.6\text{kJ/kg} \\ v_1 = v_2 = 0.7964\text{m}^3/\text{kg} \\ u_1 = u_2 = 2728.7\text{kJ/kg} \end{array}$$

Analysis (*a*) We take the cylinder as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $$m_{in} - m_{out} = \Delta m_{\text{system}} \quad \rightarrow \quad m_e = m_1 - m_2$$

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

$$Q_{in} + W_{b,in} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } ke \cong pe \cong 0)$$

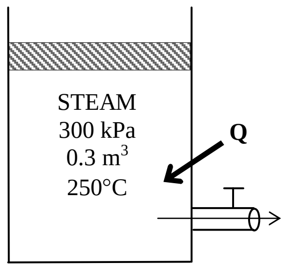
or $$Q_{in} = m_e h_e + m_2 h_2 - m_1 h_1$$

since for a constant pressure process, the W_b and the ΔU terms can be combined into ΔH.

The initial and final masses of steam in the cylinder are

$$m_1 = \frac{V_1}{v_1} = \frac{0.3\text{ m}^3}{0.7964\text{ m}^3/\text{kg}} = 0.377\text{ kg}$$

$$m_2 = \frac{V_2}{v_2} = \frac{0.1\text{ m}^3}{0.7964\text{ m}^3/\text{kg}} = 0.126\text{ kg}$$



Then from the mass balance,

$$m_e = m_1 - m_2 = 0.377 - 0.126 = \mathbf{0.251\ kg}$$

(*b*) Noting that $h_e = h_1 = h_2 = h$ and $m_e = m_1 - m_2$, the energy balance relation reduces to

$$\begin{aligned} Q_{in} &= m_e h_e + m_2 h_2 - m_1 h_1 \\ &= (m_e + m_2 - m_1)h \\ &= \mathbf{0} \end{aligned}$$

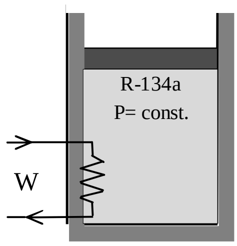
Therefore, there will be no heat transfer during this process.

Review Problems

5-168 A cylinder is initially filled with saturated R-134a vapor at a specified pressure. The refrigerant is heated both electrically and by heat transfer at constant pressure for 6 min. The electric current is to be determined, and the process is to be shown on a *T-v* diagram.

Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are negligible. **2** The thermal energy stored in the cylinder itself and the wires is negligible. **3** The compression or expansion process is quasi-equilibrium.

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as



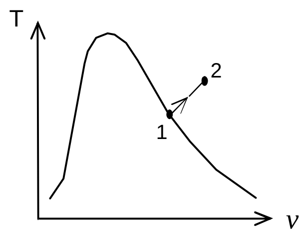
$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{in} + W_{e,in} - W_{b,out} = \Delta U \quad (\text{since } Q = \text{KE} = \text{PE} = 0)$$

$$Q_{in} + W_{e,in} = m(h_2 - h_1)$$

$$Q_{in} + (VI\Delta t) = m(h_2 - h_1)$$

since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. The properties of R-134a are (Tables A-11 through A-13)



$$\left.\begin{array}{l} P_1 = 200\text{kPa} \\ \text{sat.vapor} \end{array}\right\} h_1 = h_{g@200\text{kPa}} = 241.30\text{kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 200\text{kPa} \\ T_1 = 70^0\text{C} \end{array}\right\} h_2 = 314.02\text{kJ/kg}$$

Substituting,

$$250{,}000\text{VA} + (110\text{V})(I)(6 \times 60\text{s}) = (12\text{kg})(314.02 - 241.30)\text{kJ/kg}\left(\frac{1000\text{VA}}{1\text{kJ/s}}\right)$$

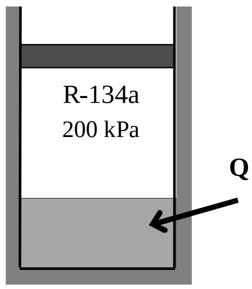
$$I = \mathbf{15.72A}$$

5-169 A cylinder is initially filled with saturated liquid-vapor mixture of R-134a at a specified pressure. Heat is transferred to the cylinder until the refrigerant vaporizes completely at constant pressure. The initial volume, the work done, and the total heat transfer are to be determined, and the process is to be shown on a *P*-*v* diagram.

Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are negligible. **2** The thermal energy stored in the cylinder itself is negligible. **3** The compression or expansion process is quasi-equilibrium.

Analysis (*a*) Using property data from R-134a tables (Tables A-11 through A-13), the initial volume of the refrigerant is determined to be

$$\left.\begin{array}{l} P_1 = 200\text{kPa} \\ x_1 = 0.25 \end{array}\right\} \begin{array}{ll} v_f = 0.0007532, & v_g = 0.0993\text{m}^3/\text{kg} \\ u_f = 36.69, & u_g = 221.43\text{kJ/kg} \end{array}$$



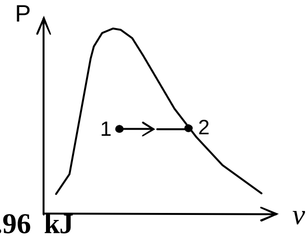
$$v_1 = v_f + x_1 v_{fg} = 0.0007532 + 0.25\times(0.0993 - 0.0007532) = 0.02539\text{m}^3/\text{kg}$$

$$u_1 = u_f + x_1 u_{fg} = 36.69 + 0.25\times(221.43 - 36.69) = 82.88\text{kJ/kg}$$

$$V_1 = mv_1 = (0.2\text{kg})(0.02539\text{m}^3/\text{kg}) = \mathbf{0.005078m^3}$$

(*b*) The work done during this constant pressure process is

$$\left.\begin{array}{l} P_2 = 200\text{kPa} \\ sat.vapor \end{array}\right\} \begin{array}{l} v_2 = v_{g@200\text{kPa}} = 0.0993\text{m}^3/\text{kg} \\ u_2 = u_{g@200\text{kPa}} = 221.43\text{kJ/kg} \end{array}$$



$$W_{b,out} = \int_1^2 PdV = P(V_2 - V_1) = mP(v_2 - v_1)$$

$$= (0.2\text{ kg})(200\text{ kPa})(0.0993 - 0.02539)\text{m}^3/\text{kg}\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) = \mathbf{2.96\ kJ}$$

(*c*) We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

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

$$Q_{in} - W_{b,out} = \Delta U$$

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

Substituting,

$$Q_{in} = (0.2\text{ kg})(221.43 - 82.88)\text{kJ/kg} + 2.96 = \mathbf{30.67\ kJ}$$

5-170 A cylinder is initially filled with helium gas at a specified state. Helium is compressed polytropically to a specified temperature and pressure. The heat transfer during the process is to be determined.

Assumptions **1** Helium is an ideal gas with constant specific heats. **2** The cylinder is stationary and thus the kinetic and potential energy changes are negligible. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

Properties The gas constant of helium is R = 2.0769 kPa.m^3/kg.K (Table A-1). Also, C_v = 3.1156 kJ/kg.K (Table A-2).

Analysis The mass of helium and the exponent n are determined to be

$$m = \frac{P_1V_1}{RT_1} = \frac{(150\text{kPa})(0.5\text{m}^3)}{(2.0769\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\text{K})} = 0.123\text{kg}$$

$$\frac{P_1V_1}{RT_1} = \frac{P_2V_2}{RT_2} \longrightarrow V_2 = \frac{T_2P_1}{T_1P_2}V_1 = \frac{413\text{K}}{293\text{K}} \times \frac{150\text{kPa}}{400\text{kPa}} \times 0.5\text{m}^3 = 0.264\text{m}^3$$

$$P_2V_2^n = P_1V_1^n \longrightarrow \left(\frac{P_2}{P_1}\right) = \left(\frac{V_1}{V_2}\right)^n \longrightarrow \frac{400}{150} = \left(\frac{0.5}{0.264}\right)^n \longrightarrow n = 1.536$$

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

$$W_{b,in} = -\int_1^2 PdV = -\frac{P_2V_2 - P_1V_1}{1-n} = -\frac{mR(T_2 - T_1)}{1-n}$$

$$= -\frac{(0.123\text{kg})(2.0769\text{kJ/kg}\cdot\text{K})(413-293)\text{K}}{1-1.536} = \mathbf{57.2\ kJ}$$

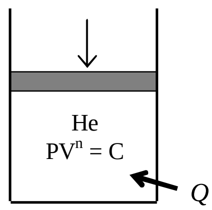
We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. Taking the direction of heat transfer to be to the cylinder, the energy balance for this stationary closed system can be expressed as

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

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

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

$$= mC_v(T_2 - T_1) - W_{b,in}$$



Substituting,

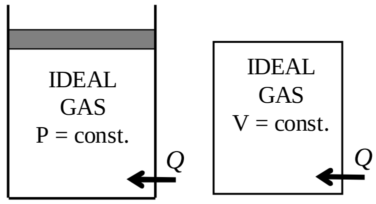
Q_{in} = (0.123 kg)(3.1156 kJ/kg·K)(413 - 293)K - (57.2 kJ) = **-11.2 kJ**

The negative sign indicates that heat is lost from the system.

5-171 A cylinder and a rigid tank initially contain the same amount of an ideal gas at the same state. The temperature of both systems is to be raised by the same amount. The amount of extra heat that must be transferred to the cylinder is to be determined.

Analysis In the absence of any work interactions, other than the boundary work, the ΔH and ΔU represent the heat transfer for ideal gases for constant pressure and constant volume processes, respectively. Thus the extra heat that must be supplied to the air maintained at constant pressure is

$$Q_{\text{in, extra}} = \Delta H - \Delta U = mC_p\Delta T - mC_v\Delta T = m(C_p - C_v)\Delta T = mR\Delta T$$



where

$$R = \frac{R_u}{M} = \frac{8.314 \text{ kJ/kmol}\cdot\text{K}}{25 \text{ kg/kmol}} = 0.3326 \text{ kJ/kg}\cdot\text{K}$$

Substituting,

$$Q_{\text{in, extra}} = (12 \text{ kg})(0.3326 \text{ kJ/kg·K})(15 \text{ K}) = \mathbf{59.9\ kJ}$$

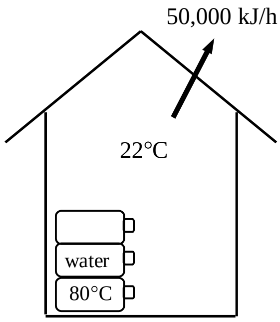
5-172 The heating of a passive solar house at night is to be assisted by solar heated water. The length of time that the electric heating system would run that night with or without solar heating are to be determined.

Assumptions **1** Water is an incompressible substance with constant specific heats. **2** The energy stored in the glass containers themselves is negligible relative to the energy stored in water. **3** The house is maintained at 22°C at all times.

Properties The density and specific heat of water at room temperature are ρ = 1 kg/L and C = 4.18 kJ/kg·°C (Table A-3).

Analysis The total mass of water is

$$m_w = \rho V = (1\text{kg/L})(50\times 20\text{L}) = 1000\text{kg}$$



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

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

$$W_{e,in} - Q_{out} = \Delta U = (\Delta U)_{\text{water}} + (\Delta U)_{\text{air}}$$

$$= (\Delta U)_{\text{water}}$$

$$= mC(T_2 - T_1)_{\text{water}}$$

or,

$$\dot{W}_{e,in}\Delta t - Q_{out} = [mC(T_2 - T_1)]_{\text{water}}$$

Substituting,

(15 kJ/s)Δt - (50,000 kJ/h)(10 h) = (1000 kg)(4.18 kJ/kg·°C)(22 - 80)°C

It gives

Δt = 17,170 s = **4.77 h**

(b) If the house incorporated no solar heating, the energy balance relation above would simplify further to

$$\dot{W}_{e,in}\Delta t - Q_{out} = 0$$

Substituting,

(15 kJ/s)Δt - (50,000 kJ/h)(10 h) = 0

It gives

Δt = 33,333 s = **9.26 h**

5-173 An electric resistance heater is immersed in water. The time it will take for the electric heater to raise the water temperature to a specified temperature is to be determined.

Assumptions **1** Water is an incompressible substance with constant specific heats. **2** The energy stored in the container itself and the heater is negligible. **3** Heat loss from the container is negligible.

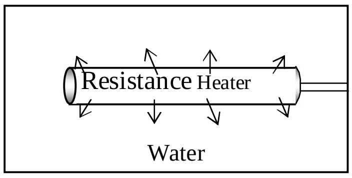
Properties The density and specific heat of water at room temperature are ρ = 1 kg/L and C = 4.18 kJ/kg·°C (Table A-3).

Analysis Taking the water in the container as the system, the energy balance can be expressed as

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

$$W_{e,in} = (\Delta U)_{\text{water}}$$

$$\dot{W}_{e,in}\Delta t = mC(T_2 - T_1)_{\text{water}}$$



Substituting,

(800 J/s)Δt = (40 kg)(4180 J/kg·°C)(80 - 20)°C

Solving for Δt gives

Δt = **12,540 s = 209.0 min = 3.483 h**

5-174 One ton of liquid water at 80°C is brought into a room. The final equilibrium temperature in the room is to be determined.

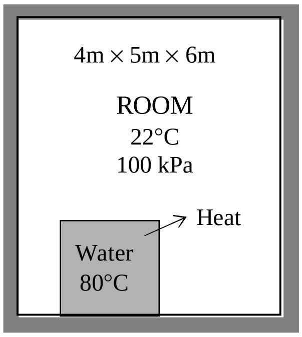
Assumptions **1** The room is well insulated and well sealed. **2** The thermal properties of water and air are constant.

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

Analysis The volume and the mass of the air in the room are

$$V = 4\text{x}5\text{x}6 = 120 \text{ m}^3$$

$$m_{air} = \frac{P_1V_1}{RT_1} = \frac{(100 \text{ kPa})(120\text{P m}^3)}{(0.2870 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(295 \text{ K})} = 141.7 \text{ kg}$$



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

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

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

Substituting,

$$(1000 \text{ kg})(4.180 \text{ kJ/kg}\cdot^{\circ}\text{C})(T_f - 80)^{\circ}\text{C} + (147.7 \text{ kg})(0.718 \text{ kJ/kg}\cdot^{\circ}\text{C})(T_f - 22)^{\circ}\text{C} = 0$$

It gives T_f = **78.6°C**

where T_f is the final equilibrium temperature in the room.

5-175 A room is to be heated by 1 ton of hot water contained in a tank placed in the room. The minimum initial temperature of the water is to be determined if it to meet the heating requirements of this room for a 25-h period.

Assumptions **1** Water is an incompressible substance with constant specific heats. **2** Air is an ideal gas with constant specific heats. **3** The energy stored in the container itself is negligible relative to the energy stored in water. **4** The room is maintained at 20°C at all times. **5** The hot water is to meet the heating requirements of this room for a 25-h period.

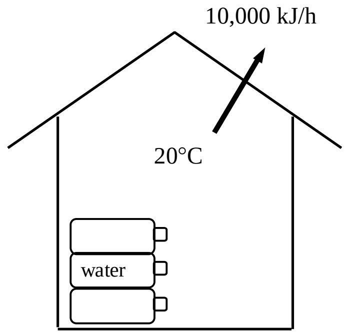
Properties The specific heat of water at room temperature is C = 4.18 kJ/kg·°C (Table A-3).

Analysis Heat loss from the room during a 25-h period is

$$Q_{\text{loss}} = (10{,}000 \text{ kJ/h})(24 \text{ h}) = 240{,}000 \text{ kJ}$$

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

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



or

$$-Q_{\text{out}} = [mC(T_2 - T_1)]_{\text{water}}$$

Substituting,

$$-240{,}000 \text{ kJ} = (1000 \text{ kg})(4.18 \text{ kJ/kg·°C})(20 - T_1)$$

It gives

$$T_1 = \mathbf{77.4°C}$$

where T_1 is the temperature of the water when it is first brought into the room.

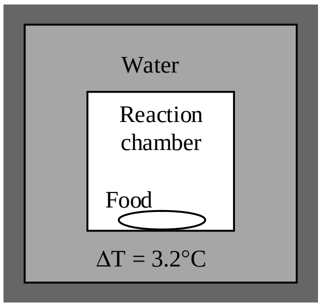
5-176 A sample of a food is burned in a bomb calorimeter, and the water temperature rises by 3.2°C when equilibrium is established. The energy content of the food is to be determined. √

Assumptions **1** Water is an incompressible substance with constant specific heats. **2** Air is an ideal gas with constant specific heats. **3** The energy stored in the reaction chamber is negligible relative to the energy stored in water. **4** The energy supplied by the mixer is negligible.

Properties The specific heat of water at room temperature is $C = 4.18$ kJ/kg·°C (Table A-3). The constant volume specific heat of air at room temperature is $C_v = 0.718$ kJ/kg·°C (Table A-2).

Analysis The chemical energy released during the combustion of the sample is transferred to the water as heat. Therefore, disregarding the change in the sensible energy of the reaction chamber, the energy content of the food is simply the heat transferred to the water. Taking the water as our system, the energy balance can be written as

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



or $$Q_{in} = (\Delta U)_{\text{water}} = [mC(T_2 - T_1)]_{\text{water}}$$

Substituting ,

$$Q_{\text{in}} = (3\text{ kg})(4.18\text{ kJ/kg·°C})(3.2\text{°C}) = 40.13\text{ kJ}$$

for a 2-g sample. Then the energy content of the food per unit mass is

$$\frac{40.13\text{ kJ}}{2\text{ g}}\left(\frac{1000\text{ g}}{1\text{ kg}}\right) = \mathbf{20{,}060\ kJ/kg}$$

To make a rough estimate of the error involved in neglecting the thermal energy stored in the reaction chamber, we treat the entire mass within the chamber as air and determine the change in sensible internal energy:

$$(\Delta U)_{\text{chamber}} = [mC_v(T_2 - T_1)]_{\text{chamber}} = (0.102\text{ kg})(0.718\text{ kJ/kg·}^\circ\text{C})(3.2^\circ\text{C}) = 0.23\text{ kJ}$$

which is less than 1% of the internal energy change of water. Therefore, it is reasonable to disregard the change in the sensible energy content of the reaction chamber in the analysis.

5-177 A man drinks one liter of cold water at 3°C in an effort to cool down. The drop in the average body temperature of the person under the influence of this cold water is to be determined.

Assumptions **1** Thermal properties of the body and water are constant. **2** The effect of metabolic heat generation and the heat loss from the body during that time period are negligible.

Properties The density of water is very nearly 1 kg/L, and the specific heat of water at room temperature is C = 4.18 kJ/kg·°C (Table A-3). The average specific heat of human body is given to be 3.6 kJ/kg.°C.

Analysis. The mass of the water is

$$m_w = \rho V = (1\text{ kg/L})(1\text{ L}) = 1\text{ kg}$$

We take the man and the water as our system, and disregard any heat and mass transfer and chemical reactions. Of course these assumptions may be acceptable only for very short time periods, such as the time it takes to drink the water. Then the energy balance can be written as

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

$$0 = \Delta U$$

$$0 = \Delta U_{\text{body}} + \Delta U_{\text{water}}$$

$$[mC_v(T_2 - T_1)]_{\text{body}} + [mC_v(T_2 - T_1)]_{\text{water}} = 0$$

Substituting,

$$(68\text{ kg})(3.6\text{ kJ/kg}\cdot^{\circ}\text{C})(T_f - 39)^{\circ}\text{C} + (1\text{ kg})(4.18\text{ kJ/kg}\cdot^{\circ}\text{C})(T_f - 3)^{\circ}\text{C} = 0$$

It gives

T_f = **38.4°C**

Therefore, the average body temperature of this person should drop about half a degree Celsius.

5-178 A 0.2-L glass of water at 20°C is to be cooled with ice to 5°C. The amount of ice or cold water that needs to be added to the water is to be determined.

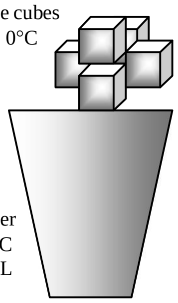
Assumptions **1** Thermal properties of the ice and water are constant. **2** Heat transfer to the glass is negligible. **3** There is no stirring by hand or a mechanical device (it will add energy).

Properties The density of water is 1 kg/L, and the specific heat of water at room temperature is C = 4.18 kJ/kg·°C (Table A-3). The specific heat of ice at about 0°C is C = 2.11 kJ/kg·°C (Table A-3). The melting temperature and the heat of fusion of ice at 1 atm are 0°C and 333.7 kJ/kg,.

Analysis (*a*) The mass of the water is

$$m_w = \rho V = (1\,\text{kg/L})(0.2\,\text{L}) = 0.2\text{kg}$$

We take the ice and the water as our system, and disregard any heat and mass transfer. This is a reasonable assumption since the time period of the process is very short. Then the energy balance can be written as



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

$$0 = \Delta U$$

$$0 = \Delta U_{\text{ice}} + \Delta U_{\text{water}}$$

$$[mC(0^\circ\text{C} - T_1)_{\text{solid}} + mh_{if} + mC(T_2 - 0^\circ\text{C})_{\text{liquid}}]_{\text{ice}} + [mC(T_2 - T_1)]_{\text{water}} = 0$$

Noting that $T_{1,\text{ ice}}$ = 0°C and T_2 = 5°C and substituting gives

$$m[0 + 333.7\text{ kJ/kg} + (4.18\text{ kJ/kg·°C})(5\text{-}0)\text{°C}] + (0.2\text{ kg})(4.18\text{ kJ/kg·°C})(5\text{-}20)\text{°C} = 0$$

$$m = 0.0354\text{ kg} = \mathbf{35.4\ g}$$

(*b*) When $T_{1,\text{ ice}}$ = -8°C instead of 0°C, substituting gives

$$m[(2.11\text{ kJ/kg·°C})[0\text{-}(\text{-}8)]\text{°C} + 333.7\text{ kJ/kg} + (4.18\text{ kJ/kg·°C})(5\text{-}0)\text{°C}]$$
$$+ (0.2\text{ kg})(4.18\text{ kJ/kg·°C})(5\text{-}20)\text{°C} = 0$$

$$m = 0.0338\text{ kg} = \mathbf{33.8\ g}$$

Cooling with cold water can be handled the same way. All we need to do is replace the terms for ice by a term for cold water at 0°C:

$$(\Delta U)_{\text{cold water}} + (\Delta U)_{\text{water}} = 0$$

$$[mC(T_2 - T_1)]_{\text{cold water}} + [mC(T_2 - T_1)]_{\text{water}} = 0$$

Substituting,

$$[m_{\text{cold water}}\,(4.18\text{ kJ/kg·°C})(5 - 0)\text{°C}] + (0.2\text{ kg})(4.18\text{ kJ/kg·°C})(5\text{-}20)\text{°C} = 0$$

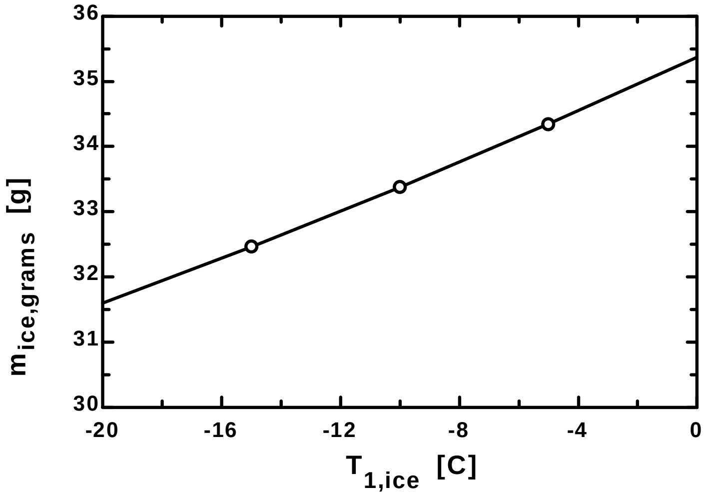
It gives m = 0.6 kg = **600 g**

Discussion Note that this is 17 times the amount of ice needed, and it explains why we use ice instead of water to cool drinks. Also, the temperature of ice does not seem to make a significant difference.

5-179 Problem 5-178 is reconsidered. The effect of the initial temperature of the ice on the final mass of ice required as the ice temperature varies from -20°C to 0°C is to be investigated. The mass of ice is to be plotted against the initial temperature of ice.

```
"Knowns"
rho_water = 1"[kg/L]"
V = 0.2 "[L]"
T_1_ice = 0"[C]"
T_1 = 20"[C]"
T_2 = 5"[C]"
C_ice = 2.11 "[kJ/kg-C]"
C_water = 4.18 "[kJ/kg-C]"
h_if = 333.7 "[kJ/kg]"
T_1_ColdWater = 0"[C]"

"The mass of the water is:"
m_water = rho_water*V "[kg]"

"The system is the water plus the ice.  Assume a short time period and neglect
any heat  and mass transfer.  The energy balance becomes:"
E_in - E_out = DELTAE_sys "[kJ]"
E_in = 0 "[kJ]"
E_out = 0"[kJ]"

DELTAE_sys = DELTAU_water+DELTAU_ice"[kJ]"

DELTAU_water = m_water*C_water*(T_2 - T_1)"[kJ]"
DELTAU_ice = DELTAU_solid_ice+DELTAU_melted_ice"[kJ]"
DELTAU_solid_ice =m_ice*C_ice*(0-T_1_ice) + m_ice*h_if"[kJ]"
DELTAU_melted_ice=m_ice*C_water*(T_2 - 0)"[kJ]"
m_ice_grams=m_ice*convert(kg,g)"[g]"

"Cooling with Cold Water:"
DELTAE_sys = DELTAU_water+DELTAU_ColdWater"[kJ]"
DELTAU_water = m_water*C_water*(T_2_ColdWater - T_1)"[kJ]"
DELTAU_ColdWater = m_ColdWater*C_water*(T_2_ColdWater - T_1_ColdWater)"[kJ]"
m_ColdWater_grams=m_ColdWater*convert(kg,g)"[g]"
```

$m_{ice,grams}$ [g]	$T_{1,ice}$ [C]
31.6	-20
32.47	-15
33.38	-10
34.34	-5
35.36	0



5-180 A 1- ton (1000 kg) of water is to be cooled in a tank by pouring ice into it. The final equilibrium temperature in the tank is to be determined.

Assumptions **1** Thermal properties of the ice and water are constant. **2** Heat transfer to the water tank is negligible. **3** There is no stirring by hand or a mechanical device (it will add energy).

Properties The specific heat of water at room temperature is C = 4.18 kJ/kg·°C, and the specific heat of ice at about 0°C is C = 2.11 kJ/kg·°C (Table A-3). The melting temperature and the heat of fusion of ice at 1 atm are 0°C and 333.7 kJ/kg.

Analysis We take the ice and the water as our system, and disregard any heat transfer between the system and the surroundings. Then the energy balance for this process can be written as

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

$$0 = \Delta U$$

$$0 = \Delta U_{\text{ice}} + \Delta U_{\text{water}}$$

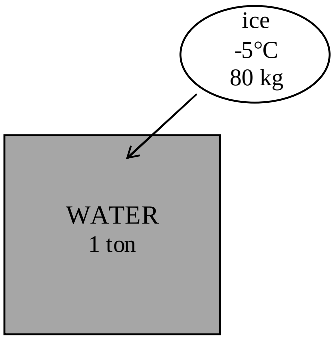
$$[mC(0^{\circ}\text{C} - T_1)_{\text{solid}} + mh_{if} + mC(T_2 - 0^{\circ}\text{C})_{\text{liquid}}]_{\text{ice}} + [mC(T_2 - T_1)]_{\text{water}} = 0$$

Substituting,

$$(80\text{ kg})\{(2.11\text{ kJ/kg}\cdot^{0}\text{C})[0-(\text{-}5)]^{0}\text{C} + 333.7\text{ kJ/kg} + (4.18\text{ kJ/kg}\cdot^{0}\text{C})(T_2 - 0)^{0}\text{C}\}$$
$$+(1000\text{ kg})(4.18\text{ kJ/kg}\cdot^{0}\text{C})(T_2 - 20)^{0}\text{C} = 0$$

It gives T_2 = **12.4°C**

which is the final equilibrium temperature in the tank.



5-181 An insulated cylinder initially contains a saturated liquid-vapor mixture of water at a specified temperature. The entire vapor in the cylinder is to be condensed isothermally by adding ice inside the cylinder. The amount of ice that needs to be added is to be determined.

Assumptions **1** Thermal properties of the ice are constant. **2** The cylinder is well-insulated and thus heat transfer is negligible. **3** There is no stirring by hand or a mechanical device (it will add energy).

Properties The specific heat of ice at about 0°C is C = 2.11 kJ/kg·°C (Table A-3). The melting temperature and the heat of fusion of ice at 1 atm are given to be 0°C and 333.7 kJ/kg.

Analysis (*a*) We take the contents of the cylinder (ice and saturated water) as our system, which is a closed system. Noting that the temperature and thus the pressure remains constant during this phase change process and thus $W_b + \Delta U = \Delta H$, the energy balance for this system can be written as

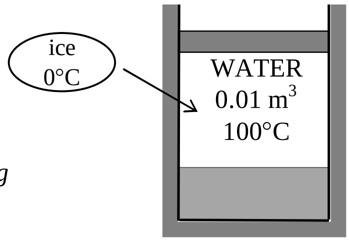
$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{b,in} = \Delta U \quad \rightarrow \quad \Delta H = 0$$

$$\Delta H_{\text{ice}} + \Delta H_{\text{water}} = 0$$

$$[mC(0^0\text{C} - T_1)_{\text{solid}} + mh_{if} + mC(T_2 - 0^0\text{C})_{\text{liquid}}]_{\text{ice}} + [m(h_2 - h_1)]_{\text{water}} = 0$$

The properties of water at 100°C are (Table A-4)



$$v_f = 0.001044, \quad v_g = 1.6729 \text{ m}^3/\text{kg}$$
$$h_f = 419.04, \quad h_{fg} = 2257.0 \text{ kJ.kg}$$

Then,

$$v_1 = v_f + x_1 v_{fg} = 0.001044 + 0.2 \times (1.6729 - 0.001044) = 0.3354 m^3 / kg$$

$$h_1 = h_f + x_1 h_{fg} = 419.04 + 0.2 \times 2257.0 = 870.4 kJ / kg$$

$$h_2 = h_{f@100°\text{C}} = 419.04 \text{ kJ/kg}$$

$$m_{steam} = \frac{V_1}{v_1} = \frac{0.01 \, m^3}{0.3354 \, m^3/kg} = 0.0298 \, kg$$

Noting that $T_{1,\text{ ice}}$ = 0°C and T_2 = 100°C and substituting gives

m[0 + 333.7 kJ/kg + (4.18 kJ/kg·°C)(100-0)°C] + (0.0298 kg)(419.04 – 870.4) kJ/kg = 0

m = 0.0179 kg = **17.9 g ice**

5-182 The cylinder of a steam engine initially contains saturated vapor of water at 100 kPa. The cylinder is cooled by pouring cold water outside of it, and some of the steam inside condenses. If the piston is stuck at its initial position, the friction force acting on the piston and the amount of heat transfer are to be determined.

Assumptions The device is air-tight so that no air leaks into the cylinder as the pressure drops.

Analysis We take the contents of the cylinder (the saturated liquid-vapor mixture) as the system, which is a closed system. Noting that the volume remains constant during this phase change process, the energy balance for this system can be expressed as

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

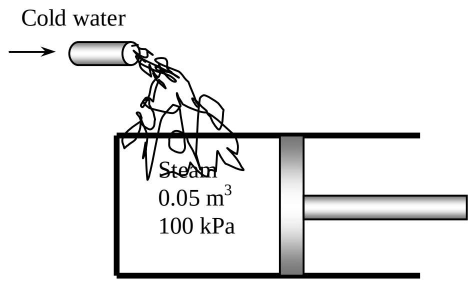
$$-Q_{out} = \Delta U = m(u_2 - u_1)$$

The saturation properties of water at 100 kPa and at 30°C are (Tables A-4 and A-5)

$$P_1 = 100 \text{ kPa} \longrightarrow \begin{array}{l} v_f = 0.001043 \text{ m}^3/\text{kg}, \quad v_g = 1.6940 \text{ m}^3/\text{kg} \\ u_f = 417.36 \text{ kJ/kg}, \quad u_g = 2506.1 \text{ kJ/kg} \end{array}$$

$$T_2 = 30^\circ\text{C} \longrightarrow \begin{array}{l} v_f = 0.001004 \text{ m}^3/\text{kg}, v_g = 32.89 \text{ m}^3/\text{kg} \\ u_f = 125.78 \text{ kJ/kg}, \quad u_{fg} = 2290.8 \text{ kJ/kg} \\ P_{sat} = 4.246 \text{ kPa} \end{array}$$

Then,



$$P_2 = P_{sat @ 30^\circ\text{C}} = 4.246 \text{ kPa}$$

$$v_1 = v_{g @ 100 \text{ kPa}} = 1.694 \text{ m}^3/\text{kg}$$

$$u_1 = u_{g @ 100 \text{ kPa}} = 2506.1 \text{ kJ/kg}$$

and

$$m = \frac{V_1}{v_1} = \frac{0.05 \text{m}^3}{1.6940 \text{m}^3/\text{kg}} = 0.0295 \text{kg}$$

$$v_2 = v_1 \longrightarrow x_2 = \frac{v_2 - v_f}{v_{fg}} = \frac{1.694 - 0.001}{32.89 - 0.001} = 0.05148$$

$$u_2 = u_f + x_2 u_{fg} = 125.78 + 0.05148 \times 2290.8 = 243.7 \text{kJ/kg}$$

The friction force that develops at the piston-cylinder interface balances the force acting on the piston, and is equal to

$$F = A(P_1 - P_2) = (0.1 \text{ m}^3)(100 - 4.246)\text{kPa}\left(\frac{1000 \text{ N/m}^2}{1 \text{ kPa}}\right) = \mathbf{9575 \ N}$$

The heat transfer is determined from the energy balance to be

$$\begin{aligned} Q_{out} &= m(u_1 - u_2) \\ &= (0.0295 \text{ kg})(2506.1 - 243.7)\text{kJ/kg} \\ &= \mathbf{66.7 \ kJ} \end{aligned}$$

5-183 Water is boiled at sea level (1 atm pressure) in a coffee maker, and half of the water evaporates in 25 min. The power rating of the electric heating element and the time it takes to heat the cold water to the boiling temperature are to be determined. √

Assumptions **1** The electric power consumption by the heater is constant. **2** Heat losses from the coffee maker are negligible.

Properties The enthalpy of vaporization of water at the saturation temperature of 100°C is h_{fg} = 2257 kJ/kg (Table A-4). At an average temperature of (100+18)/2 = 59°C, the specific heat of water is C = 4.18 kJ/kg.°C, and the density is about 1 kg/L (Table A-3).

Analysis The density of water at room temperature is very nearly 1 kg/L, and thus the mass of 1 L water at 18°C is nearly 1 kg. Noting that the enthalpy of vaporization represents the amount of energy needed to vaporize a liquid at a specified temperature, the amount of electrical energy needed to vaporize 0.5 kg of water in 25 min is

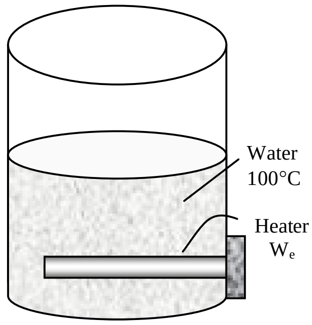
$$W_e = \dot{W}_e \Delta t = m h_{fg} \rightarrow \dot{W}_e = \frac{m h_{fg}}{\Delta t} = \frac{(0.5\text{ kg})(2257\text{ kJ/kg})}{(25\times 60\text{ s})} = \mathbf{0.752\ kW}$$

Therefore, the electric heater consumes (and transfers to water) 0.752 kW of electric power.

Noting that the specific heat of water at the average temperature of (18+100)/2 = 59°C is C = 4.18 kJ/kg·°C, the time it takes for the entire water to be heated from 18°C to 100°C is determined to be

$$W_e = \dot{W}_e \Delta t = mC\Delta T \rightarrow \Delta t = \frac{mC\Delta T}{\dot{W}_e} = \frac{(1\text{ kg})(4.18\text{ kJ/kg}\cdot{}^\circ\text{C})(100-18)^\circ\text{C}}{0.752\text{ kJ/s}} = 456\text{ s} = \mathbf{7.60\ min}$$

Discussion We can also solve this problem using v_f data (instead of density), and h_f data instead of specific heat. At 100°C, we have v_f = 0.001044 m^3/kg and h_f = 419.04 kJ/kg. At 18°C, we have h_f = 75.57 kJ/kg (Table A-4). The two results will be practically the same.



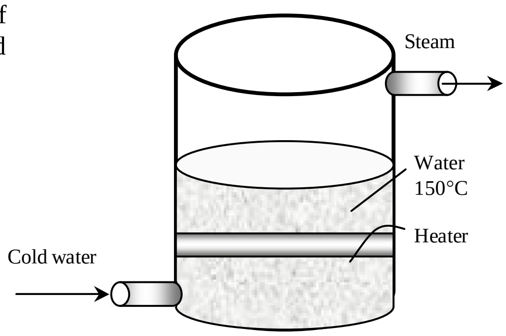
5-184 Water is boiled at a specified temperature by hot gases flowing through a stainless steel pipe submerged in water. The rate of evaporation of is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat losses from the outer surfaces of the boiler are negligible.

Properties The enthalpy of vaporization of water at 150°C is h_{fg} = 2114.3 kJ/kg (Table A-4).

Analysis The rate of heat transfer to water is given to be 74 kJ/s. Noting that the enthalpy of vaporization represents the amount of energy needed to vaporize a unit mass of a liquid at a specified temperature, the rate of evaporation of water is determined to be

$$\dot{m}_{\text{evaporation}} = \frac{\dot{Q}_{\text{boiling}}}{h_{fg}} = \frac{74\text{ kJ/s}}{2114.3\text{ kJ/kg}} = \mathbf{0.0350\ kg/s}$$



5-185 Cold water enters a steam generator at 20°C, and leaves as saturated vapor at T_{sat} = 100°C. The fraction of heat used to preheat the liquid water from 20°C to saturation temperature of 100°C is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat losses from the steam generator are negligible. **3** The specific heat of water is constant at the average temperature.

Properties The heat of vaporization of water at 100°C is h_{fg} = 2257 kJ/kg (Table A-4), and the specific heat of liquid water at the average temperature of (20+100)/2 = 60°C is C = 4.18 kJ/kg.°C (Table A-3).

Analysis The heat of vaporization of water represents the amount of heat needed to vaporize a unit mass of liquid at a specified temperature. Using the average specific heat, the amount of heat transfer needed to preheat a unit mass of water from 20°C to 100°C is

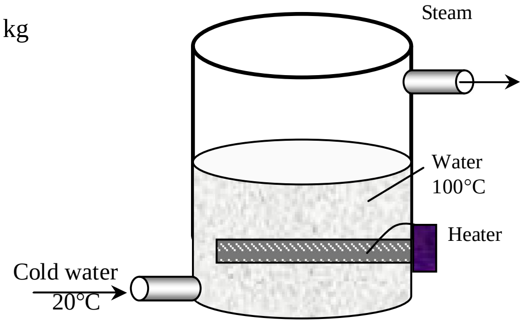
$$q_{\text{preheating}} = C\Delta T = (4.18\ \text{kJ/kg}\cdot°\text{C})(100-20)°\text{C} = 334.4\ \text{kJ/kg}$$

and

$$q_{\text{total}} = q_{\text{boiling}} + q_{\text{preheating}} = 2257 + 334.4 = 2591.4\ \text{kJ/kg}$$

Therefore, the fraction of heat used to preheat the water is

$$\text{Fraction to preheat} = \frac{q_{\text{preheating}}}{q_{\text{total}}} = \frac{334.4}{2591.4} = \mathbf{0.129}\ \ (\text{or } \mathbf{12.9\%})$$



5-186 Cold water enters a steam generator at 20°C and is boiled, and leaves as saturated vapor at boiler pressure. The boiler pressure at which the amount of heat needed to preheat the water to saturation temperature that is equal to the heat of vaporization is to be determined.

Assumptions Heat losses from the steam generator are negligible.

Properties The enthalpy of liquid water at 20°C is 83.96 kJ/kg. Other properties needed to solve this problem are the heat of vaporization h_{fg} and the enthalpy of saturated liquid at the specified temperatures, and they can be obtained from Table A-4.

Analysis The heat of vaporization of water represents the amount of heat needed to vaporize a unit mass of liquid at a specified temperature, and Δh represents the amount of heat needed to preheat a unit mass of water from 20°C to the saturation temperature. Therefore,

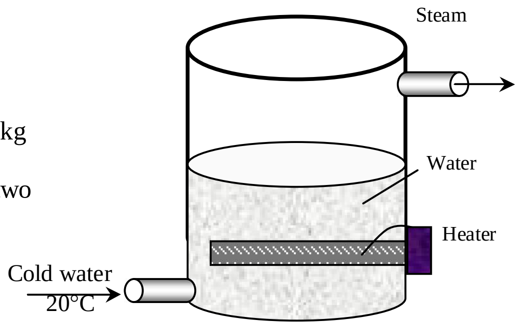
$$q_{\text{preheating}} = q_{\text{boiling}}$$
$$(h_{f@T_{\text{sat}}} - h_{f@20°\text{C}}) = h_{fg@T_{\text{sat}}}$$
$$h_{f@T_{\text{sat}}} - 83.96\ \text{kJ/kg} = h_{fg@T_{\text{sat}}} \quad \rightarrow \quad h_{f@T_{\text{sat}}} - h_{fg@T_{\text{sat}}} = 83.96\ \text{kJ/kg}$$

The solution of this problem requires choosing a boiling temperature, reading h_f and h_{fg} at that temperature, and substituting the values into the relation above to see if it is satisfied. By trial and error, (Table A-4)

At 310°C: $h_{f@T_{\text{sat}}} - h_{fg@T_{\text{sat}}} = 1401.3 - 1326 = 75.3$ kJ/kg

At 315°C: $h_{f@T_{\text{sat}}} - h_{fg@T_{\text{sat}}} = 1431.0 - 1283.5 = 147.5$ kJ/kg

The temperature that satisfies this condition is determined from the two values above by interpolation to be 310.6°C. The saturation pressure corresponding to this temperature is **9.94 MPa.**



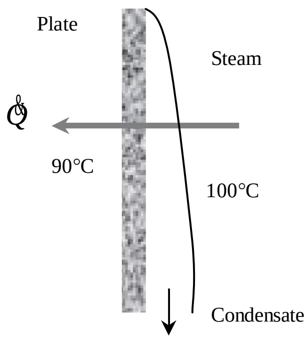
5-187 Saturated steam at 1 atm pressure and thus at a saturation temperature of T_{sat} = 100°C condenses on a vertical plate maintained at 90°C by circulating cooling water through the other side. The rate of condensation of steam is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The steam condenses and the condensate drips off at 100°C. (In reality, the condensate temperature will be between 90 and 100, and the cooling of the condensate a few °C should be considered if better accuracy is desired).

Properties The enthalpy of vaporization of water at 100°C is h_{fg} = 2257 kJ/kg (Table A-4).

Analysis The rate of heat transfer during this condensation process is given to be 180 kJ/s. Noting that the heat of vaporization of water represents the amount of heat released as a unit mass of vapor at a specified temperature condenses, the rate of condensation of steam is determined from

$$\dot{m}_{condensation} = \frac{\dot{Q}}{h_{fg}} = \frac{180\text{ kJ/s}}{2257\text{ kJ/kg}} = \mathbf{0.0798\ kg/s}$$



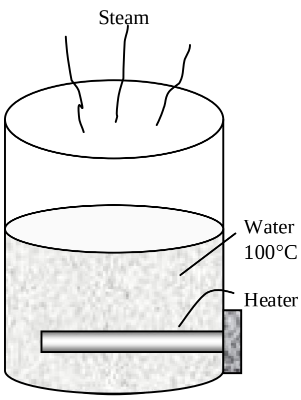
5-188 Water is boiled at T_{sat} = 100°C by an electric heater. The rate of evaporation of water is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat losses from the outer surfaces of the water tank are negligible.

Properties The enthalpy of vaporization of water at 100°C is h_{fg} = 2257 kJ/kg (Table A-4).

Analysis Noting that the enthalpy of vaporization represents the amount of energy needed to vaporize a unit mass of a liquid at a specified temperature, the rate of evaporation of water is determined to be

$$\dot{m}_{evaporation} = \frac{\dot{W}_{e,boiling}}{h_{fg}} = \frac{5\text{ kJ/s}}{2257\text{ kJ/kg}} = \mathbf{0.00222\ kg/s = 7.98\ kg/h}$$

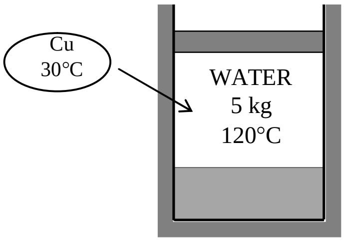


5-189 An insulated cylinder initially contains saturated liquid-vapor mixture of water at a specified temperature. A cold copper block is dropped into the cylinder. The final equilibrium temperature inside the cylinder and the final mass of water vapor are to be determined.

Assumptions **1** The copper block has a constant specific heat at the average temperature. **2** The cylinder is well-insulated and thus heat transfer is negligible. **3** Only part of the vapor in the cylinder condenses as a result of copper block being dropped into the cylinder, so that the temperature inside the cylinder remains constant (will be verified).

Properties The specific heat of copper at the average temperature of (30+120)/2 = 75°C is C = 0.391 kJ/kg·°C (Table A-3). The enthalpy of vaporization of water at 120°C is h_{fg} = 2202.6 kJ/kg (Table A-4).

Analysis We take the copper block as the system, which is closed system. Assuming the final temperature to be 120°C, the energy balance for copper can be expressed as



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

$$Q_{in} = \Delta U = [mC(T_2 - T_1)]_{\text{copper}}$$

Substituting,

$$Q_{\text{in}} = (5\text{ kg})(0.391\text{ kJ/kg.°C})(120 - 30)\text{°C} = 176.0\text{ kJ}$$

This energy will come from the water vapor inside. Noting that the enthalpy of vaporization represents the amount of energy released when a unit mass of vapor condenses at a specified temperature, the amount of vapor that condenses to release 176 kJ of heat is

$$m_{\text{condensed}} = \frac{Q}{h_{fg}} = \frac{176\text{ kJ}}{2202.6\text{ kJ/kg}} = 0.080\text{ kg}$$

Initially, the cylinder contained 1 kg of vapor. Then the amount of water vapor in the cylinder at the end of the process becomes

$$m_{\text{final}} = m_{\text{initial}} - m_{\text{condensed}} = 1.0 - 0.080 = \mathbf{0.920\ kg}$$

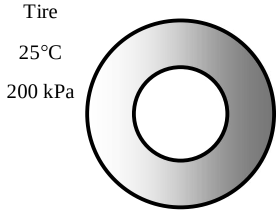
This verifies our assumption that the cylinder still contains some vapor at the end of the process. Then the final temperature in the cylinder must be the same as the initial temperature since saturation conditions still exist inside,

$$T_2 = T_1 = \mathbf{120°C}$$

5-190 The air pressure in a tire is measured before and after a trip. The temperature rise of air inside the tire is to be estimated.

Assumptions **1** Air is an ideal gas. **2** The volume of the tire remains constant. **3** No air leaks out of the tire during the trip.

Analysis Using the ideal gas relation between the two states, the final temperature in the tire is determined to be



$$\frac{P_1 V}{T_1} = \frac{P_2 V}{T_2} \quad \rightarrow \quad T_2 = \frac{P_2}{P_1} T_1 = \frac{(220+90)\text{ kPa}}{(200+90)\text{ kPa}}(25+273\text{ K}) = 318.6\text{ K} = 45.6\text{°C}$$

Therefore, the temperature rise of air in the tire during the trip is

$$\Delta T = T_2 - T_1 = 45.6 - 25 = \mathbf{20.6°C}$$

5-191 Two identical buildings in Los Angeles and Denver have the same infiltration rate. The ratio of the heat losses by infiltration at the two cities under identical conditions is to be determined.

Assumptions **1** Both buildings are identical and both are subjected to the same conditions except the atmospheric conditions. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Steady flow conditions exist.

Analysis We can view infiltration as a steady stream of air that is heated as it flows in an imaginary duct passing through the building. The energy balance for this imaginary steady-flow system can be expressed in the rate form as

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

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

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

$$\dot{Q}_{in} = \dot{m}C_p(T_2 - T_1) = \rho\dot{V}C_p(T_2 - T_1)$$

Los Angeles: 101 kPa
Denver: 83 kPa

Then the sensible infiltration heat loss (heat gain for the infiltrating air) can be expressed

$$\dot{Q}_{\text{infiltration}} = \dot{m}_{\text{air}} C_p (T_i - T_o) = \rho_{o,\text{air}} (ACH)(V_{\text{building}}) C_p (T_i - T_o)$$

where *ACH* is the infiltration volume rate in *air changes per hour*. Therefore, the infiltration heat loss is proportional to the density of air, and thus the ratio of infiltration heat losses at the two cities is simply the densities of outdoor air at those cities,

$$\text{Infiltration heat loss ratio} = \frac{\dot{Q}_{\text{infiltration, Los Angeles}}}{\dot{Q}_{\text{infiltration, Denver}}} = \frac{\rho_{o,\text{ air, Los Angeles}}}{\rho_{o,\text{ air, Denver}}}$$

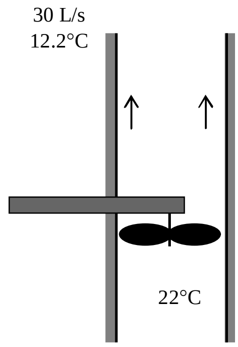
$$= \frac{(P_0 / RT_0)_{\text{Los Angeles}}}{(P_0 / RT_0)_{\text{Denver}}} = \frac{P_{o,\text{ Los Angeles}}}{P_{0,\text{ Denver}}}$$

$$= \frac{101\,\text{kPa}}{83\,\text{kPa}} = \mathbf{1.22}$$

Therefore, the infiltration heat loss in Los Angeles will be 22% higher than that in Denver under identical conditions.

5-192 The ventilating fan of the bathroom of an electrically heated building in San Francisco runs continuously. The amount and cost of the heat "vented out" per month in winter are to be determined.

Assumptions **1** We take the atmospheric pressure to be 1 atm = 101.3 kPa since San Francisco is at sea level. **2** The building is maintained at 22°C at all times. **3** The infiltrating air is heated to 22°C before it exfiltrates. **4** Air is an ideal gas with constant specific heats at room temperature. **5** Steady flow conditions exist.



Properties The gas constant of air is R = 0.287 kPa.m^3/kg·K (Table A-1). The specific heat of air at room temperature is C_p = 1.005 kJ/kg·°C (Table A-2).

Analysis The density of air at the indoor conditions of 1 atm and 22°C is

$$\rho_o = \frac{P_o}{RT_o} = \frac{(101.3\text{ kPa})}{(0.287\text{ kPa.m}^3/\text{kg.K})(22+273\text{ K})} = 1.20\text{ kg/m}^3$$

Then the mass flow rate of air vented out becomes

$$\dot{m}_{\text{air}} = \rho \dot{V}_{\text{air}} = (1.20\text{ kg/m}^3)(0.030\text{ m}^3/\text{s}) = 0.036\text{ kg/s}$$

We can view infiltration as a steady stream of air that is heated as it flows in an imaginary duct passing through the house. The energy balance for this imaginary steady-flow system can be expressed in the rate form as

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

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

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

$$\dot{Q}_{in} = \dot{m}C_p(T_2 - T_1)$$

Noting that the indoor air vented out at 22°C is replaced by infiltrating outdoor air at 12.2°C, the sensible infiltration heat loss (heat gain for the infiltrating air) due to venting by fans can be expressed

$$\begin{aligned}\dot{Q}_{\text{loss by fan}} &= \dot{m}_{\text{air}} C_{\text{p}} (T_{\text{indoors}} - T_{\text{outdoors}}) \\ &= (0.036\text{ kg/s})(1.005\text{ kJ/kg.°C})(22-12.2)\text{°C} = 0.355\text{ kJ/s} = 0.355\text{ kW}\end{aligned}$$

Then the amount and cost of the heat "vented out" per month (1 month = 30×24 = 720 h) becomes

$$\text{Energy loss} = \dot{Q}_{\text{loss by fan}} \Delta t = (0.355\text{ kW})(720\text{ h/month}) = \mathbf{256\ kWh/month}$$

$$\text{Money loss} = (\text{Energy loss})(\text{Unit cost of energy}) = (256\text{ kWh/month})(\$0.09/\text{kWh}) = \mathbf{\$23.0/month}$$

Discussion Note that the energy and money loss associated with ventilating fans can be very significant. Therefore, ventilating fans should be used with care.

5-193 Chilled air is to cool a room by removing the heat generated in a large insulated classroom by lights and students. The required flow rate of air that needs to be supplied to the room is to be determined.

Assumptions **1** The moisture produced by the bodies leave the room as vapor without any condensing, and thus the classroom has no latent heat load. **2** Heat gain through the walls and the roof is negligible. **4** Air is an ideal gas with constant specific heats at room temperature. **5** Steady operating conditions exist.

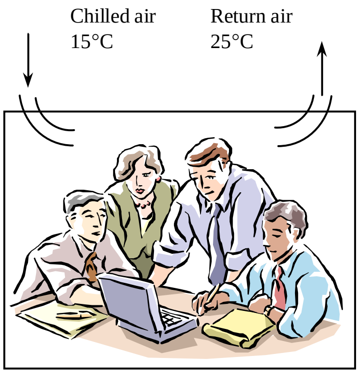
Properties The specific heat of air at room temperature is 1.005 kJ/kg·°C (Table A-2). The average rate of sensible heat generation by a person is given to be 60 W.

Analysis The rate of sensible heat generation by the people in the room and the total rate of sensible internal heat generation are

$$\dot{Q}_{\text{gen, sensible}} = \dot{q}_{\text{gen, sensible}}(\text{No. of people}) = (60\ \text{W/person})(150\ \text{persons}) = 9000\ \text{W}$$

$$\dot{Q}_{\text{total, sensible}} = \dot{Q}_{\text{gen, sensible}} + \dot{Q}_{\text{lighting}} = 9000 + 4000 = 13{,}000\ \text{W}$$

Both of these effects can be viewed as heat gain for the chilled air stream, which can be viewed as a steady stream of cool air that is heated as it flows in an imaginary duct passing through the room. The energy balance for this imaginary steady-flow system can be expressed in the rate form as



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

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

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

$$\dot{Q}_{in} = \dot{Q}_{\text{total, sensible}} = \dot{m}C_p(T_2 - T_1)$$

Then the required mass flow rate of chilled air becomes

$$\dot{m}_{\text{air}} = \frac{\dot{Q}_{\text{total, sensible}}}{C_p \Delta T} = \frac{13\ \text{kJ/s}}{(1.005\ \text{kJ/kg}\cdot{}^\circ\text{C})(25-15)^\circ\text{C}} = \mathbf{1.29\ kg/s}$$

Discussion The latent heat will be removed by the air-conditioning system as the moisture condenses outside the cooling coils.

5-194 Chickens are to be cooled by chilled water in an immersion chiller. The rate of heat removal from the chicken and the mass flow rate of water are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The thermal properties of chickens and water are constant.

Properties The specific heat of chicken are given to be 3.54 kJ/kg.°C. The specific heat of water is 4.18 kJ/kg.°C (Table A-3).

Analysis (*a*) Chickens are dropped into the chiller at a rate of 500 per hour. Therefore, chickens can be considered to flow steadily through the chiller at a mass flow rate of

$$\dot{m}_{\text{chicken}} = (500\text{ chicken / h})(2.2\text{ kg / chicken}) = 1100\text{ kg / h} = 0.3056\text{ kg / s}$$

Taking the chicken flow stream in the chiller as the system, the energy balance for steadily flowing chickens can be expressed in the rate form as

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

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

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

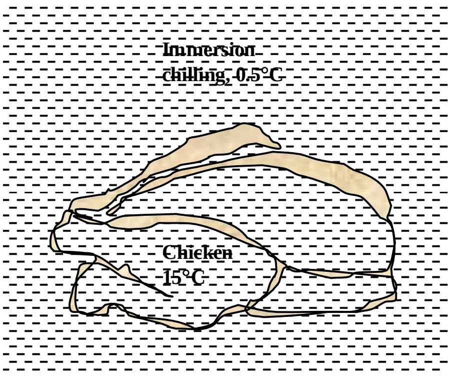
$$\dot{Q}_{out} = \dot{Q}_{\text{chicken}} = \dot{m}_{\text{chicken}} C_p (T_1 - T_2)$$

Then the rate of heat removal from the chickens as they are cooled from 15°C to 3ºC becomes

$$\dot{Q}_{\text{chicken}} = (\dot{m}C_p \Delta T)_{\text{chicken}} = (0.3056\text{ kg/s})(3.54\text{ kJ/kg.}^\circ\text{C})(15-3)^\circ\text{C} = \mathbf{13.0\text{ kW}}$$

The chiller gains heat from the surroundings at a rate of 200 kJ/h = 0.0556 kJ/s. Then the total rate of heat gain by the water is

$$\dot{Q}_{\text{water}} = \dot{Q}_{\text{chicken}} + \dot{Q}_{\text{heat gain}} = 13.0 + 0.056 = 13.056\text{ kW}$$



Noting that the temperature rise of water is not to exceed 2ºC as it flows through the chiller, the mass flow rate of water must be at least

$$\dot{m}_{\text{water}} = \frac{\dot{Q}_{\text{water}}}{(C_p \Delta T)_{\text{water}}} = \frac{13.056\text{ kW}}{(4.18\text{ kJ/kg.}^\circ\text{C})(2^\circ\text{C})} = \mathbf{1.56\text{ kg/s}}$$

If the mass flow rate of water is less than this value, then the temperature rise of water will have to be more than 2°C.

5-195 Chickens are to be cooled by chilled water in an immersion chiller. The rate of heat removal from the chicken and the mass flow rate of water are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The thermal properties of chickens and water are constant. **3** Heat gain of the chiller is negligible.

Properties The specific heat of chicken are given to be 3.54 kJ/kg.°C. The specific heat of water is 4.18 kJ/kg.°C (Table A-3).

Analysis (*a*) Chickens are dropped into the chiller at a rate of 500 per hour. Therefore, chickens can be considered to flow steadily through the chiller at a mass flow rate of

$$\dot{m}_{\text{chicken}} = (500 \text{ chicken / h})(2.2 \text{ kg / chicken}) = 1100 \text{ kg / h} = 0.3056 \text{ kg / s}$$

Taking the chicken flow stream in the chiller as the system, the energy balance for steadily flowing chickens can be expressed in the rate form as

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

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

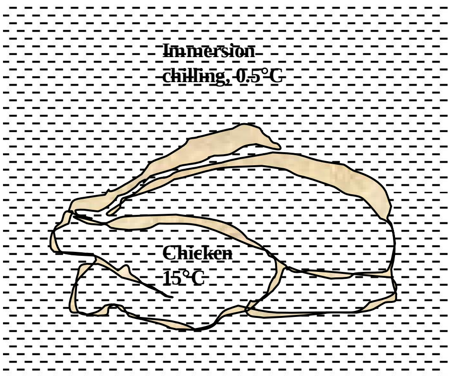
$$\dot{m}h_1 = \dot{Q}_{out} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{out} = \dot{Q}_{\text{chicken}} = \dot{m}_{\text{chicken}} C_p (T_1 - T_2)$$

Then the rate of heat removal from the chickens as they are cooled from 15°C to 3ºC becomes

$$\dot{Q}_{\text{chicken}} = (\dot{m}C_p \Delta T)_{\text{chicken}} = (0.3056 \text{ kg/s})(3.54 \text{ kJ/kg.}^\circ\text{C})(15-3)^\circ\text{C} = \mathbf{13.0\ kW}$$

Heat gain of the chiller from the surroundings is negligible. Then the total rate of heat gain by the water is



$$\dot{Q}_{\text{water}} = \dot{Q}_{\text{chicken}} = 13.0 \text{ kW}$$

Noting that the temperature rise of water is not to exceed 2ºC as it flows through the chiller, the mass flow rate of water must be at least

$$\dot{m}_{\text{water}} = \frac{\dot{Q}_{\text{water}}}{(C_p \Delta T)_{\text{water}}} = \frac{13.0 \text{ kW}}{(4.18 \text{ kJ/kg.}^\circ\text{C})(2^\circ\text{C})} = \mathbf{1.56\ kg/s}$$

If the mass flow rate of water is less than this value, then the temperature rise of water will have to be more than 2°C.

5-196 A regenerator is considered to save heat during the cooling of milk in a dairy plant. The amounts of fuel and money such a generator will save per year are to be determined.

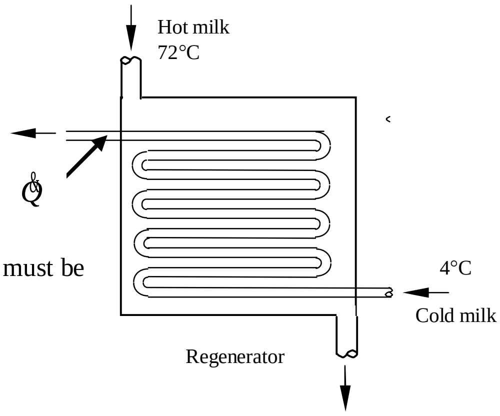
Assumptions 1 Steady operating conditions exist. **2** The properties of the milk are constant.

Properties The average density and specific heat of milk can be taken to be $\rho_{\text{milk}} \cong \rho_{\text{water}} = 1$ kg/L and $C_{p,\text{ milk}}$= 3.79 kJ/kg.°C (Table A-3).

Analysis The mass flow rate of the milk is

$$\dot m_{\text{milk}} = \rho \dot{V}_{\text{milk}} = (1\ \text{kg/L})(12\ \text{L/s}) = 12\ \text{kg/s} = 43{,}200\ \text{kg/h}$$

Taking the pasteurizing section as the system, the energy balance for this steady-flow system can be expressed in the rate form as



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

$$\dot E_{in} = \dot E_{out}$$

$$\dot Q_{in} + \dot m h_1 = \dot m h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot Q_{in} = \dot m_{\text{milk}} C_p (T_2 - T_1)$$

Therefore, to heat the milk from 4 to 72°C as being done currently, heat must be transferred to the milk at a rate of

$$\dot Q_{\text{current}} = [\dot m C_{\text{p}}(T_{\text{pasturization}} - T_{\text{refrigeration}})]_{\text{milk}}$$
$$= (12\ \text{kg/s})(3.79\ \text{kJ/kg.}^{\text{o}}\text{C})(72-4)^{\text{o}}\text{C} = 3093\ \text{kJ/s}$$

The proposed regenerator has an effectiveness of ε = 0.82, and thus it will save 82 percent of this energy. Therefore,

$$\dot Q_{\text{saved}} = \varepsilon \dot Q_{\text{current}} = (0.82)(3093\ \text{kJ/s}) = 2536\ \text{kJ/s}$$

Noting that the boiler has an efficiency of η_{boiler} = 0.82, the energy savings above correspond to fuel savings of

$$\text{Fuel Saved} = \frac{\dot Q_{\text{saved}}}{\eta_{\text{boiler}}} = \frac{(2536\ \text{kJ/s})}{(0.82)}\frac{(1\text{therm})}{(105{,}500\ \text{kJ})} = 0.02931\ \text{therm/s}$$

Noting that 1 year = 365×24=8760 h and unit cost of natural gas is \$0.52/therm, the annual fuel and money savings will be

Fuel Saved = (0.02931 therms/s)(8760×3600 s) = **924,450 therms/yr**

$$\begin{aligned}\text{Money saved } &= (\text{Fuel saved})(\text{Unit cost of fuel})\\ &= (924{,}450\ \text{therm/yr})(\$0.52/\text{therm}) = \mathbf{\$480{,}700\ /\ yr}\end{aligned}$$

5-197E A refrigeration system is to cool eggs by chilled air at a rate of 10,000 eggs per hour. The rate of heat removal from the eggs, the required volume flow rate of air, and the size of the compressor of the refrigeration system are to be determined. √

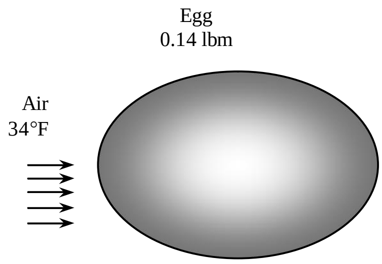
Assumptions **1** Steady operating conditions exist. **2** The eggs are at uniform temperatures before and after cooling. **3** The cooling section is well-insulated. **4** The properties of eggs are constant. **5** The local atmospheric pressure is 1 atm.

Properties The properties of the eggs are given to ρ = 67.4 lbm/ft^3 and C_p = 0.80 Btu/lbm.°F. The specific heat of air at room temperature C_p = 0.24 Btu/lbm. °F (Table A-2E). The gas constant of air is R = 0.3704 psia.ft^3/lbm.R (Table A-1E).

Analysis (*a*) Noting that eggs are cooled at a rate of 10,000 eggs per hour, eggs can be considered to flow steadily through the cooling section at a mass flow rate of

$$\dot{m}_{\text{egg}} = (10{,}000\,\text{eggs/h})(0.14\,\text{lbm/egg}) = 1400\,\text{lbm/h} = 0.3889\,\text{lbm/s}$$

Taking the egg flow stream in the cooler as the system, the energy balance for steadily flowing eggs can be expressed in the rate form as



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

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

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

$$\dot{Q}_{out} = \dot{Q}_{\text{egg}} = \dot{m}_{\text{egg}} C_p (T_1 - T_2)$$

Then the rate of heat removal from the eggs as they are cooled from 90°F to 50ºF at this rate becomes

$$\dot{Q}_{\text{egg}} = (\dot{m}C_p\Delta T)_{\text{egg}} = (1400\text{ lbm/h})(0.80\text{ Btu/lbm.}^\circ\text{F})(90-50)^\circ\text{F} = \mathbf{44{,}800\,Btu/h}$$

(*b*) All the heat released by the eggs is absorbed by the refrigerated air since heat transfer through he walls of cooler is negligible, and the temperature rise of air is not to exceed 10°F. The minimum mass flow and volume flow rates of air are determined to be

$$\dot{m}_{air} = \frac{\dot{Q}_{air}}{(C_p\Delta T)_{air}} = \frac{44{,}800\,\text{Btu/h}}{(0.24\,\text{Btu/lbm.}^\circ\text{F})(10^\circ\text{F})} = 18{,}667\,\text{lbm/h}$$

$$\rho_{air} = \frac{P}{RT} = \frac{14.7\text{ psia}}{(0.3704\,\text{psia.ft}^3/\text{lbm.R})(34+460)\text{R}} = 0.0803\,\text{lbm/ft}^3$$

$$\dot{V}_{air} = \frac{\dot{m}_{air}}{\rho_{air}} = \frac{18{,}667\text{ lbm/h}}{0.0803\ \ \text{lbm/ft}^3} = \mathbf{232{,}500\ ft^3/h}$$

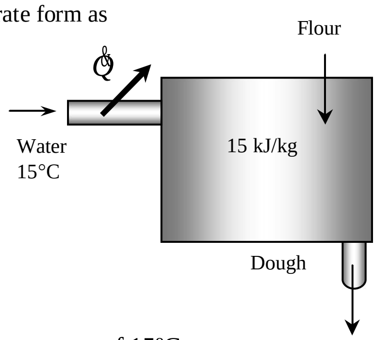
5-198 Dough is made with refrigerated water in order to absorb the heat of hydration and thus to control the temperature rise during kneading. The temperature to which the city water must be cooled before mixing with flour is to be determined to avoid temperature rise during kneading.

Assumptions **1** Steady operating conditions exist. **2** The dough is at uniform temperatures before and after cooling. **3** The kneading section is well-insulated. **4** The properties of water and dough are constant.

Properties The specific heats of the flour and the water are given to be 1.76 and 4.18 kJ/kg.°C, respectively. The heat of hydration of dough is given to be 15 kJ/kg.

Analysis It is stated that 2 kg of flour is mixed with 1 kg of water, and thus 3 kg of dough is obtained from each kg of water. Also, 15 kJ of heat is released for each kg of dough kneaded, and thus 3×15 = 45 kJ of heat is released from the dough made using 1 kg of water.

Taking the cooling section of water as the system, which is a steady-flow control volume, the energy balance for this steady-flow system can be expressed in the rate form as



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

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

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

$$\dot{Q}_{out} = \dot{Q}_{\text{water}} = \dot{m}_{\text{water}} C_p (T_1 - T_2)$$

In order for water to absorb all the heat of hydration and end up at a temperature of 15ºC, its temperature before entering the mixing section must be reduced to

$$Q_{in} = Q_{\text{dough}} = mC_p(T_2 - T_1) \quad \rightarrow \quad T_1 = T_2 - \frac{Q}{mC_p} = 15°\text{C} - \frac{45 \text{ kJ}}{(1 \text{ kg})(4.18 \text{ kJ/kg.°C})} = \mathbf{4.2°C}$$

That is, the water must be precooled to 4.2ºC before mixing with the flour in order to absorb the entire heat of hydration.

5-199 Glass bottles are washed in hot water in an uncovered rectangular glass washing bath. The rates of heat and water mass that need to be supplied to the water are to be determined. √

Assumptions **1** Steady operating conditions exist. **2** The entire water body is maintained at a uniform temperature of 55°C. **3** Heat losses from the outer surfaces of the bath are negligible. **4** Water is an incompressible substance with constant properties.

Properties The specific heat of water at room temperature is C_p = 4.18 kJ/kg.°C. Also, the specific heat of glass is 0.80 kJ/kg.°C (Table A-3).

Analysis (*a*) The mass flow rate of glass bottles through the water bath in steady operation is

$$\dot{m}_{\text{bottle}} = m_{\text{bottle}} \times \text{Bottle flow rate} = (0.150 \text{ kg / bottle})(800 \text{ bottles / min}) = 120 \text{ kg / min} = 2 \text{ kg / s}$$

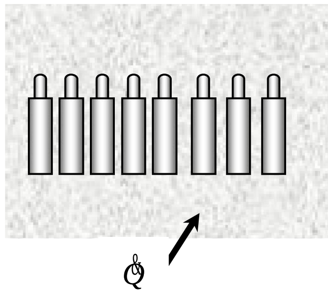
Taking the bottle flow section as the system, which is a steady-flow control volume, the energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

$$\dot{Q}_{in} = \dot{Q}_{\text{bottle}} = \dot{m}_{\text{water}} C_p (T_2 - T_1)$$



Then the rate of heat removal by the bottles as they are heated from 20 to 55°C is

$$\dot{Q}_{\text{bottle}} = \dot{m}_{\text{bottle}} C_p \Delta T = (2 \text{ kg/s})(0.8 \text{kJ/kg.}^\circ\text{C})(55 - 20)^\circ\text{C} = 56{,}000 \text{ W}$$

The amount of water removed by the bottles is

$$\dot{m}_{\text{water,out}} = (\text{Flow rate of bottles})(\text{Water removed per bottle})$$

$$= (800 \text{bottles / min})(0.2 \text{ g/bottle}) = 160 \text{ g/min} = \mathbf{2.67 \times 10^{-3} \text{ kg/s}}$$

Noting that the water removed by the bottles is made up by fresh water entering at 15°C, the rate of heat removal by the water that sticks to the bottles is

$$\dot{Q}_{\text{water removed}} = \dot{m}_{\text{water removed}} C_p \Delta T = (2.67 \times 10^{-3} \text{ kg/s})(4180 \text{ J/kg.}^\circ\text{C})(55 - 15)^\circ\text{C} = 446 \text{W}$$

Therefore, the total amount of heat removed by the wet bottles is

$$\dot{Q}_{\text{total, removed}} = \dot{Q}_{\text{glass removed}} + \dot{Q}_{\text{water removed}} = 56{,}000 + 446 = \mathbf{56{,}446 W}$$

Discussion In practice, the rates of heat and water removal will be much larger since the heat losses from the tank and the moisture loss from the open surface are not considered.

5-200 Glass bottles are washed in hot water in an uncovered rectangular glass washing bath. The rates of heat and water mass that need to be supplied to the water are to be determined. √

Assumptions **1** Steady operating conditions exist. **2** The entire water body is maintained at a uniform temperature of 50°C. **3** Heat losses from the outer surfaces of the bath are negligible. **4** Water is an incompressible substance with constant properties.

Properties The specific heat of water at room temperature is C_p = 4.18 kJ/kg.°C. Also, the specific heat of glass is 0.80 kJ/kg.°C (Table A-3).

Analysis (*a*) The mass flow rate of glass bottles through the water bath in steady operation is

$$\dot{m}_{\text{bottle}} = m_{\text{bottle}} \times \text{Bottle flow rate} = (0.150 \text{ kg / bottle})(800 \text{ bottles / min}) = 120 \text{ kg / min} = 2 \text{ kg / s}$$

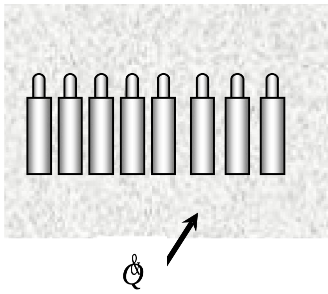
Taking the bottle flow section as the system, which is a steady-flow control volume, the energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

$$\dot{Q}_{in} = \dot{Q}_{\text{bottle}} = \dot{m}_{\text{water}} C_p (T_2 - T_1)$$



Then the rate of heat removal by the bottles as they are heated from 20 to 50°C is

$$\dot{Q}_{\text{bottle}} = \dot{m}_{\text{bottle}} C_p \Delta T = (2 \text{ kg/s})(0.8 \text{ kJ/kg.}^\circ\text{C})(50 - 20)^\circ\text{C} = 48{,}000 \text{ W}$$

The amount of water removed by the bottles is

$$\dot{m}_{\text{water,out}} = (\text{Flow rate of bottles})(\text{Water removed per bottle})$$
$$= (800 \text{ bottles / min})(0.2 \text{ g/bottle}) = 160 \text{ g/min} = \mathbf{2.67 \times 10^{-3} \text{ kg/s}}$$

Noting that the water removed by the bottles is made up by fresh water entering at 15°C, the rate of heat removal by the water that sticks to the bottles is

$$\dot{Q}_{\text{water removed}} = \dot{m}_{\text{water removed}} C_p \Delta T = (82.67 \times 10^{-3} \text{ kg/s})(4180 \text{ J/kg}\cdot^\circ\text{C})(50 - 15)^\circ\text{C} = 391 \text{ W}$$

Therefore, the total amount of heat removed by the wet bottles is

$$\dot{Q}_{\text{total, removed}} = \dot{Q}_{\text{glass removed}} + \dot{Q}_{\text{water removed}} = 48{,}000 + 391 = \mathbf{48{,}391 \text{ W}}$$

Discussion In practice, the rates of heat and water removal will be much larger since the heat losses from the tank and the moisture loss from the open surface are not considered.

5-201 Long aluminum wires are extruded at a velocity of 10 m/min, and are exposed to atmospheric air. The rate of heat transfer from the wire is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The thermal properties of the wire are constant.

Properties The properties of aluminum are given to be ρ = 2702 kg/m^3 and C_p = 0.896 kJ/kg.°C.

Analysis The mass flow rate of the extruded wire through the air is

$$\dot{m} = \rho\dot{V} = \rho(\pi r_0^2)V = (2702\,\text{kg/m}^3)\pi(0.0015\,\text{m})^2(10\,\text{m/min}) = 0.191\,\text{kg/min}$$

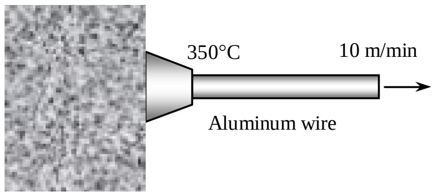
Taking the volume occupied by the extruded wire as the system, which is a steady-flow control volume, the energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

$$\dot{Q}_{out} = \dot{Q}_{\text{wire}} = \dot{m}_{\text{wire}}C_p(T_1 - T_2)$$



Then the rate of heat transfer from the wire to the air becomes

$$\dot{Q} = \dot{m}C_p[T(t) - T_\infty] = (0.191\text{ kg / min})(0.896\text{ kJ / kg.°C})(350 - 50)\text{°C} = 51.3\text{ kJ / min} = \mathbf{0.856\ kW}$$

5-202 Long copper wires are extruded at a velocity of 10 m/min, and are exposed to atmospheric air. The rate of heat transfer from the wire is to be determined. √

Assumptions **1** Steady operating conditions exist. **2** The thermal properties of the wire are constant.

Properties The properties of copper are given to be ρ = 8950 kg/m^3 and C_p = 0.383 kJ/kg.°C.

Analysis The mass flow rate of the extruded wire through the air is

$$\dot{m} = \rho\dot{V} = \rho(\pi r_0^2)V = (8950\,\text{kg/m}^3)\pi(0.0015\,\text{m})^2(10\,\text{m/min}) = 0.633\,\text{kg/min}$$

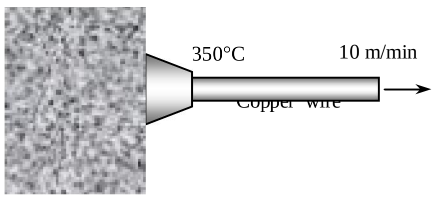
Taking the volume occupied by the extruded wire as the system, which is a steady-flow control volume, the energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

$$\dot{Q}_{out} = \dot{Q}_{\text{wire}} = \dot{m}_{\text{wire}}C_p(T_1 - T_2)$$



Then the rate of heat transfer from the wire to the air becomes

$$\dot{Q} = \dot{m}C_p[T(t) - T_\infty] = (0.633\text{ kg / min})(0.383\text{ kJ / kg.°C})(350 - 50)\text{°C} = 72.7\text{ kJ / min} = \mathbf{1.21\ kW}$$

5-203 Steam at a saturation temperature of T_{sat} = 40°C condenses on the outside of a thin horizontal tube. Heat is transferred to the cooling water that enters the tube at 25°C and exits at 35°C. The rate of condensation of steam is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Water is an incompressible substance with constant properties at room temperature. **3** The changes in kinetic and potential energies are negligible.

Properties The properties of water at room temperature are ρ = 997 kg/m^3 and C_p = 4.18 kJ/kg.°C. The enthalpy of vaporization of water at a saturation temperature of 40°C is h_{fg} = 2406.7 kJ/kg (Table A-4).

Analysis The mass flow rate of water through the tube is

$$\dot{m}_{\text{water}} = \rho V A_c = (997\ \text{kg/m}^3)(2\ \text{m/s})[\pi(0.03\ \text{m})^2/4] = 1.409\ \text{kg/s}$$

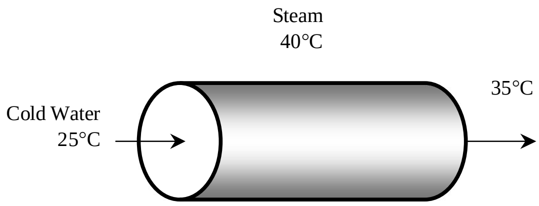
Taking the volume occupied by the cold water in the tube as the system, which is a steady-flow control volume, the energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

$$\dot{Q}_{in} = \dot{Q}_{\text{water}} = \dot{m}_{\text{water}} C_p (T_2 - T_1)$$



Then the rate of heat transfer to the water and the rate of condensation become

$$\dot{Q} = \dot{m}C_p(T_{out} - T_{in}) = (1.409\ \text{kg/s})(4.18\ \text{kJ/kg}\cdot°\text{C})(35-25)°\text{C} = 58.9\ \text{kW}$$

$$\dot{Q} = \dot{m}_{\text{evap}} h_{fg} \rightarrow \dot{m}_{\text{evap}} = \frac{\dot{Q}}{h_{fg}} = \frac{58.9\ \text{kJ/s}}{2406.7\ \text{kJ/kg}} = \mathbf{0.0245\ kg/s}$$

5-204E Saturated steam at a saturation pressure of 0.95 psia and thus at a saturation temperature of T_{sat} = 100°F (Table A-4E) condenses on the outer surfaces of 144 horizontal tubes by circulating cooling water arranged in a 12 × 12 square array. The rate of heat transfer to the cooling water and the average velocity of the cooling water are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The tubes are isothermal. **3** Water is an incompressible substance with constant properties at room temperature. **4** The changes in kinetic and potential energies are negligible.

Properties The properties of water at room temperature are ρ = 62.1 lbm/ft^3 and C_p = 1.00 Btu/lbm.°F. The enthalpy of vaporization of water at a saturation temperature of 100°F is h_{fg} = 1037 Btu/lbm (Table A-4E).

Analysis The rate of heat transfer from the steam to the cooling water is equal to the heat of vaporization released as the vapor condenses at the specified temperature,

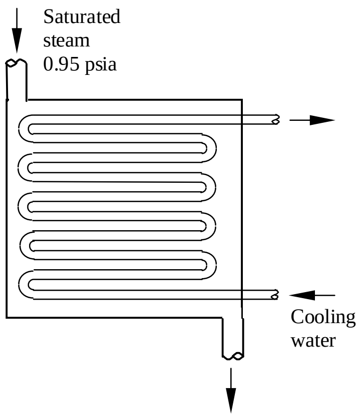
$$\dot{Q} = \dot{m}h_{fg} = (6800 \text{ lbm/h})(1037 \text{ Btu/lbm}) = \mathbf{7{,}051{,}600\ Btu/h = 1959\ Btu/s}$$

All of this energy is transferred to the cold water. Therefore, the mass flow rate of cold water must be

$$\dot{Q} = \dot{m}_{\text{water}} C_p \Delta T \quad\rightarrow\quad \dot{m}_{\text{water}} = \frac{\dot{Q}}{C_p \Delta T} = \frac{1959 \text{ Btu/s}}{(1.00 \text{ Btu/lbm.°F})(8\text{°F})} = 244.8 \text{ lbm/s}$$

Then the average velocity of the cooling water through the 144 tubes becomes

$$\dot{m} = \rho A \mathbf{V} \quad\rightarrow\quad \mathbf{V} = \frac{\dot{m}}{\rho A} = \frac{\dot{m}}{\rho(n\pi D^2/4)} = \frac{244.8 \text{ lbm/s}}{(62.1 \text{ lbm/ft}^3)[144\pi(2/12 \text{ ft})^2/4]} = \mathbf{1.26\ ft/s}$$



5-205 Saturated refrigerant-134a vapor at a saturation temperature of T_{sat} = 30°C condenses inside a tube. The rate of heat transfer from the refrigerant for the condensate exit temperatures of 16°C and 20°C are to be determined.

Assumptions **1** Steady flow conditions exist. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions involved.

Properties The properties of saturated refrigerant-134a at 30°C are h_f = 91.49 kJ/kg, are h_g = 263.50 kJ/kg, and are h_{fg} = 172.00 kJ/kg. The enthalpy of saturated liquid refrigerant at 16°C is h_f = 71.69 kJ/kg, (Table A-11).

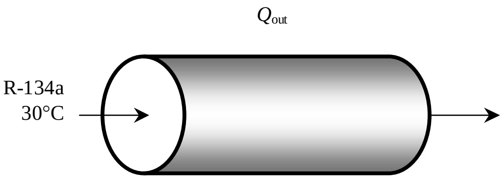
Analysis We take the *tube and the refrigerant in it* as the system. This is a *control volume* since mass crosses the system boundary during the process. We note that there is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Noting that heat is lost from the system, the energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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



where at the inlet state $h_1 = h_g$ = 263.50 kJ/kg. Then the rates of heat transfer during this condensation process for both cases become

Case 1: T_2 = 30°C: $h_2 = h_{f@30°C}$ = 91.49 kJ/kg.

$$\dot{Q}_{out} = (0.1 \text{ kg/min})(263.5 - 91.49) \text{ kJ/kg} = \mathbf{17.2 \text{ kW}}$$

Case 2: T_2 = 16°C: $h_2 \cong h_{f@16°C}$ =71.69 kJ/kg.

$$\dot{Q}_{out} = (0.1 \text{ kg/min})(263.5 - 71.69) \text{ kJ/kg} = \mathbf{19.2 \text{ kW}}$$

Discussion Note that the rate of heat removal is greater in the second case since the liquid is subcooled in that case.

5-206E A winterizing project is to reduce the infiltration rate of a house from 2.2 ACH to 1.1 ACH. The resulting cost savings are to be determined.

Assumptions **1** The house is maintained at 72°F at all times. **2** The latent heat load during the heating season is negligible. **3** The infiltrating air is heated to 72°F before it exfiltrates. **4** Air is an ideal gas with constant specific heats at room temperature. **5** The changes in kinetic and potential energies are negligible. **6** Steady flow conditions exist.

Properties The gas constant of air is 0.3704 psia.ft^3/lbm·R (Table A-1E). The specific heat of air at room temperature is 0.24 Btu/lbm·°F (Table A-2E).

Analysis The density of air at the outdoor conditions is

$$\rho_o = \frac{P_o}{RT_o} = \frac{13.5\text{ psia}}{(0.3704\text{ psia.ft}^3/\text{lbm.R})(496.5\text{ R})} = 0.0734\text{ lbm/ft}^3$$

The volume of the house is

$$V_{\text{building}} = (\text{Floor area})(\text{Height}) = (3000\text{ ft}^2)(9\text{ ft}) = 27{,}000\text{ ft}^3$$

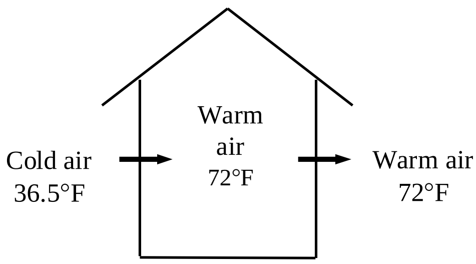
We can view infiltration as a steady stream of air that is heated as it flows in an imaginary duct passing through the house. The energy balance for this imaginary steady-flow system can be expressed in the rate form as

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

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

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

$$\dot{Q}_{in} = \dot{m}C_p(T_2 - T_1) = \rho \dot{V} C_p (T_2 - T_1)$$



The reduction in the infiltration rate is 2.2 – 1.1 = 1.1 ACH.
The reduction in the sensible infiltration heat load corresponding to it is

$$\begin{aligned}\dot{Q}_{\text{infiltration, saved}} &= \rho_o C_p (ACH_{\text{saved}})(V_{\text{building}})(T_i - T_o)\\ &= (0.0734\text{ lbm/ft}^3)(0.24\text{ Btu/lbm.°F})(1.1/\text{h})(27{,}000\text{ ft}^3)(72 - 36.5)\text{°F}\\ &= 18{,}573\text{ Btu/h} = 0.18573\text{ therm/h}\end{aligned}$$

since 1 therm = 100,000 Btu. The number of hours during a six month period is 6×30×24 = 4320 h. Noting that the furnace efficiency is 0.65 and the unit cost of natural gas is \$0.62/therm, the energy and money saved during the 6-month period are

$$\begin{aligned}\text{Energy savings} &= (\dot{Q}_{\text{infiltration, saved}})(\text{No. of hours per year})/\text{Efficiency}\\ &= (0.18573\text{ therm/h})(4320\text{ h/year})/0.65\\ &= 1234\text{ therms/year}\end{aligned}$$

$$\begin{aligned}\text{Cost savings} &= (\text{Energy savings})(\text{Unit cost of energy})\\ &= (1234\text{ therms/year})(\$0.62/\text{therm})\\ &= \mathbf{\$765/year}\end{aligned}$$

Therefore, reducing the infiltration rate by one-half will reduce the heating costs of this homeowner by \$765 per year.

5-207 Outdoors air at -10°C and 90 kPa enters the building at a rate of 35 L/s while the indoors is maintained at 20°C. The rate of sensible heat loss from the building due to infiltration is to be determined.

Assumptions **1** The house is maintained at 20°C at all times. **2** The latent heat load is negligible. **3** The infiltrating air is heated to 20°C before it exfiltrates. **4** Air is an ideal gas with constant specific heats at room temperature. **5** The changes in kinetic and potential energies are negligible. **6** Steady flow conditions exist.

Properties The gas constant of air is R = 0.287 kPa.m^3/kg·K (Table A-1). The specific heat of air at room temperature is C_p = 1.0 kJ/kg·°C (Table A-2).

Analysis The density of air at the outdoor conditions is

$$\rho_o = \frac{P_o}{RT_o} = \frac{90\ \text{kPa}}{(0.287\ \text{kPa.m}^3/\text{kg.K})(-10+273\ \text{K})} = 1.19\ \text{kg/m}^3$$

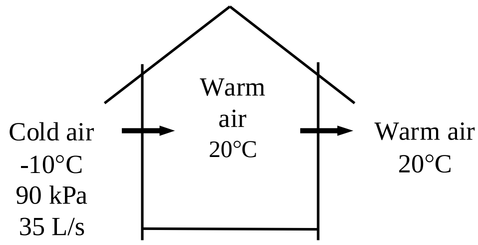
We can view infiltration as a steady stream of air that is heated as it flows in an imaginary duct passing through the building. The energy balance for this imaginary steady-flow system can be expressed in the rate form as

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

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

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

$$\dot{Q}_{in} = \dot{m}C_p(T_2 - T_1)$$



Then the sensible infiltration heat load corresponding to an infiltration rate of 35 L/s becomes

$$\begin{aligned}\dot{Q}_{\text{infiltration}} &= \rho_o \dot{V}_{air} C_p (T_i - T_o) \\ &= (1.19\ \text{kg/m}^3)(0.035\ \text{m}^3/\text{s})(1.005\ \text{kJ/kg.°C})[20 - (-10)]°\text{C} \\ &= \mathbf{1.256\ kW}\end{aligned}$$

Therefore, sensible heat will be lost at a rate of 1.335 kJ/s due to infiltration.

5-208 The maximum flow rate of a standard shower head can be reduced from 13.3 to 10.5 L/min by switching to low-flow shower heads. The ratio of the hot-to-cold water flow rates and the amount of electricity saved by a family of four per year by replacing the standard shower heads by the low-flow ones are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time at any point and thus $\Delta m_{\rm CV} = 0$ and $\Delta E_{\rm CV} = 0$. **2** The kinetic and potential energies are negligible, $ke \cong pe \cong 0$. **3** Heat losses from the system are negligible and thus $\dot{Q} \cong 0$. **4** There are no work interactions involved. **5** .Showers operate at maximum flow conditions during the entire shower. **6** Each member of the household takes a 5-min shower every day. **7** Water is an incompressible substance with constant properties. **8** The efficiency of the electric water heater is 100%.

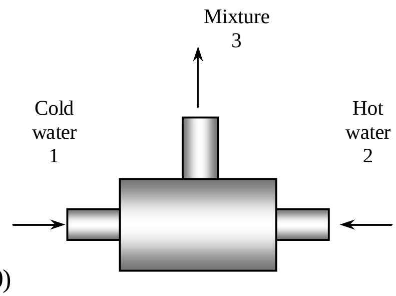
Properties The density and specific heat of water at room temperature are ρ = 1 kg/L and C = 4.18 kJ/kg.°C (Table A-3).

Analysis (*a*) We take the *mixing chamber* as the system. This is a *control volume* since mass crosses the system boundary during the process. We note that there are two inlets and one exit. The mass and energy balances for this steady-flow system can be expressed in the rate form as follows:

Mass balance:

$$\dot{m}_{in} - \dot{m}_{out} = \Delta \dot{m}_{\rm system}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{m}_{in} = \dot{m}_{out} \;\rightarrow\; \dot{m}_1 + \dot{m}_2 = \dot{m}_3$$



Energy balance:

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

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

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

Combining the mass and energy balances and rearranging,

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = (\dot{m}_1 + \dot{m}_2) h_3$$

$$\dot{m}_2 (h_2 - h_3) = \dot{m}_1 (h_3 - h_1)$$

Then the ratio of the mass flow rates of the hot water to cold water becomes

$$\frac{\dot{m}_2}{\dot{m}_1} = \frac{h_3 - h_1}{h_2 - h_3} = \frac{C(T_3 - T_1)}{C(T_2 - T_3)} = \frac{T_3 - T_1}{T_2 - T_3} = \frac{(42 - 15)^\circ\text{C}}{(55 - 42)^\circ\text{C}} = \mathbf{2.08}$$

(*b*) The low-flow heads will save water at a rate of

$$\dot{V}_{\rm saved} = [(13.3\text{-}10.5)\ \text{L/min}](5\ \text{min/person.day})(4\ \text{persons})(365\ \text{days/yr}) = 20{,}440\ \text{L/year}$$

$$\dot{m}_{\rm saved} = \rho \dot{V}_{\rm saved} = (1\ \text{kg/L})(20{,}440\ \text{L/year}) = 20{,}440\ \text{kg/year}$$

Then the energy saved per year becomes

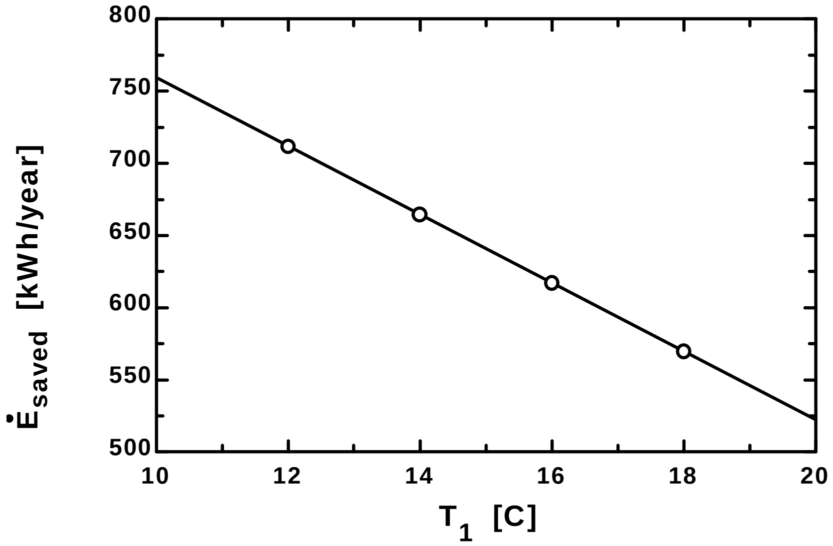
$$\begin{aligned}\text{Energy saved} &= \dot{m}_{\rm saved} C \Delta T = (20{,}440\ \text{kg/year})(4.18\ \text{kJ/kg.}^\circ\text{C})(42\text{-}15)^\circ\text{C} \\ &= 2{,}307{,}000\ \text{kJ/year} \\ &= \mathbf{641\ kWh} \quad (\text{since } 1\ \text{kWh} = 3600\ \text{kJ})\end{aligned}$$

Therefore, switching to low-flow shower heads will save about 641 kWh of electricity per year.

5-209 Problem 5-208 is reconsidered. The effect of the inlet temperature of cold water on the energy saved by using the low-flow showerhead as the inlet temperature varies from 10°C to 20°C is to be investigated. The electric energy savings is to be plotted against the water inlet temperature.

```
"Knowns:"
C_P = 4.18"[kJ/kg-K ]"
density=1"[kg/L]"
{T_1 = 15"[C]"}
T_2 =  55"[C]"
T_3 = 42"[C]"
V_dot_old = 13.3"[L/min]"
V_dot_new = 10.5"[L/min]"
m_dot_1=1"[kg/s]"  "We can set m_dot_1 = 1 without loss of generality."

"Analysis:"
 "(a) We take the mixing chamber as the system. This is a control volume since mass
crosses the system boundary during the process. We note that there are two inlets
and one exit.  The mass and energy balances for this steady-flow system can be
expressed in the rate form as follows:"

"Mass balance:"
m_dot_in - m_dot_out = DELTAm_dot_sys
DELTAm_dot_sys=0"[kg/s]"
m_dot_in =m_dot_1 + m_dot_2"[kg/s]"
m_dot_out = m_dot_3"[kg/s]"
"The ratio of the mass flow rates of the hot water to cold water is
obtained by setting m_dot_1=1[kg/s].  Then m_dot_2 represents
the ratio of m_dot_2/m_dot_1"

"Energy balance:"
E_dot_in - E_dot_out = DELTAE_dot_sys
DELTAE_dot_sys=0"[kW]"
E_dot_in = m_dot_1*h_1 + m_dot_2*h_2"[kW]"
E_dot_out = m_dot_3*h_3"[kW]"
h_1 = C_P*T_1"[kJ/kg]"
h_2 = C_P*T_2"[kJ/kg]"
h_3 = C_P*T_3"[kJ/kg]"

"(b) The low-flow heads will save water at a rate of "
V_dot_saved = (V_dot_old  -  V_dot_new)"L/min"*(5"min/person-
day")*(4"persons")*(365"days/year") "[L/year]"
m_dot_saved=density*V_dot_saved"[kg/year]"
"Then the energy saved per year becomes"
E_dot_saved=m_dot_saved*C_P*(T_3 - T_1)"kJ/year"*convert(kJ,kWh) "[kWh/year]"
"Therefore, switching to low-flow shower heads will save about 641 kWh of electricity per
year. "
"Ratio of hot-to-cold water flow rates:"
m_ratio = m_dot_2/m_dot_1
```

E_{saved} [kWh/year]	**T_1 [C]**
759.5	10
712	12
664.5	14
617.1	16
569.6	18
522.1	20


800
750
700
650
600
550
500
10
12
14
16
18
20
T1 [C]
Ėsaved [kWh/year]

5-210 A fan is powered by a 0.5 hp motor, and delivers air at a rate of 85 m^3/min. The highest possible air velocity at the fan exit is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time at any point and thus $\Delta m_{CV} = 0$ and $\Delta E_{CV} = 0$. **2** The inlet velocity and the change in potential energy are negligible, $\mathbf{V}_1 \cong 0$ and $\Delta pe \cong 0$. **3** There are no heat and work interactions other than the electrical power consumed by the fan motor. **4** The efficiencies of the motor and the fan are 100% since best possible operation is assumed. **5** Air is an ideal gas with constant specific heats at room temperature.

Properties The density of air is given to be ρ = 1.18 kg/m^3. The constant pressure specific heat of air at room temperature is C_p = 1.005 kJ/kg.°C (Table A-2).

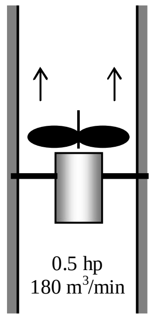
Analysis We take the *fan-motor assembly* as the system. This is a *control volume* since mass crosses the system boundary during the process. We note that there is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$.

The velocity of air leaving the fan will be highest when all of the entire electrical energy drawn by the motor is converted to kinetic energy, and the friction between the air layers is zero. In this best possible case, no energy will be converted to thermal energy, and thus the temperature change of air will be zero, $T_2 = T_1$. Then the energy balance for this steady-flow system can be expressed in the rate form as

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

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

$$\dot{W}_{e,in} + \dot{m}h_1 = \dot{m}(h_2 + \mathbf{V}_2^2/2) \quad \text{(since } \mathbf{V}_1 \cong 0 \text{ and } \Delta\text{pe} \cong 0)$$



Noting that the temperature and thus enthalpy remains constant, the relation above simplifies further to

$$\dot{W}_{e,in} = \dot{m}\mathbf{V}_2^2/2$$

where

$$\dot{m} = \rho\dot{V} = (1.18\,\text{kg/m}^3)(85\,\text{m}^3/\text{min}) = 100.3\,\text{kg/min} = 1.67\,\text{kg/s}$$

Solving for $\mathbf{V}_2$ and substituting gives

$$\mathbf{V}_2 = \sqrt{\frac{2\dot{W}_{e,in}}{\dot{m}}} = \sqrt{\frac{2(0.5\,\text{hp})}{1.67\,\text{kg/s}}\left(\frac{745.7\,\text{W}}{1\,\text{hp}}\right)\left(\frac{1\,\text{m}^2/\text{s}^2}{1\,\text{W}}\right)} = \mathbf{21.1\ m/s}$$

Discussion In reality, the velocity will be less because of the inefficiencies of the motor and the fan.

5-211 The average air velocity in the circular duct of an air-conditioning system is not to exceed 10 m/s. If the fan converts 70 percent of the electrical energy into kinetic energy, the size of the fan motor needed and the diameter of the main duct are to be determined.

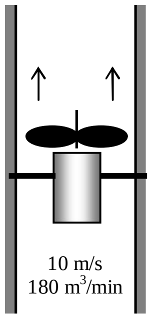
Assumptions **1** This is a steady-flow process since there is no change with time at any point and thus $\Delta m_{\rm CV} = 0$ and $\Delta E_{\rm CV} = 0$. **2** The inlet velocity is negligible, $\mathbf{V}_1 \cong 0$. **3** There are no heat and work interactions other than the electrical power consumed by the fan motor. **4** Air is an ideal gas with constant specific heats at room temperature.

Properties The density of air is given to be ρ = 1.20 kg/m^3. The constant pressure specific heat of air at room temperature is C_p = 1.005 kJ/kg.°C (Table A-2).

Analysis We take the *fan-motor assembly* as the system. This is a *control volume* since mass crosses the system boundary during the process. We note that there is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The change in the kinetic energy of air as it is accelerated from zero to 10 m/s at a rate of 180 m^3/s is

$$\dot{m} = \rho \dot{V} = (1.20\ \text{kg/m}^3)(180\ \text{m}^3/\text{min}) = 216\ \text{kg/min} = 3.6\ \text{kg/s}$$

$$\Delta \dot{\text{KE}} = \dot{m}\frac{V_2^2 - V_1^2}{2} = (3.6\ \text{kg/s})\frac{(10\ \text{m/s})^2 - 0}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = 0.18\ \text{kW}$$



It is stated that this represents 70% of the electrical energy consumed by the motor. Then the total electrical power consumed by the motor is determined to be

$$0.7\dot{W}_{\text{motor}} = \Delta \dot{KE} \quad \rightarrow \quad \dot{W}_{\text{motor}} = \frac{\Delta \dot{KE}}{0.7} = \frac{0.18\ \text{kW}}{0.7} = \mathbf{0.257\ kW}$$

The diameter of the main duct is

$$\dot{V} = \mathbf{V}A = \mathbf{V}(\pi D^2/4) \ \rightarrow \ D = \sqrt{\frac{4\dot{V}}{\pi \mathbf{V}}} = \sqrt{\frac{4(180\ \text{m}^3/\text{min})}{\pi(10\ \text{m/s})}\left(\frac{1\ \text{min}}{60\ \text{s}}\right)} = \mathbf{0.618\,m}$$

Therefore, the motor should have a rated power of at least 0.257 kW, and the diameter of the duct should be at least 61.8 cm

5-212 An evacuated bottle is surrounded by atmospheric air. A valve is opened, and air is allowed to fill the bottle. The amount of heat transfer through the wall of the bottle when thermal and mechanical equilibrium is established is to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Air is an ideal gas. **3** Kinetic and potential energies are negligible. **4** There are no work interactions involved. **5** The direction of heat transfer is to the air in the bottle (will be verified).

Analysis We take the bottle as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{in} - m_{out} = \Delta m_{\text{system}} \quad \rightarrow \quad m_i = m_2 \quad (\text{since } m_{out} = m_{initial} = 0)$

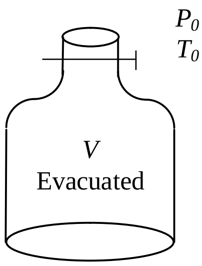
Energy balance:

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

$$Q_{in} + m_i h_i = m_2 u_2 \quad (\text{since } W \cong E_{out} = E_{initial} = ke \cong pe \cong 0)$$

Combining the two balances:

$$Q_{in} = m_2(u_2 - h_i) = m_2(C_v T_2 - C_p T_i)$$



But $T_i = T_2 = T_0$ and $C_p - C_v = R$. Substituting,

$$Q_{in} = m_2(C_v - C_p)T_0 = -m_2 R T_0 = -\frac{P_0 V}{R T_0} R T_0 = -P_0 V$$

Therefore,

$$Q_{\text{out}} = \boldsymbol{P_0 V} \quad \text{(Heat is lost from the tank)}$$

5-213 An adiabatic air compressor is powered by a direct-coupled steam turbine, which is also driving a generator. The net power delivered to the generator 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 devices are adiabatic and thus heat transfer is negligible. **4** Air is an ideal gas with variable specific heats.

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

$$\left.\begin{array}{l} P_3 = 12.5\,\text{MPa} \\ T_3 = 500^\circ\text{C} \end{array}\right\} h_3 = 3341.8\,\text{kJ/kg}$$

and

$$\left.\begin{array}{l} P_4 = 10\,\text{kPa} \\ x_4 = 0.92 \end{array}\right\} h_4 = h_f + x_4 h_{fg} = 191.83 + (0.92)(2392.8) = 2393.2\,\text{kJ/kg}$$

From the air table (Table A-17),

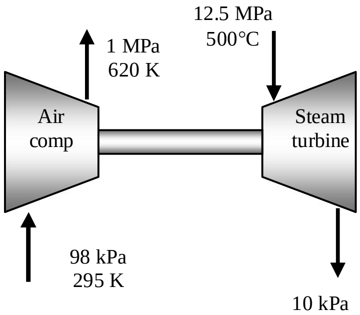
$$T_1 = 295\text{ K} \longrightarrow h_1 = 295.17\text{ kJ/kg}$$

$$T_2 = 620\text{ K} \longrightarrow h_2 = 628.07\text{ kJ/kg}$$

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

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

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



For the turbine and the compressor it becomes

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

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

Substituting,

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

$$\dot{W}_{\text{turb,out}} = (25\,\text{kg/s})(3341.8 - 2393.2)\,\text{kJ/kg} = 23{,}715\text{ kW}$$

Therefore,

$$\dot{W}_{\text{net,out}} = \dot{W}_{\text{turb,out}} - \dot{W}_{\text{comp,in}} = 23{,}715 - 3329 = \mathbf{20{,}386\ kW}$$

5-214 Water is heated from 16°C to 43°C by an electric resistance heater placed in the water pipe as it flows through a showerhead steadily at a rate of 10 L/min. The electric power input to the heater, and the money that will be saved during a 10-min shower by installing a heat exchanger with an effectiveness of 0.50 are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time at any point within the system and thus $\Delta m_{CV} = 0$ and $\Delta E_{CV} = 0$,. **2** Water is an incompressible substance with constant specific heats. **3** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **4** Heat losses from the pipe are negligible.

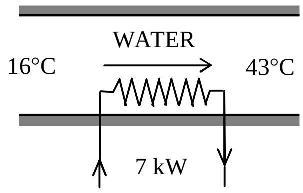
Properties The density and specific heat of water at room temperature are ρ = 1 kg/L and C = 4.18 kJ/kg·°C (Table A-3).

Analysis We take the pipe as the system. This is a *control volume* since mass crosses the system boundary during the process. We observe that there is only one inlet and one exit and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Then the energy balance for this steady-flow system can be expressed in the rate form as

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

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

$$\dot{W}_{\text{e,in}} = \dot{m}(h_2 - h_1) = \dot{m}[C(T_2 - T_1) + v(P_2 - P_1)^{\nearrow 0}] = \dot{m}C(T_2 - T_1)$$



where

$$\dot{m} = \rho\dot{V} = (1\ \text{kg/L})(10\ \text{L/min}) = 10\ \text{kg/min}$$

Substituting,

$$\dot{W}_{\text{e,in}} = (10/60\text{kg/s})(4.18\ \text{kJ/kg}\cdot^{\circ}\text{C})(43-16)^{\circ}\text{C} = \mathbf{18.8\ kW}$$

The energy recovered by the heat exchanger is

$$\begin{aligned}\dot{Q}_{\text{saved}} &= \varepsilon\dot{Q}_{\max} = \varepsilon\dot{m}C(T_{\max} - T_{\min}) \\ &= 0.5(10/60\text{kg/s})(4.18\ \text{kJ/kg.}^{\circ}\text{C})(39-16)^{\circ}\text{C} \\ &= 8.0\text{kJ/s} = 8.0\text{kW}\end{aligned}$$

Therefore, 8.0 kW less energy is needed in this case, and the required electric power in this case reduces to

$$\dot{W}_{\text{in,new}} = \dot{W}_{\text{in,old}} - \dot{Q}_{\text{saved}} = 18.8 - 8.0 = \mathbf{10.8\ kW}$$

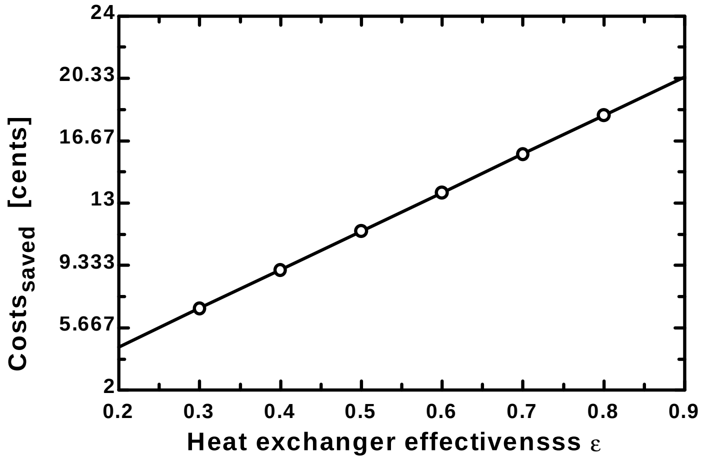
The money saved during a 10-min shower as a result of installing this heat exchanger is

$$(8.0\ \text{kW})(10/60\ \text{h})(8.5\ \text{cents/kWh}) = \mathbf{11.3\ cents}$$

5-215 Problem 5-214 is reconsidered. The effect of the heat exchanger effectiveness on the money saved as the effectiveness ranges from 20 percent to 90 percent is to be investigated, and the money saved is to be plotted against the effectiveness,

```
"Knowns:"
density = 1 "[kg/L]"
V_dot = 10 "[L/min]"
C = 4.18 "[kJ/kg-C]"
T_1 = 16"[C]"
T_2 = 43"[C]"
T_max = 39"[C]"
T_min = T_1"[C]"
epsilon = 0.5  "heat exchanger effectiveness "
EleRate = 8.5 "[cents/kWh]"

"For entrance, one exit, steady flow m_dot_in = m_dot_out = m_dot_wat er:"
m_dot_water= density*V_dot /convert(min,s)"[kg/s]"

"Energy balance for the pipe:"
W_dot_ele_in+ m_dot_water*h_1=m_dot_water*h_2 "Neglect ke and pe"

"For incompressible fluid in a constant pressure process, the enthalpy is:"
h_1 = C*T_1"[kJ/kg]"
h_2 = C*T_2"[kJ/kg]"

"The energy recovered by the heat exchanger is"
Q_dot_saved=epsilon*Q_dot_max "[kW]"
Q_dot_max = m_dot_water*C*(T_max - T_min)"[kW]"

"Therefore, 8.0 kW less energy is needed in this case, and the required
electric power in this case reduces to"
W_dot_ele_new = W_dot_ele_in - Q_dot_saved "[kW]"

"The money saved during a 10-min shower as a result of installing this heat exchanger is"
Costs_saved = Q_dot_saved*10"min"*convert(min,h)*EleRate"[cents]"
```

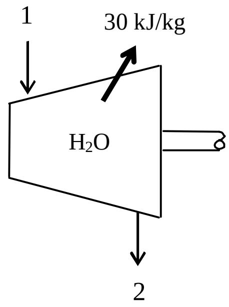
$Costs_{saved}$ [cents]	ε
4.54	0.2
6.81	0.3
9.08	0.4
11.35	0.5
13.62	0.6
15.89	0.7
18.16	0.8
20.43	0.9



5-216 [*Also solved by EES on enclosed CD***]** Steam expands in a turbine steadily. The mass flow rate of the steam, the exit velocity, and the power output 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.

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



$$\left.\begin{array}{l} P_1 = 10 \text{ MPa} \\ T_1 = 550^\circ\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.03564 \text{ m}^3/\text{kg} \\ h_1 = 3500.9 \text{ kJ/kg} \end{array}$$

and

$$\left.\begin{array}{l} P_2 = 25 \text{ kPa} \\ x_2 = 0.95 \end{array}\right\} \begin{array}{l} v_2 = v_f + x_2 v_{fg} = 0.00102 + (0.95)(6.203) = 5.894 \text{ m}^3/\text{kg} \\ h_2 = h_f + x_2 h_{fg} = 271.93 + (0.95)(2346.3) = 2500.9 \text{ kJ/kg} \end{array}$$

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

$$\dot{m} = \frac{1}{v_1}\mathbf{V}_1 A_1 = \frac{1}{0.03564 \text{m}^3/\text{kg}}(60 \text{m/s})(0.015 \text{m}^2) = \mathbf{25.3 kg/s}$$

(*b*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Then the exit velocity is determined from

$$\dot{m} = \frac{1}{v_2}\mathbf{V}_2 A_2 \longrightarrow \mathbf{V}_2 = \frac{\dot{m} v_2}{A_2} = \frac{(25.3 \text{ kg/s})(5.894 \text{ m}^3/\text{kg})}{0.14 \text{ m}^2} = \mathbf{1065 \text{ m/s}}$$

(*c*) 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

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

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

$$\dot{m}(h_1 + \mathbf{V}_1^2/2) = \dot{W}_{\text{out}} + \dot{Q}_{\text{out}} + \dot{m}(h_2 + \mathbf{V}_2^2/2) \quad (\text{since } \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{out}} = -\dot{Q}_{\text{out}} - \dot{m}\left(h_2 - h_1 + \frac{\mathbf{V}_2^2 - \mathbf{V}_1^2}{2}\right)$$

Then the power output of the turbine is determined by substituting to be

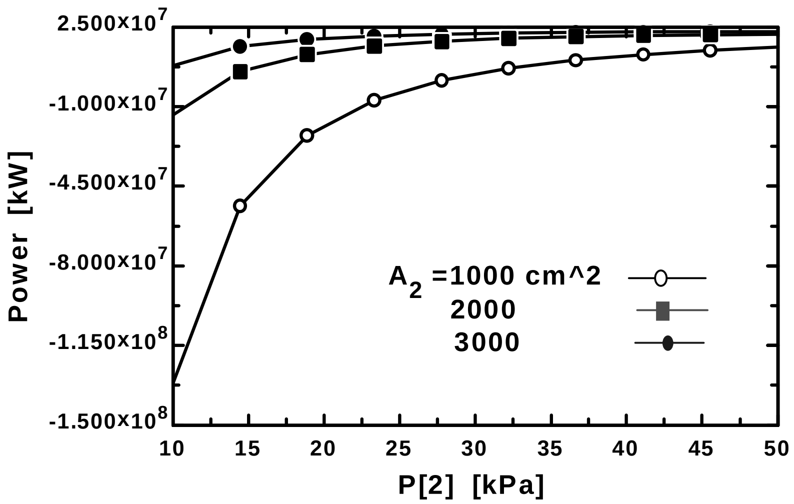
$$\dot{W}_{\text{out}} = -(25.3 \times 30) \text{kJ/s} - (25.3 \text{kg/s})\left(2500.9 - 3500.9 + \frac{(1065 \text{m/s})^2 - (60 \text{m/s})^2}{2}\left(\frac{1 \text{kJ/kg}}{1000 \text{m}^2/\text{s}^2}\right)\right)$$

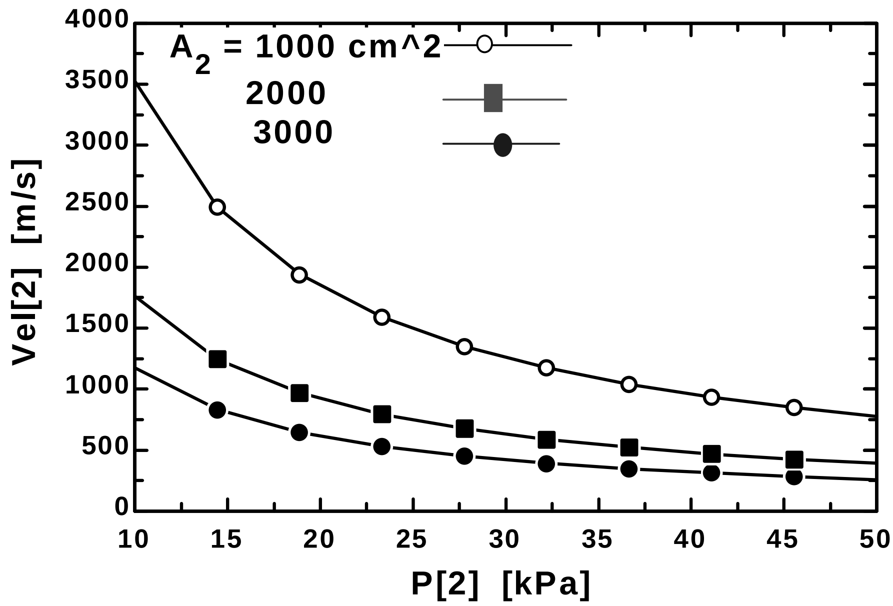
$$= \mathbf{10{,}240 kW}$$

5-217 Problem 5-216 is reconsidered. The effects of turbine exit area and turbine exit pressure on the exit velocity and power output of the turbine as the exit pressure varies from 10 kPa to 50 kPa (with the same quality), and the exit area to varies from 1000 cm2 to 3000 cm2 is to be investigated. The exit velocity and the power output are to be plotted against the exit pressure for the exit areas of 1000, 2000, and 3000 cm2.

```
A[1]=150 "[cm^2]"
T[1]=550"[c]"
P[1]=10000"[kPa]"
Vel[1]= 60 "[m/s]"
A[2]=1400"[cm^2]"
P[2]=25"[kPa]"
q_out = 30"[kJ/kg"
m_dot = A[1]*Vel[1]/v[1]*convert(cm^2,m^2) "[kg/s]"
v[1]=volume(steam, T=T[1], P=P[1]) "[m^3/kg]"  "specific volume of steam at state 1"
Vel[2]=m_dot*v[2]/(A[2]*convert(cm^2,m^2)) "[m/s]"
v[2]=volume(steam, x=0.95, P=P[2]) "[m^3/kg]"  "specific volume of steam at state 2"
T[2]=temperature(steam, P=P[2], v=v[2]) "[C]" "not required, but good to know"

"[conservation of Energy for steady-flow:"
"Ein_dot - Eout_dot = DeltaE_dot"  "For steady-flow, DeltaE_dot = 0"
DELTAE_dot=0 "[kW]"
"For the turbine as the control volume, neglecting the PE of each flow steam:"
E_dot_in=E_dot_out     "[kW]"
h[1]=enthalpy(steam,T=T[1], P=P[1])  "[kJ/kg]"
E_dot_in=m_dot*(h[1]+ Vel[1]^2/(2*1000) )"[kJ/kg]"
h[2]=enthalpy(steam,x=0.95, P=P[2])   "[kJ/kg]"
E_dot_out=m_dot*(h[2]+ Vel[2]^2/(2*1000) )+ m_dot *q_out+ W_dot_out"[kW]"
Power=W_dot_out*convert(MW,kW) "[kW]"
Q_dot_out=m_dot*q_out"[kW]"
```

Power [kW]	P_2 [kPa]	Vel_2 [m/s]
-1.377E+07	10	1763
5.469E+06	14.44	1247
1.298E+07	18.89	969.6
1.665E+07	23.33	795.1
1.870E+07	27.78	675.2
1.994E+07	32.22	587.5
2.073E+07	36.67	520.5
2.127E+07	41.11	467.6
2.164E+07	45.56	424.7
2.190E+07	50	389.2

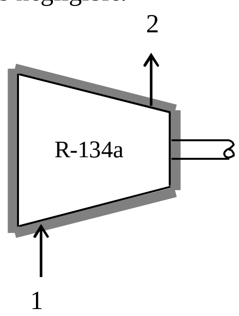

2.500x10^7
-1.000x10^7
-4.500x10^7
-8.000x10^7
-1.150x10^8
-1.500x10^8
Power [kW]
A2 =1000 cm^2
2000
3000
10 15 20 25 30 35 40 45 50
P[2] [kPa]


4000 3500 3000 2500 2000 1500 1000 500 0
Vel[2] [m/s]
A2 = 1000 cm^2
2000
3000
10 15 20 25 30 35 40 45 50
P[2] [kPa]

5-218E Refrigerant-134a is compressed steadily by a compressor. The mass flow rate of the refrigerant and the exit temperature 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)



$$\left.\begin{array}{l} P_1 = 15 \text{ psia} \\ T_1 = 20^\circ\text{F} \end{array}\right\} \begin{array}{l} v_1 = 3.2468 \text{ ft}^3/\text{lbm} \\ h_1 = 106.34 \text{ Btu/lbm} \end{array}$$

Analysis (*a*) The mass flow rate of refrigerant is

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{10 \text{ ft}^3/\text{s}}{3.2468 \text{ ft}^3/\text{lbm}} = \mathbf{3.08\ lbm/s}$$

(*b*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the 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 in the rate form as

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

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

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

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

Substituting,

$$(60 \text{ hp})\left(\frac{0.7068 \text{ Btu/s}}{1 \text{ hp}}\right) = (3.08 \text{ lbm/s})(h_2 - 106.34) \text{Btu/lbm}$$

$$h_2 = 120.11 \text{ Btu/lbm}$$

Then the exit temperature becomes

$$\left.\begin{array}{l} P_2 = 120 \text{ psia} \\ h_2 = 120.11 \text{ Btu/lbm} \end{array}\right\} T_2 = \mathbf{114.6^\circ F}$$

5-219 Air is preheated by the exhaust gases of a gas turbine in a regenerator. For a specified heat transfer rate, the exit temperature of air and the mass flow rate of exhaust gases are to be determined.

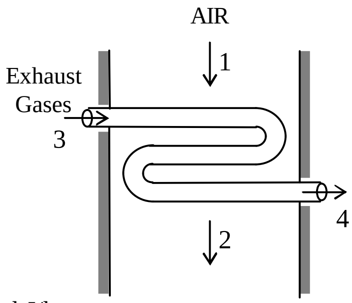
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** Heat loss from the regenerator to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **5** Exhaust gases can be treated as air. **6** Air is an ideal gas with variable specific heats.

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

$$T_1 = 550\text{ K} \rightarrow h_1 = 554.71\text{ kJ/kg}$$
$$T_3 = 800\text{ K} \rightarrow h_3 = 821.95\text{ kJ/kg}$$
$$T_4 = 600\text{ K} \rightarrow h_4 = 607.02\text{ kJ/kg}$$

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

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



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

Substituting,

$$3200\text{kJ/s} = (800/60\text{kg/s})(h_2 - 554.71\text{kJ/kg}) \rightarrow h_2 = 794.71 \text{ kJ/kg}$$

Then from Table A-17 we read $\quad T_2 = \mathbf{775.1\ K}$

(*b*) Treating the exhaust gases as an ideal gas, the mass flow rate of the exhaust gases is determined from the steady-flow energy relation applied only to the exhaust gases,

$$\dot{E}_{in} = \dot{E}_{out}$$
$$\dot{m}_{exhaust} h_3 = \dot{Q}_{out} + \dot{m}_{exhaust} h_4 \quad (\text{since } \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$
$$\dot{Q}_{out} = \dot{m}_{exhaust}(h_3 - h_4)$$

$$3200\text{kJ/s} = \dot{m}_{exhaust}(821.95 - 607.02)\text{kJ/kg}$$

It yields

$$\dot{m}_{exhaust} = \mathbf{14.9\ kg/s}$$

5-220 Water is to be heated steadily from 20°C to 55°C by an electrical resistor inside an insulated pipe. The power rating of the resistance heater and the average velocity of the water are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time at any point within the system and thus $\Delta m_{CV} = 0$ and $\Delta E_{CV} = 0$. **2** Water is an incompressible substance with constant specific heats. **3** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **4** The pipe is insulated and thus the heat losses are negligible.

Properties The density and specific heat of water at room temperature are ρ = 1000 kg/m^3 and C = 4.18 kJ/kg·°C (Table A-3).

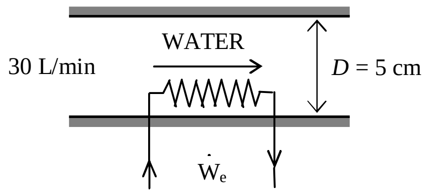
Analysis (*a*) We take the pipe as the system. This is a *control volume* since mass crosses the system boundary during the process. Also, there is only one inlet and one exit and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The energy balance for this steady-flow system can be expressed in the rate form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

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

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

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

$$\dot{W}_{e,in} = \dot{m}(h_2 - h_1) = \dot{m}[C(T_2 - T_1) + v\Delta P^{\nearrow 0}] = \dot{m}C(T_2 - T_1)$$



The mass flow rate of water through the pipe is

$$\dot{m} = \rho \dot{V}_1 = (1000 \text{ kg/m}^3)(0.030 \text{ m}^3/\text{min}) = 30 \text{ kg/min}$$

Therefore,

$$\dot{W}_{e,in} = \dot{m}C(T_2 - T_1) = (30/60 \text{ kg/s})(4.18 \text{ kJ/kg}\cdot^\circ\text{C})(55 - 20)^\circ\text{C} = \mathbf{73.2\ kW}$$

(*b*) The average velocity of water through the pipe is determined from

$$\mathbf{V}_1 = \frac{\dot{V}_1}{A_1} = \frac{\dot{V}}{\pi r^2} = \frac{0.030 \text{ m}^3/\text{min}}{\pi(0.025 \text{ m})^2} = \mathbf{15.3\ m/min}$$

5-221 Two rigid tanks that contain water at different states are connected by a valve. The valve is opened and the two tanks come to the same state at the temperature of the surroundings. The final pressure and the amount of heat transfer are to be determined.

Assumptions **1** The tanks are stationary and thus the kinetic and potential energy changes are zero. **2** The tank is insulated and thus heat transfer is negligible. **3** There are no work interactions.

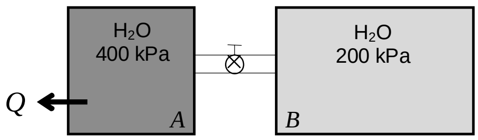
Analysis We take the entire contents of the tank as the system. This is a closed system since no mass enters or leaves. Noting that the volume of the system is constant and thus there is no boundary work, the energy balance for this stationary closed system can be expressed as

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

$$-Q_{out} = \Delta U = (\Delta U)_A + (\Delta U)_B \quad (\text{since } W = \text{KE} = \text{PE} = 0)$$

$$Q_{out} = -[U_{2,A+B} - U_{1,A} - U_{1,B}]$$

$$= -[m_{2,\text{total}} u_2 - (m_1 u_1)_A - (m_1 u_1)_B]$$



The properties of water in each tank are (Tables A-4 through A-6)

Tank A:

$$\left.\begin{array}{l} P_1 = 400 \text{kPa} \\ x_1 = 0.80 \end{array}\right\} \begin{array}{ll} v_f = 0.001084, & v_g = 0.4625 \text{m}^3/\text{kg} \\ u_f = 604.31, & u_{fg} = 1949.3 \text{kJ/kg} \end{array}$$

$$v_{1,A} = v_f + x_1 v_{fg} = 0.001084 + [0.8 \times (0.4625 - 0.001084)] = 0.3702 \text{m}^3/\text{kg}$$

$$u_{1,A} = u_f + x_1 u_{fg} = 604.31 + (0.8 \times 1949.3) = 2163.75 \text{kJ/kg}$$

Tank B:

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ T_1 = 250 \,^\circ\text{C} \end{array}\right\} \begin{array}{l} v_{1,B} = 1.1988 \text{ m}^3/\text{kg} \\ u_{1,B} = 2731.2 \text{ kJ/kg} \end{array}$$

$$m_{1,A} = \frac{V_A}{v_{1,A}} = \frac{0.2 \text{ m}^3}{0.3702 \text{ m}^3/\text{kg}} = 0.540 \text{ kg}$$

$$m_{1,B} = \frac{V_B}{v_{1,B}} = \frac{0.5 \text{ m}^3}{1.1988 \text{ m}^3/\text{kg}} = 0.417 \text{ kg}$$

$$m_t = m_{1,A} + m_{1,B} = 0.540 + 0.417 = 0.957 \text{ kg}$$

$$v_2 = \frac{V_t}{m_t} = \frac{0.7 \text{ m}^3}{0.957 \text{ kg}} = 0.731 \text{ m}^3/\text{kg}$$

$$\left.\begin{array}{l} T_2 = 25 \,^\circ\text{C} \\ v_2 = 0.731 \text{ m}^3/\text{kg} \end{array}\right\} \begin{array}{ll} v_f = 0.001003, & v_g = 43.36 \text{ m}^3/\text{kg} \\ u_f = 104.88, & u_{fg} = 2304.9 \text{ kJ/kg} \end{array}$$

Thus at the final state the system will be a saturated liquid-vapor mixture since $v_f < v_2 < v_g$. Then the final pressure must be

$$P_2 = P_{sat\ @\ 25\ °C} = \mathbf{3.169\ kPa}$$

Also,

$$x_2 = \frac{v_2 - v_f}{v_{fg}} = \frac{0.731 - 0.001}{43.36 - 0.001} = 0.0168$$

$$u_2 = u_f + x_2 u_{fg} = 104.88 + (0.0168 \times 2304.9) = 143.60 \text{kJ/kg}$$

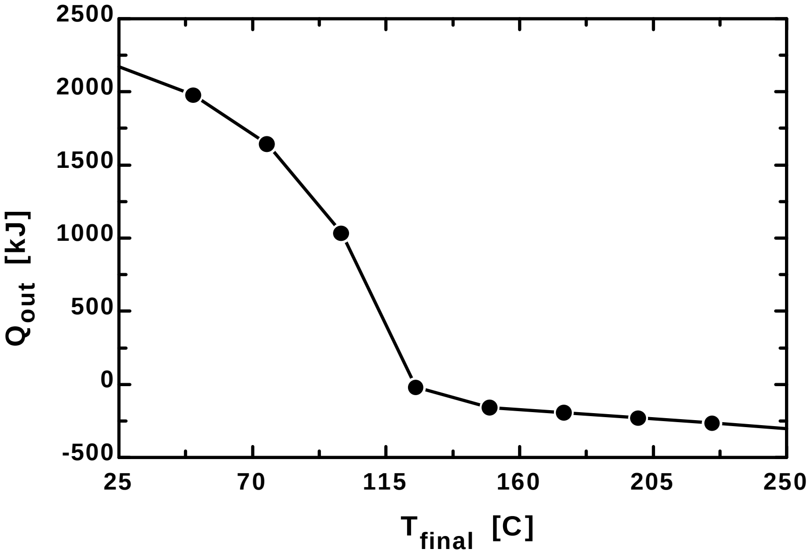
Substituting,

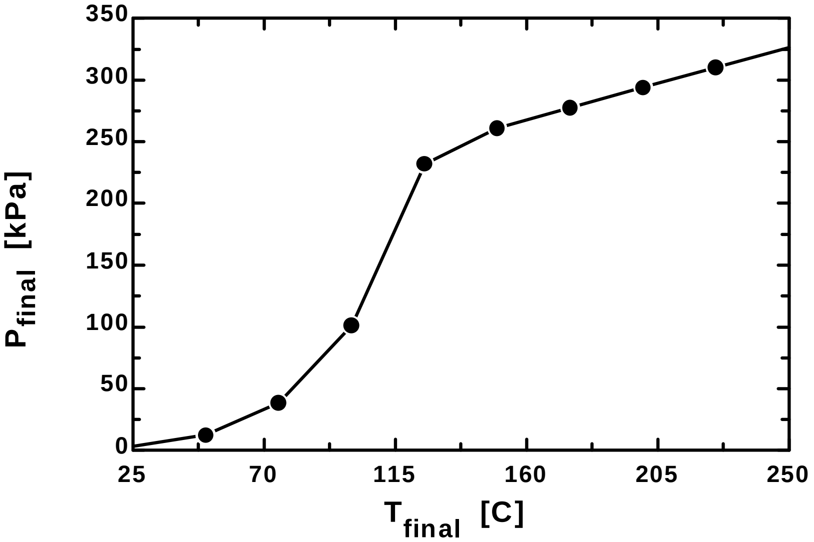
$$Q_{out} = -[(0.957)(143.6) - (0.540)(2163.75) - (0.417)(2731.2)] = \mathbf{2170\ kJ}$$

5-222 Problem 5-221 is reconsidered. The effect of the environment temperature on the final pressure and the heat transfer as the environment temperature varies from 0°C to 50°C is to be investigated. The final results are to be plotted against the environment temperature.

```
"Knowns"
Vol_A=0.2"[m^3]"
P_A[1]=400"[kPa]"
x_A[1]=0.8
T_B[1]=250"[C]"
P_B[1]=200"[kPa]"
Vol_B=0.5"[m^3]"
T_final=25"[C]"  "T_final = T_surroundings.  To do the parametric study
or to solve the problem when Q_out = 0, place this statement in {}."
{Q_out=0"[kJ]"}  "To determine the surroundings temperature that
makes Q_out = 0, remove the {} and resolve the problem."

"Solution"
"Conservation of Energy for the combined tanks:"
E_in-E_out=DELTAE
E_in=0"[kJ]"
E_out=Q_out"[kJ]"
DELTAE=m_A*(u_A[2]-u_A[1])+m_B*(u_B[2]-u_B[1])"[kJ]"
m_A=Vol_A/v_A[1]"[kg]"
m_B=Vol_B/v_B[1]"[kg]"
u_A[1]=INTENERGY(Steam,P=P_A[1], x=x_A[1])"[kJ/kg]"
v_A[1]=volume(Steam,P=P_A[1], x=x_A[1])"[m^3/kg]"
T_A[1]=temperature(Steam,P=P_A[1], x=x_A[1])"[C]"
u_B[1]=INTENERGY(Steam,P=P_B[1],T=T_B[1])"[kJ/kg]"
v_B[1]=volume(Steam,P=P_B[1],T=T_B[1])"[m^3/kg]"

"At the final state the steam has uniform properties through out the entire system."
u_B[2]=u_final"[kJ/kg]"
u_A[2]=u_final"[kJ/kg]"
m_final=m_A+m_B"[kg]"
Vol_final=Vol_A+Vol_B"[m^3]"
v_final=Vol_final/m_final"[m^3/kg]"
u_final=INTENERGY(Steam,T=T_final, v=v_final)"[kJ/kg]"
P_final=pressure(Steam,T=T_final, v=v_final)"[kPa]"
```

P_{final} [kPa]	Q_{out} [kJ]	T_{final} [C]
3.169	2170	25
12.34	1977	50
38.56	1641	75
101.3	1034	100
232	-19.89	125
261	-155.3	150
277.5	-192.5	175
293.9	-229.4	200
310.1	-266.2	225
326.3	-303.1	250


2500
2000
1500
1000
500
0
-500
Q out [kJ]
25
70
115
160
205
250
T final [C]


350
300
250
200
150
100
50
0
P final [kPa]
25
70
115
160
205
250
T final [C]

5-223 A rigid tank filled with air is connected to a cylinder with zero clearance. The valve is opened, and air is allowed to flow into the cylinder. The temperature is maintained at 30°C at all times. The amount of heat transfer with the surroundings is to be determined.

Assumptions **1** Air is an ideal gas. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** There are no work interactions involved other than the boundary work.

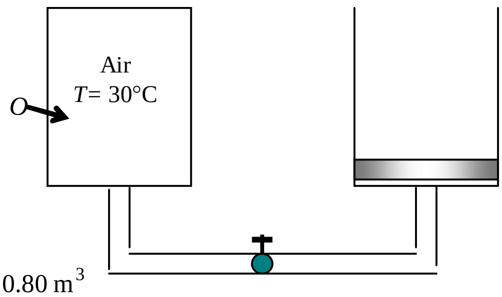
Properties The gas constant of air is R = 0.287 kPa.m^3/kg.K (Table A-1).

Analysis We take the entire air in the tank and the cylinder to be the system. This is a closed system since no mass crosses the boundary of the system. The energy balance for this closed system can be expressed as

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

$$Q_{in} - W_{b,out} = \Delta U = m(u_2 - u_1) = 0$$

$$Q_{in} = W_{b,out}$$



since $u = u(T)$ for ideal gases, and thus $u_2 = u_1$ when $T_1 = T_2$.
The initial volume of air is

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \longrightarrow V_2 = \frac{P_2}{P_1}\frac{T_2}{T_1} V_1 = \frac{400 \text{ kPa}}{200 \text{ kPa}} \times 1 \times (0.4 \text{ m}^3) = 0.80 \text{ m}^3$$

The pressure at the piston face always remains constant at 200 kPa. Thus the boundary work done during this process is

$$W_{b,out} = \int_1^2 \text{PdV} = P_2(V_2 - V_1) = (200\text{kPa})(0.8 - 0.4)\text{m}^3 \left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right) = 80 \text{ kJ}$$

Therefore, the heat transfer is determined from the energy balance to be

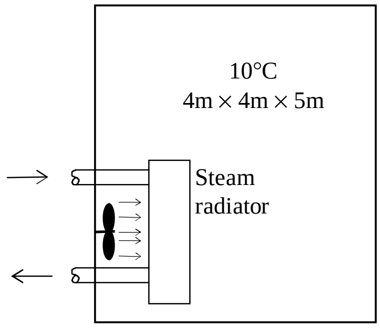
$$W_{b,out} = Q_{in} = \mathbf{80 \ kJ}$$

5-224 A well-insulated room is heated by a steam radiator, and the warm air is distributed by a fan. The average temperature in the room after 30 min is to be determined.

Assumptions **1** Air is an ideal gas with constant specific heats at room temperature. **2** The kinetic and potential energy changes are negligible. **3** The air pressure in the room remains constant and thus the air expands as it is heated, and some warm air escapes.

Properties The gas constant of air is R = 0.287 kPa.m^3/kg.K (Table A-1). Also, C_p = 1.005 kJ/kg.K for air at room temperature (Table A-2).

Analysis We first take the radiator as the system. This is a closed system since no mass enters or leaves. The energy balance for this closed system can be expressed as



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

$$-Q_{out} = \Delta U = m(u_2 - u_1) \quad \text{(since } W = \text{KE} = \text{PE} = 0\text{)}$$

$$Q_{out} = m(u_1 - u_2)$$

Using data from the steam tables (Tables A-4 through A-6), some properties are determined to be

$$\left.\begin{array}{l} P_1 = 200\text{kPa} \\ T_1 = 200^\circ\text{C} \end{array}\right\} \begin{array}{l} v_1 = 1.0803\text{m}^3/\text{kg} \\ u_1 = 2654.4\text{kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 100\text{kPa} \\ (v_2 = v_1) \end{array}\right\} \begin{array}{ll} v_f = 0.001043, & v_g = 1.6940\text{m}^3/\text{kg} \\ u_f = 417.36, & u_{fg} = 2088.7\text{kJ/kg} \end{array}$$

$$x_2 = \frac{v_2 - v_f}{v_{fg}} = \frac{1.0803 - 0.001043}{1.6940 - 0.001043} = 0.6375$$

$$u_2 = u_f + x_2 u_{fg} = 417.36 + 0.6375 \times 2088.7 = 1749 \text{ kJ/kg}$$

$$m = \frac{V_1}{v_1} = \frac{0.015 \text{ m}^3}{1.0803 \text{ m}^3/\text{kg}} = 0.0139 \text{ kg}$$

Substituting,

Q_{out} = (0.0139 kg)(2654.4 - 1749)kJ/kg = 12.6 kJ

The volume and the mass of the air in the room are V = 4×4×5 = 80 m³ and

$$m_{air} = \frac{P_1 V_1}{R T_1} = \frac{(100\text{kPa})(80\text{m}^3)}{(0.2870\text{kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(283\text{K})} = 98.5\text{kg}$$

The amount of fan work done in 30 min is

$$W_{fan,in} = \dot{W}_{fan,in} \Delta t = (0.120 \text{ kJ/s})(30 \times 60 \text{ s}) = 216 \text{ kJ}$$

We now take the air in the room as the system. The energy balance for this closed system is expressed as

$$E_{in} - E_{out} = \Delta E_{\text{system}}$$

$$Q_{in} + W_{fan,in} - W_{b,out} = \Delta U$$

$$Q_{in} + W_{fan,in} = \Delta H \cong m C_p (T_2 - T_1)$$

since the boundary work and ΔU combine into ΔH for a constant pressure expansion or compression process. It can also be expressed as

$$(\dot{Q}_{in} + \dot{W}_{fan,in})\Delta t = m C_{p,ave} (T_2 - T_1)$$

Substituting, $(12.6 \text{ kJ}) + (216 \text{ kJ}) = (98.5 \text{ kg})(1.005 \text{ kJ/kg°C})(T_2 - 10)\text{°C}$

which yields $T_2 = \mathbf{12.3°C}$

Therefore, the air temperature in the room rises from 10°C to 12.3°C in 30 min.

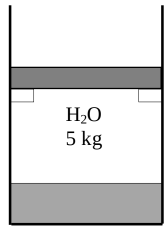
5-225 A cylinder equipped with a set of stops for the piston is initially filled with saturated liquid-vapor mixture of water a specified pressure. Heat is transferred to the water until the volume increases by 20%. The initial and final temperature, the mass of the liquid when the piston starts moving, and the work done during the process are to be determined, and the process is to be shown on a *P-v* diagram.

Assumptions **1** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **2** The thermal energy stored in the cylinder itself is negligible. **3** The compression or expansion process is quasi-equilibrium.

Analysis (*a*) Initially the system is a saturated mixture at 100 kPa pressure, and thus the initial temperature is

$$T_1 = T_{sat\,@100\text{ kPa}} = \mathbf{99.63^\circ C}$$

The total initial volume is



$$V_1 = m_f v_f + m_g v_g = 2\times 0.001043 + 3\times 1.6940 = 5.08\text{ m}^3$$

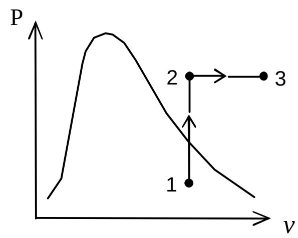
Then the total and specific volumes at the final state are

$$V_3 = 1.2V_1 = 1.2\times 5.08 = 6.10\text{ m}^3$$

$$v_3 = \frac{V_3}{m} = \frac{6.10\text{ m}^3}{5\text{ kg}} = 1.22\text{ m}^3/\text{kg}$$

Thus,

$$\left.\begin{aligned} P_3 &= 200\text{ kPa} \\ v_3 &= 1.22\text{ m}^3/\text{kg} \end{aligned}\right\} T_3 = \mathbf{259.0^\circ C}$$



(*b*) When the piston first starts moving, P_2 = 200 kPa and $V_2 = V_1$ = 5.08 m^3. The specific volume at this state is

$$v_2 = \frac{V_2}{m} = \frac{5.08\text{ m}^3}{5\text{kg}} = 1.016\text{ m}^3/\text{kg}$$

which is greater than v_g = 0.8857 m^3/kg at 200 kPa. Thus **no liquid** is left in the cylinder when the piston starts moving.

(*c*) No work is done during process 1-2 since $V_1 = V_2$. The pressure remains constant during process 2-3 and the work done during this process is

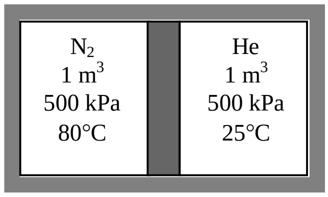
$$W_b = \int_2^3 P\,dV = P_2(V_3 - V_2) = (200\text{ kPa})(6.10 - 5.08)\text{m}^3\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) = \mathbf{204\,kJ}$$

5-226 An insulated cylinder is divided into two parts. One side of the cylinder contains N_2 gas and the other side contains He gas at different states. The final equilibrium temperature in the cylinder when thermal equilibrium is established is to be determined for the cases of the piston being fixed and moving freely.

Assumptions **1** Both N_2 and He are ideal gases with constant specific heats. **2** The energy stored in the container itself is negligible. **3** The cylinder is well-insulated and thus heat transfer is negligible.

Properties The gas constants and the constant volume specific heats are R = 0.2968 kPa.m^3/kg.K is C_v = 0.743 kJ/kg·°C for N_2, and R = 2.0769 kPa.m^3/kg.K is C_v = 3.1156 kJ/kg·°C for He (Tables A-1 and A-2)

Analysis The mass of each gas in the cylinder is



$$m_{N_2} = \left(\frac{P_1V_1}{RT_1}\right)_{N_2} = \frac{(500\text{ kPa})(1\text{ m}^3)}{(0.2968\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(353\text{ K})} = 4.77\text{ kg}$$

$$m_{He} = \left(\frac{P_1V_1}{RT_1}\right)_{He} = \frac{(500\text{ kPa})(1\text{ m}^3)}{(2.0769\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\text{ K})} = 0.808\text{ kg}$$

Taking the entire contents of the cylinder as our system, the 1st law relation can be written as

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

$$0 = \Delta U = (\Delta U)_{N_2} + (\Delta U)_{He}$$

$$0 = [mC_v(T_2 - T_1)]_{N_2} + [mC_v(T_2 - T_1)]_{He}$$

Substituting,

$$(4.77\text{kg})(0.743\text{kJ/kg}\cdot^\circ\text{C})(T_f - 80)^\circ\text{C} + (0.808\text{kg})(3.1156\text{kJ/kg}\cdot^\circ\text{C})(T_f - 25)^\circ\text{C} = 0$$

It gives $\qquad T_f$ = **57.2°C**

where T_f is the final equilibrium temperature in the cylinder.

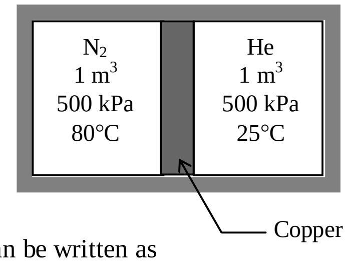
The answer would be the **same** if the piston were not free to move since it would effect only pressure, and not the specific heats.

Discussion Using the relation PV = NR_uT, it can be shown that the total number of moles in the cylinder is 0.170 + 0.202 = 0.372 kmol, and the final pressure is 510.6 kPa.

5-227 An insulated cylinder is divided into two parts. One side of the cylinder contains N_2 gas and the other side contains He gas at different states. The final equilibrium temperature in the cylinder when thermal equilibrium is established is to be determined for the cases of the piston being fixed and moving freely.

Assumptions **1** Both N_2 and He are ideal gases with constant specific heats. **2** The energy stored in the container itself, except the piston, is negligible. **3** The cylinder is well-insulated and thus heat transfer is negligible. **4** Initially, the piston is at the average temperature of the two gases.

Properties The gas constants and the constant volume specific heats are R = 0.2968 kPa.m^3/kg.K is C_v = 0.743 kJ/kg·°C for N_2, and R = 2.0769 kPa.m^3/kg.K is C_v = 3.1156 kJ/kg·°C for He (Tables A-1 and A-2). The specific heat of copper piston is C = 0.386 kJ/kg·°C (Table A-3).



Analysis The mass of each gas in the cylinder is

$$m_{N_2} = \left(\frac{P_1V_1}{RT_1}\right)_{N_2} = \frac{(500\text{kPa})(1\text{m}^3)}{(0.2968\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(353\text{K})} = 4.77\text{kg}$$

$$m_{He} = \left(\frac{P_1V_1}{RT_1}\right)_{He} = \frac{(500\text{kPa})(1\text{m}^3)}{(2.0769\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(353\text{K})} = 0.808\text{kg}$$

Taking the entire contents of the cylinder as our system, the 1st law relation can be written as

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

$$0 = \Delta U = (\Delta U)_{N_2} + (\Delta U)_{He} + (\Delta U)_{Cu}$$

$$0 = [mC_v(T_2 - T_1)]_{N_2} + [mC_v(T_2 - T_1)]_{He} + [mC(T_2 - T_1)]_{Cu}$$

where

$$T_{1,\,Cu} = (80 + 25) / 2 = 52.5°C$$

Substituting,

$$(4.77\text{kg})(0.743\text{kJ/kg}\cdot^\circ\text{C})(T_f - 80)^\circ\text{C} + (0.808\text{kg})(3.1156\text{kJ/kg}\cdot^\circ\text{C})(T_f - 25)^\circ\text{C}$$
$$+ (5.0\text{kg})(0.386\text{kJ/kg}\cdot^\circ\text{C})(T_f - 52.5)^\circ\text{C} = 0$$

It gives

$$T_f = \mathbf{56.0°C}$$

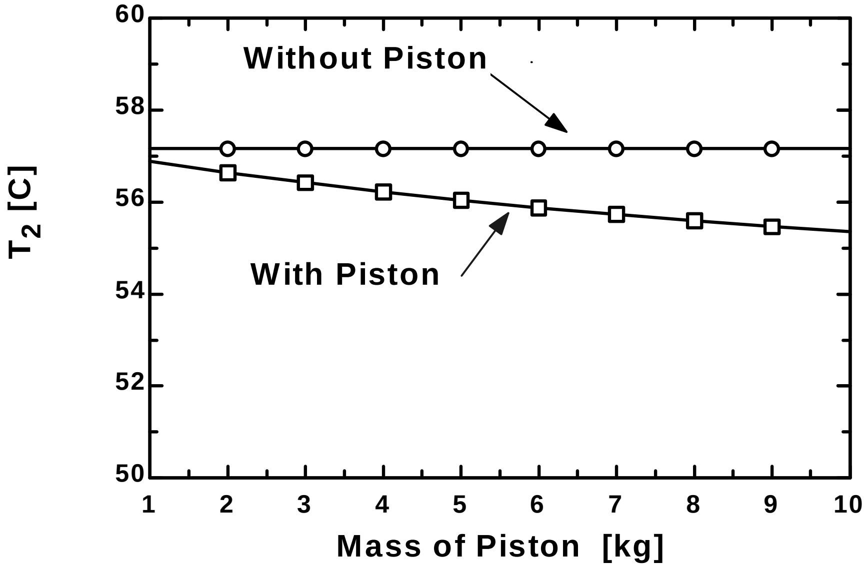
where T_f is the final equilibrium temperature in the cylinder.

The answer would be the **same** if the piston were not free to move since it would effect only pressure, and not the specific heats.

5-228 Problem 5-227 is reconsidered. The effect of the mass of the copper piston on the final equilibrium temperature as the mass of piston varies from 1 kg to 10 kg is to be investigated. The final temperature is to be plotted against the mass of piston.

```
"Knowns:"
R_u=8.314"[kJ/kmol-K]"
V_N2[1]=1"[m^3]"
Cv_N2=0.743"[kJ/kg-K]"  "From Table A-2(a) at 27C"
R_N2=0.2968"[kJ/kg-K]"  "From Table A-2(a)"
T_N2[1]=80"[C]"
P_N2[1]=500"[kPa]"
V_He[1]=1"[m^3]"
Cv_He=3.1156"[kJ/kg-K]" "From Table A-2(a) at 27C"
T_He[1]=25"[C]"
P_He[1]=500"[kPa]"
R_He=2.0769"[kJ/kg-K]"  "From Table A-2(a)"
m_Pist=5"[kg]"
Cv_Pist=0.386"[kJ/kg-K]"  "Use Cp for Copper from Table A-3(b) at 27C"

"Solution:"
"mass calculations:"
P_N2[1]*V_N2[1]=m_N2*R_N2*(T_N2[1]+273)
P_He[1]*V_He[1]=m_He*R_He*(T_He[1]+273)
"The entire cylinder is considered to be a closed system, neglecting the piston."
"Conservation of Energy for the closed system:"
"E_in - E_out = DELTAE_negPist,  we neglect DELTA KE and DELTA PE for the cylinder."
E_in - E_out = DELTAE_neglPist
E_in =0"[kJ]"
E_out = 0"[kJ]"

"At the final equilibrium state, N2 and He will have a common temperature."
DELTAE_neglPist= m_N2*Cv_N2*(T_2_neglPist-T_N2[1])+m_He*Cv_He*(T_2_neglPist-
T_He[1])"[kJ]"

"The entire cylinder is considered to be a closed system, including the piston."
"Conservation of Energy for the closed system:"
"E_in - E_out = DELTAE_withPist,  we neglect DELTA KE and DELTA PE for the cylinder."
E_in - E_out = DELTAE_withPist

"At the final equilibrium state, N2 and He will have a common temperature."
DELTAE_withPist= m_N2*Cv_N2*(T_2_withPist-T_N2[1])+m_He*Cv_He*(T_2_withPist-
T_He[1])+m_Pist*Cv_Pist*(T_2_withPist-T_Pist[1])"[kJ]"
T_Pist[1]=(T_N2[1]+T_He[1])/2"[C]"

"Total volume of gases:"
V_total=V_N2[1]+V_He[1]"[m^3]"

"Final pressure at equilibrium:"
"Neglecting effect of piston, P_2 is:"
P_2_neglPist*V_total=N_total*R_u*(T_2_neglPist+273)"[kPa]"

"Including effect of piston, P_2 is:"
N_total=m_N2/molarmass(nitrogen)+m_He/molarmass(Helium)"[kmol]"
P_2_withPist*V_total=N_total*R_u*(T_2_withPist+273)"[kPa]"
```

m_{Pist} [kg]	$T_{2,neglPist}$ [C]	$T_{2,withPist}$ [C]
1	57.17	56.89
2	57.17	56.64
3	57.17	56.42
4	57.17	56.22
5	57.17	56.04
6	57.17	55.88
7	57.17	55.73
8	57.17	55.59
9	57.17	55.47
10	57.17	55.35



5-229 A relation for the explosive energy of a fluid is given. A relation is to be obtained for the explosive energy of an ideal gas, and the value for air at a specified state is to be evaluated.

Analysis The explosive energy per unit volume is given as

$$e_{\text{explosion}} = \frac{u_1 - u_2}{v_1}$$

For an ideal gas, $u_1 - u_2 = C_v(T_1 - T_2)$

$$C_p - C_v = R$$

$$v_1 = \frac{RT_1}{P_1}$$

and thus

$$\frac{C_v}{R} = \frac{C_v}{C_p - C_v} = \frac{1}{C_p / C_v - 1} = \frac{1}{k-1}$$

Substituting,

$$e_{\text{explosion}} = \frac{C_v(T_1 - T_2)}{RT_1 / P_1} = \frac{P_1}{k-1}\left(1 - \frac{T_2}{T_1}\right)$$

which is the desired result.

Using the relation above, the total explosive energy of 20 m³ of air at 5 MPa and 100°C when the surroundings are at 20°C is determined to be

$$E_{\text{explosion}} = Ve_{\text{explosion}} = \frac{P_1V_1}{k-1}\left(1 - \frac{T_2}{T_1}\right) = \frac{(5000 kPa)(20 m^3)}{1.4-1}\left(1 - \frac{293 K}{373 K}\right)\left(\frac{1 kJ}{1 kPa \cdot m^3}\right) = \mathbf{53{,}619 kJ}$$

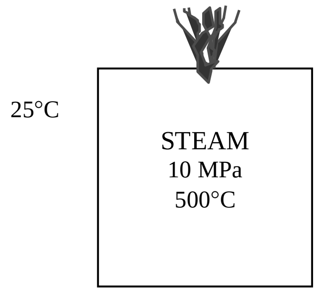
5-230 Using the relation for explosive energy given in the previous problem, the explosive energy of steam and its TNT equivalent at a specified state are to be determined.

Assumptions Steam condenses and becomes a liquid at room temperature after the explosion.

Properties The properties of steam at the initial and the final states are (Table A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 10\text{MPa} \\ T_1 = 500^{\circ}\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.03279\text{m}^3/\text{kg} \\ u_1 = 3045.8\text{kJ/kg} \end{array}$$

$$\left.\begin{array}{l} T_2 = 25^{\circ}\text{C} \\ \textit{Comp.liquid} \end{array}\right\} u_2 \cong u_{f@25^{\circ}\text{C}} = 104.88\text{kJ/kg}$$



Analysis The mass of the steam is

$$m = \frac{V}{v_1} = \frac{20\text{ m}^3}{0.03279\text{ m}^3/\text{kg}} = 609.9\text{ kg}$$

Then the total explosive energy of the steam is determined from

$$E_{\text{explosive}} = m(u_1 - u_2) = (609.9\text{kg})(3045.8 - 104.88)\text{kJ/kg} = \mathbf{1{,}793{,}667\,kJ}$$

which is equivalent to

$$\frac{1{,}793{,}667\text{ kJ}}{3250\text{ kJ/kg of TNT}} = \mathbf{552\ kg\ of\ TNT}$$

5-231 Solar energy is to be stored as sensible heat using phase-change materials, granite rocks, and water. The amount of heat that can be stored in a 5-m^3 = 5000 L space using these materials as the storage medium is to be determined.

Assumptions **1** The materials have constant properties at the specified values. **2** No allowance is made for voids, and thus the values calculated are the upper limits.

Analysis The amount of energy stored in a medium is simply equal to the increase in its internal energy, which, for incompressible substances, can be determined from $\Delta U = mC(T_2 - T_1)$.

(*a*) The latent heat of glaubers salts is given to be 329 kJ/L. Disregarding the sensible heat storage in this case, the amount of energy stored is becomes

$$\Delta U_{\text{salt}} = mh_{\text{if}} = (5000 \text{ L})(329 \text{ kJ/L}) = \mathbf{1{,}645{,}000 \text{ kJ}}$$

This value would be even larger if the sensible heat storage due to temperature rise is considered.

(*b*) The density of granite is 2700 kg/m^3 (Table A-3), and its specific heat is given to be C = 2.32 kJ/kg.°C. Then the amount of energy that can be stored in the rocks when the temperature rises by 20°C becomes

$$\Delta U_{\text{rock}} = \rho V C \Delta T = (2700 \text{ kg/m}^3)(5 \text{ m}^3)(2.32 \text{ kJ/kg.°C})(20\text{°C}) = \mathbf{626{,}400 \text{ kJ}}$$

(*c*) The density of water is about 1000 kg/m^3 (Table A-3), and its specific heat is given to be C = 4.0 kJ/kg.°C. Then the amount of energy that can be stored in the water when the temperature rises by 20°C becomes

$$\Delta U_{\text{rock}} = \rho V C \Delta T = (1000 \text{ kg/m}^3)(5 \text{ m}^3)(4.0 \text{ kJ/kg.°C})(20\text{°C}) = \mathbf{400{,}00 \text{ kJ}}$$

Discussion Note that the greatest amount of heat can be stored in phase-change materials essentially at constant temperature. Such materials are not without problems, however, and thus they are not widely used.

5-232 The feedwater of a steam power plant is preheated using steam extracted from the turbine. The ratio of the mass flow rates of the extracted seam the feedwater 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. **4** Heat loss from the device to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid.

Properties The enthalpies of steam and feedwater at are (Tables A-4 through A-6)

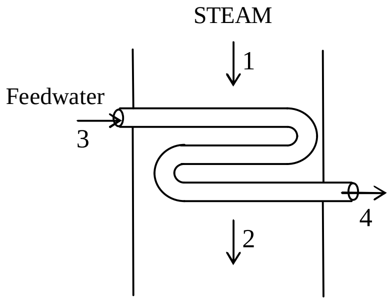
$$\left.\begin{array}{l} P_1 = 1.2\text{MPa} \\ T_1 = 250^\circ\text{C} \end{array}\right\} h_1 = 2827.9\text{kJ/kg}$$

$$\left.\begin{array}{l} P_1 = 1\text{MPa} \\ sat.liquid \end{array}\right\} \begin{array}{l} h_2 = h_{f@1\text{MPa}} = 762.81\text{kJ/kg} \\ T_2 = 179.91^\circ\text{C} \end{array}$$

and

$$\left.\begin{array}{l} P_3 = 2.5\text{MPa} \\ T_3 = 50^\circ\text{C} \end{array}\right\} h_3 \cong h_{f@50^\circ\text{C}} = 209.33\text{kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 2.5\text{MPa} \\ T_4 = T_2 - 10 \cong 170^\circ\text{C} \end{array}\right\} h_4 \cong h_{f@170^\circ\text{C}} = 719.2\text{kJ/kg}$$



Analysis We take the 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):

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

Energy balance (for the heat exchanger):

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

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

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

Combining the two, $\dot{m}_s(h_2 - h_1) = \dot{m}_{fw}(h_3 - h_4)$

Dividing by $\dot{m}_{fw}$ and substituting,

$$\frac{\dot{m}_s}{\dot{m}_{fw}} = \frac{h_3 - h_4}{h_2 - h_1} = \frac{(719.2 - 209.33)\text{kJ/kg}}{(2827.9 - 762.81)\text{kJ/kg}} = \mathbf{0.247}$$

5-233 A building is to be heated by a 30-kW electric resistance heater placed in a duct inside. The time it takes to raise the interior temperature from 14°C to 24°C, and the average mass flow rate of air as it passes through the heater in the duct are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible. **4** The heating duct is adiabatic, and thus heat transfer through it is negligible. **5** No air leaks in and out of the building.

Properties The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1). The specific heats of air at room temperature are C_p = 1.005 and C_v = 0.718 kJ/kg·K (Table A-2).

Analysis (*a*) The total mass of air in the building is

$$m = \frac{P_1 V_1}{RT_1} = \frac{(95\ \text{kPa})(400\ \text{m}^3)}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(287\ \text{K})} = 461.3\ \text{kg}.$$

We first take the *entire building* as our system, which is a closed system since no mass leaks in or out. The time required to raise the air temperature to 24°C is determined by applying the energy balance to this constant volume closed system:

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

$$W_{e,in} + W_{fan,in} - Q_{out} = \Delta U \quad (\text{since } \Delta\text{KE} = \Delta\text{PE} = 0)$$

$$\Delta t\left(\dot{W}_{e,in} + \dot{W}_{fan,in} - \dot{Q}_{out}\right) = mC_{v,ave}(T_2 - T_1)$$

Solving for Δt gives

$$\Delta t = \frac{mC_{v,ave}(T_2 - T_1)}{\dot{W}_{e,in} + \dot{W}_{fan,in} - \dot{Q}_{out}} = \frac{(461.3\ \text{kg})(0.718\ \text{kJ/kg}\cdot{}^\circ\text{C})(24-14)^\circ\text{C}}{(30\ \text{kJ/s}) + (0.25\ \text{kJ/s}) - (450/60\ \text{kJ/s})} = \mathbf{146\ \ s}$$

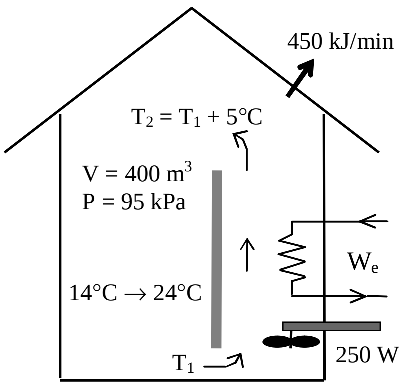
(*b*) We now take the *heating duct* as the system, which is a control volume since mass crosses the boundary. There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The energy balance for this adiabatic steady-flow system can be expressed in the rate form as

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

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

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

$$\dot{W}_{e,in} + \dot{W}_{fan,in} = \dot{m}(h_2 - h_1) = \dot{m}C_p(T_2 - T_1)$$



Thus,

$$\dot{m} = \frac{\dot{W}_{e,in} + \dot{W}_{fan,in}}{C_p \Delta T} = \frac{(30 + 0.25)\ \text{kJ/s}}{(1.005\ \text{kJ/kg}\cdot{}^\circ\text{C})(5^\circ\text{C})} = \mathbf{6.02\ kg/s}$$

5-234 [*Also solved by EES on enclosed CD***]** An insulated cylinder equipped with an external spring initially contains air. The tank is connected to a supply line, and air is allowed to enter the cylinder until its volume doubles. The mass of the air that entered and the final temperature in the cylinder are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** The expansion process is quasi-equilibrium. **3** Kinetic and potential energies are negligible. **4** The spring is a linear spring. **5** The device is insulated and thus heat transfer is negligible. **6** Air is an ideal gas with constant specific heats.

Properties The gas constant of air is R = 0.287 kJ/kg·K (Table A-1). The specific heats of air at room temperature are C_v = 0.718 and C_p = 1.005 kJ/kg·K (Table A-2a). Also, $u = C_vT$ and $h = C_pT$.

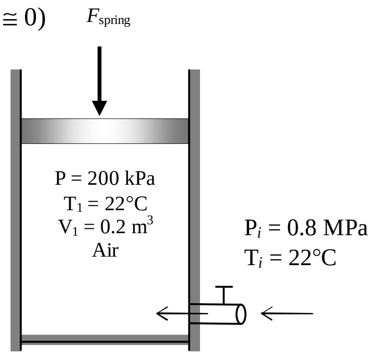
Analysis We take the cylinder as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{in} - m_{out} = \Delta m_{\text{system}} \quad \to \quad m_i = m_2 - m_1$

Energy balance:

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

$$m_i h_i = W_{b,out} + m_2u_2 - m_1u_1 \quad (\text{since } Q \cong ke \cong pe \cong 0)$$



Combining the two relations, $\quad (m_2 - m_1)h_i = W_{b,out} + m_2u_2 - m_1u_1$

or, $\quad (m_2 - m_1)C_pT_i = W_{b,out} + m_2C_vT_2 - m_1C_vT_1$

The initial and the final masses in the tank are

$$m_1 = \frac{P_1V_1}{RT_1} = \frac{(200\,\text{kPa})(0.2\,\text{m}^3)}{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(295\,\text{K})} = 0.472\,\text{kg}$$

$$m_2 = \frac{P_2V_2}{RT_2} = \frac{(600\,\text{kPa})(0.4\,\text{m}^3)}{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})\text{T}_2} = \frac{836.2}{\text{T}_2}$$

Then from the mass balance becomes $\quad m_i = m_2 - m_1 = \dfrac{836.2}{T_2} - 0.472$

The spring is a linear spring, and thus the boundary work for this process can be determined from

$$W_b = Area = \frac{P_1 + P_2}{2}(V_2 - V_1) = \frac{(200 + 600)\,\text{kPa}}{2}(0.4 - 0.2)\,\text{m}^3 = 80\,\text{kJ}$$

Substituting into the energy balance, the final temperature of air T_2 is determined to be

$$-80 = -\left(\frac{836.2}{T_2} - 0.472\right)(1.005)(295) + \left(\frac{836.2}{T_2}\right)(0.718)(T_2) - (0.472)(0.718)(295)$$

It yields $\quad T_2$ = **344.1 K**

Thus, $\quad m_2 = \dfrac{836.2}{\text{T}_2} = \dfrac{836.2}{344.1} = 2.430\,\text{kg}$

and $\quad m_i = m_2 - m_1$ = 2.430 - 0.472 = **1.958 kg**

5-235 Pressurized air stored in a large cave is to be used to drive a turbine. The amount of work delivered by the turbine for specified turbine exit conditions is to be determined. √

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the exit temperature (and enthalpy) of air remains constant. **2** Kinetic and potential energies are negligible. **3** The system is insulated and thus heat transfer is negligible. **4** Air is an ideal gas with constant specific heats at room temperature.

Properties The gas constant of air is R = 0.287 kJ/kg·K (Table A-1). The specific heats of air at room temperature are C_v = 0.718 and C_p = 1.005 kJ/kg·K (Table A-2a). Also, $u = C_vT$ and $h = C_pT$.

Analysis We take the *cave* as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $$m_{in} - m_{out} = \Delta m_{\text{system}} \quad \rightarrow \quad m_e = m_1 - m_2$$

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

$$-m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong W \cong ke \cong pe \cong 0)$$

Combining the two: $$(m_1 - m_2)h_e + m_2u_2 - m_1u_1 = 0$$

or, $$\left(\frac{P_1V}{RT_1} - \frac{P_2V}{RT_2}\right)C_p\frac{T_1+T_2}{2} + \frac{P_2V}{RT_2}C_vT_2 - \frac{P_1V}{RT_1}C_vT_1 = 0$$

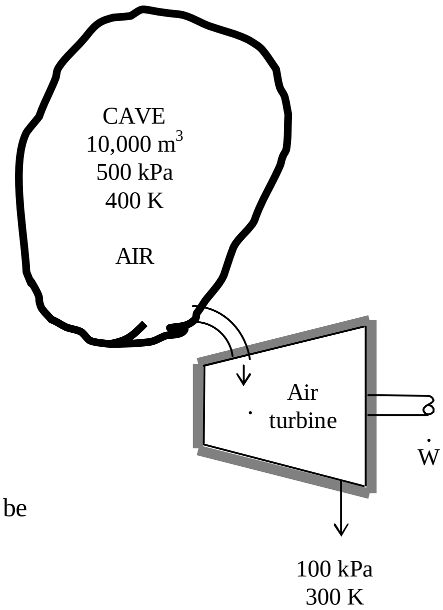
Multiply by R/VC_v: $$\left(\frac{P_1}{T_1} - \frac{P_2}{T_2}\right)k\frac{T_1+T_2}{2} + P_2 - P_1 = 0$$

Substituting, $$\left(\frac{500}{400} - \frac{300}{T_2}\right)(1.4)\frac{400+T_2}{2} + 300 - 500 = 0$$

$$T_2^2 - 68.57T_2 - 96{,}000 = 0$$

It yields

$$T_2 = 346 \text{ K}$$



The initial and the final masses of air in the cave are determined to be

$$m_1 = \frac{P_1V}{RT_1} = \frac{(500\text{kPa})(10^4\text{m}^3)}{(0.287\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(400\text{K})} = 43{,}554\text{kg}$$

$$m_2 = \frac{P_2V}{RT_2} = \frac{(300\text{kPa})(10^4\text{m}^3)}{(0.287\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(346\text{K})} = 30{,}211\text{kg}$$

Then from the mass balance we get $$m_e = m_1 - m_2 = 43{,}554 - 30{,}211 = 13{,}343 \text{ kg}$$

The average temperature at the turbine inlet is (400 + 346)/2 = 373 K. Taking the turbine as system and assuming the air properties at the turbine inlet to be constant at the average temperature, the turbine work output is determined from the steady-flow energy balance $\dot{E}_{in} = \dot{E}_{out}$ to be

$$m_ih_i - m_eh_e - W_{out} = 0$$

or $$W_{out} = m_e(h_i - h_e)_{\text{turbine}} = (13{,}343\text{ kg})(373.7 - 300.19)\text{kJ/kg} = \mathbf{981\,MJ}$$

5-236E Steam is decelerated in a diffuser from a velocity of 500 ft/s to 100 ft/s. The mass flow rate of steam, the rate of heat transfer, and the inlet area of the diffuser are to be determined.

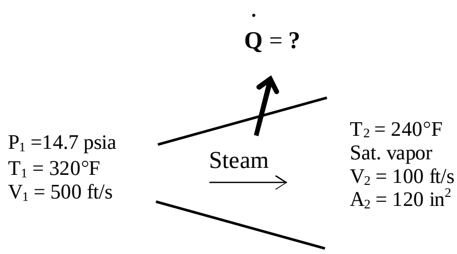
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** There are no work interactions.

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

$$\left.\begin{array}{l} P_1 = 14.7\text{ psia} \\ T_1 = 320^\circ\text{F} \end{array}\right\} \begin{array}{l} v_1 = 31.36\text{ ft}^3/\text{lbm} \\ h_1 = 1202.1\text{ Btu/lbm} \end{array}$$

and

$$\left.\begin{array}{l} T_2 = 240^\circ\text{F} \\ sat.vapor \end{array}\right\} \begin{array}{l} v_2 = 16.327\text{ ft}^3/\text{lbm} \\ h_2 = 1160.7\text{ Btu/lbm} \end{array}$$



Analysis (*a*) The mass flow rate of the steam can be determined from its definition to be

$$\dot{m} = \frac{1}{v_2}\mathbf{V}_2 A_2 = \frac{1}{16.327\text{ ft}^3/\text{lbm}}(100\text{ ft/s})(120/144\text{ ft}^2) = \mathbf{5.104\ lbm/s}$$

(*b*) We take diffuser as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

$$\dot{Q}_{\text{in}} + \dot{m}(h_1 + \mathbf{V}_1^2/2) = \dot{m}(h_2 + \mathbf{V}_1^2/2) \quad (\text{since } \dot{W} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}\left(h_2 - h_1 + \frac{\mathbf{V}_2^2 - \mathbf{V}_1^2}{2}\right)$$

Substituting,

$$\dot{Q}_{in} = (5.104\text{ lbm/s})\left(1160.7 - 1202.1 + \frac{(100\text{ ft/s})^2 - (500\text{ ft/s})^2}{2}\left(\frac{1\text{ Btu/lbm}}{25{,}037\text{ ft}^2/\text{s}^2}\right)\right) = \mathbf{-235.8\ Btu/s}$$

(*c*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Then the inlet area of the diffuser becomes

$$\dot{m} = \frac{1}{v_1}\mathbf{V}_1 A_1 \longrightarrow A_1 = \frac{\dot{m} v_1}{\mathbf{V}_1} = \frac{(5.104\text{ lbm/s})(31.36\text{ ft}^3/\text{lbm})}{500\text{ ft/s}} = \mathbf{0.320\ ft^2}$$

5-237 20% of the volume of a pressure cooker is initially filled with liquid water. Heat is transferred to the cooker at a rate of 400 W. The time it will take for the cooker to run out of liquid is to be determined.

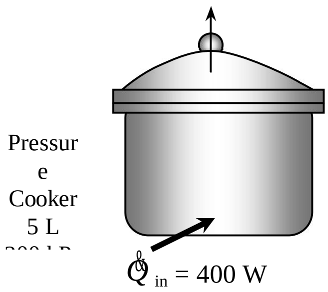
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid leaving the device remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved.

Properties The properties of water are (Tables A-4 through A-6)

$$P_1 = 200 \text{ kPa} \quad \rightarrow \quad v_f = 0.001061 \text{ m}^3/\text{kg}, v_g = 0.8857 \text{ m}^3/\text{kg}$$
$$u_f = 504.49 \text{ kJ/kg}, u_g = 2529.5 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 200 \text{ kPa} \\ \text{sat.vapor} \end{array}\right\} \begin{array}{l} v_2 = v_{g@200\text{kPa}} = 0.8857 \text{ m}^3/\text{kg} \\ u_2 = u_{g@200\text{kPa}} = 2529.5 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_e = 200 \text{ kPa} \\ \text{sat.vapor} \end{array}\right\} h_e = h_{g@200 \text{ kPa}} = 2706.7 \text{ kJ/kg}$$



Analysis We take the cooker as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{in} - m_{out} = \Delta m_{\text{system}} \quad \rightarrow \quad m_e = m_1 - m_2$

Energy balance:

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

$$Q_{in} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

The initial mass, initial internal energy, and final mass in the tank are

$$m_1 = m_f + m_g = \frac{V_f}{v_f} + \frac{V_g}{v_g} = \frac{0.001 \text{ m}^3}{0.001061 \text{ m}^3/\text{kg}} + \frac{0.004 \text{ m}^3}{0.8857 \text{ m}^3/\text{kg}} = 0.9425 + 0.0045 = 0.9470 \text{ kg}$$

$$U_1 = m_1 u_1 = m_f u_f + m_g u_g = (0.9425)(504.49) + (0.0045)(2529.5) = 486.86 \text{ kJ}$$

$$m_2 = \frac{V}{v_2} = \frac{0.005 \text{ m}^3}{0.8857 \text{ m}^3/\text{kg}} = 0.0056 \text{ kg}$$

From mass and energy balances,

$$m_e = m_1 - m_2 = 0.9470 - 0.0056 = 0.9414 \text{ kg}$$

and

$$\dot{Q}\Delta t = m_e h_e + m_2 u_2 - m_1 u_1$$
$$(0.4 \text{ kJ/s})\Delta t = (0.9414 \text{ kg})(2706.7 \text{ kJ/kg}) + (0.0056 \text{ kg})(2529.5 \text{ kJ/kg}) - 486.86 \text{ kJ}$$

It yields Δt = 5188 s = **1.44 h**

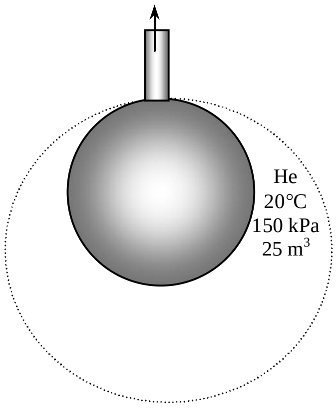
5-238 A balloon is initially filled with pressurized helium gas. Now a valve is opened, and helium is allowed to escape until the pressure inside drops to atmospheric pressure. The final temperature of helium in the balloon and the mass of helium that has escaped are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process by assuming the properties of helium that escape to be constant at average conditions. **2** Kinetic and potential energies are negligible. **3** There are no work interactions other than boundary work. **4** Helium is an ideal gas with constant specific heats. **5** Heat transfer is negligible.

Properties The gas constant of helium is R =2.0769 kPa.m^3/kg.K (Table A-1). The specific heats of helium are C_p = 5.1926 and C_v = 3.1156 kJ/kg·K (Table A-2).

Analysis (*a*) The properties of helium leaving the balloon are changing during this process. But we will treat them as a constant at the average temperature. Thus $T_e \cong (T_1 + T_2)/2$. Also $h = C_pT$ and $u = C_vT$.

We take the balloon as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, the mass and energy balances for this uniform-flow system can be expressed as



Mass balance: $$m_{in} - m_{out} = \Delta m_{\text{system}} \quad \rightarrow \quad m_e = m_1 - m_2$$

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

$$W_{b,in} - m_e h_e = m_2u_2 - m_1u_1 \quad (\text{since } Q \cong ke \cong pe \cong 0)$$

or $$W_{b,in} = m_e C_p \frac{T_1 + T_2}{2} + m_2C_vT_2 - m_1C_vT_1$$

The final volume of helium is

$$P_1 = -100 + bV_1 \longrightarrow b = (P_1 + 100)/V_1 = (150 + 100)/25 = 10$$

$$P_2 = -100 + 10V_2 \longrightarrow V_2 = (P_2 + 100)/10 = (100 + 100)/10 = 20\text{m}^3$$

The initial and the final masses of helium in the balloon are

$$m_1 = \frac{P_1V_1}{RT_1} = \frac{(150\text{kPa})(25\text{m}^3)}{(2.0769\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\text{K})} = 6.162\text{kg}$$

$$m_2 = \frac{P_2V_2}{RT_2} = \frac{(100\text{kPa})(20\text{m}^3)}{(2.0769\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})T_2} = \frac{962.974}{T_2}$$

Then from the mass balance we have $$m_i = m_2 - m_1 = 6.162 - \frac{962.974}{T_2}$$

The boundary work done during this process is

$$W_{b,in} = \frac{P_1 + P_2}{2}(V_1 - V_2) = \frac{(150 + 100)\text{kPa}}{2}(25 - 20)\text{m}^3 = 625\text{kJ}$$

Then from the energy balance,

$$W_{b,in} = m_eC_p\frac{T_1 + T_2}{2} + m_2C_vT_2 - m_1C_vT_1$$

$$625 = \left(6.162 - \frac{962.974}{T_2}\right)(5.1926)\frac{293 + T_2}{2} + \left(\frac{962.974}{T_2}\right)(3.1156)(T_2) - (6.162)(3.1156)(293)$$

It yields $$T_2^2 - 66.37T_2 - 45{,}789 = 0$$

Solving for T_2 yields $$T_2 = \mathbf{249.7\ K}$$

(*b*) The amount of helium that has escaped is

$$m_e = m_1 - m_2 = 6.162 - \frac{962.974}{T_2} = 6.162 - \frac{962.974}{249.7} = \mathbf{2.306\ kg}$$

5-239 Problem 5-238 is to be solved using a stepwise approach. Use (a) 5, (b) 20, and (c) 50 increments for pressure between the initial value of 150 kPa and the final value of 100 kPa. Take the starting point of the first step to be the initial state of the helium (150 kPa, 20°C, and 25 m^3). The starting point of the second step is the state of the helium at the end of the first step, and so on. Compare your results with those obtained by using the uniform-flow approximation (i.e., a one-step solution).

```
Procedure EnergyFunc(P1,T1, V1,P2,T2,a,b:V2,W_b,mout,F,T_out)
C_P = 5.1926"[kJ/kg-K ]"
C_V = 3.1156 "[kJ/kg-K ]"
R=2.0769 "[kPa-m^3/kg-K]"
V2 = (P2-a)/b"[m^3]"
Tout=(T1+T2)/2
T_out=Tout

"Analysis:"
"Mass balance:"
m1=P1*V1/(R*T1)
m2=P2*V2/(R*T2)
min = 0"[kg]"
mout = min - m2 + m1

"Energy balance:"
"Boundary Work: Due to pressure changing linearly with volume.  Note the minus sign for
work in"
W_b = -(P1+P2)/2*(V2-V1)
u1=C_V*T1"[kJ/kg]"
u2= C_V*T2"[kJ/kg]"
hout = C_P*Tout"[kJ/kg]"
Ein= W_b"[kJ]"
Eout = mout*hout"[kJ]"
DELTAEsys = m2*u2-m1*u1
F=Ein - Eout - DELTAEsys
End

Procedure StepSolution(P_1,T_1,
V_1,P_2,DELTAP_step,a,b:V_2_step,T_2_step,W_b_step,m_out_step,T_out_step)

R=2.0769 "[kPa-m^3/kg-K]"

P_1_step=P_1
T_1_step=T_1
 V_1_step=V_1
DELTAT=1
W_b_step =0
m_out_step=0

Repeat
    P_2_step=P_1_step-DELTAP_step
   T_2_step=T_1_step-DELTAT
   T2=T_2_step
   T_out_step=(T_1_step + T2)/2
   V_1_step_a=V_1_step
   T_1_step_a=T_1_step
```

```
    V_2_step = (P_2_step-a)/b"[m^3]"

    Call EnergyFunc(P_1_step,T_1_step, V_1_step,P_2_step,T2,a,b:V2,W_b,mout,F,Tout)
    F1=F
    T22 = 0.99*T2
     ICT = 0
     Repeat

         ICT = ICT +1

         Call EnergyFunc(P_1_step,T_1_step,
V_1_step,P_2_step,T22,a,b:V2,W_b,mout,F,Tout)
         F2=F

         T_2_step = T22-F2/(F2-F1)*(T22-T2)

         T2=T22
         F1=F2
         T22=T_2_step
     Until (abs(F2)<1E-4) or (ICT>100)

  T_out_step = Tout
  V_2_step = V2
  W_b_step=W_b_step+ W_b
  m_out_step=m_out_step+(P_1_step*V_1_step/(R*T_1_step)-
P_2_step*V_2_step/(R*T_2_step))
  V_1_step=V_2_step
  T_1_step=T_2_step
  P_1_step=P_2_step

Until (P_2_step<=P_2)
END

"UNIFORM_FLOW SOLUTION:"
"Knowns:"
C_P = 5.1926"[kJ/kg-K ]"
C_V = 3.1156 "[kJ/kg-K ]"
R=2.0769 "[kPa-m^3/kg-K]"
P_1= 150"[kPa]"
T_1 = 293"[K]"
V_1 = 25"[m^3]"
"P = a + b*V, where"
a = -100 "[kPa]"
b=(P_1-a)/V_1"[kPa/m^3]"
T_out =  (T_1 + T_2)/2"[K]"
{DELTAP_step=50"[kPa]"}
P_2= 100"[kPa]"
V_2 = (P_2-a)/b"[m^3]"

"Analysis:
We take the balloon as the system, which is a control volume since mass crosses the
boundary. Noting that the microscopic energies of flowing and nonflowing fluids are
represented by enthalpy h and internal energy u, the mass and energy balances for this
uniform-flow system can be expressed as"

"Mass balance:"
m_in = 0"[kg]"
m_in - m_out = m_2 - m_1
```

```
"Energy balance:"
E_in - E_out = DELTAE_sys
E_in = W_b_in"[kJ]"
E_out = m_out*h_out"[kJ]"
h_out  = C_P*T_out"[kJ/kg]"
DELTAE_sys = m_2*u_2-m_1*u_1
u_1=C_V*T_1"[kJ/kg]"
u_2= C_V*T_2"[kJ/kg]"

"The volume flow rates of air are determined to be:"
P_1*V_1=m_1*R*T_1
P_2*V_2=m_2*R*T_2
"Boundary Work: Due to pressure changing linearly with volume.  Note the minus sign for
work in"
W_b_in = -(P_1+P_2)/2*(V_2-V_1)  "The minus sign is for work in."
Call StepSolution(P_1,T_1, V_1,P_2,DELTAP_step,a,b :
V_2_step,T_2_step,W_b_step,m_out_step,T_out_step)
```

SOLUTION

```
Variables in Main
a=-100 [kPa]
b=10 [kPa/m^3]
C_P=5.193 [kJ/kg-K ]
C_V=3.116 [kJ/kg-K ]
DELTAE_sys=-2625 [kJ]
DELTAP_step=50 [kPa]
E_in=625 [kJ]
E_out=3250 [kJ]
h_out=1409 [kJ/kg]
m_1=6.162 [kg]
m_2=3.856 [kg]
m_in=0 [kg]
m_out=2.307 [kg]
m_out_step=2.307 [kg]
P_1=150 [kPa]
P_2=100 [kPa]
R=2.077 [kPa-m^3/kg-K]
T_1=293 [K]
T_2=249.7 [K]
T_2_step=249.7 [K]
T_out=271.4 [K]
T_out_step=271.4 [K]
u_1=912.9 [kJ/kg]
u_2=778.1 [kJ/kg]
V_1=25 [m^3]
V_2=20 [m^3]
V_2_step=20
W_b_in=625 [kJ]
W_b_step=625 [kJ]
```

5- 240 … 2-246 Design and Essay Problems

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