

The Quality of Energy

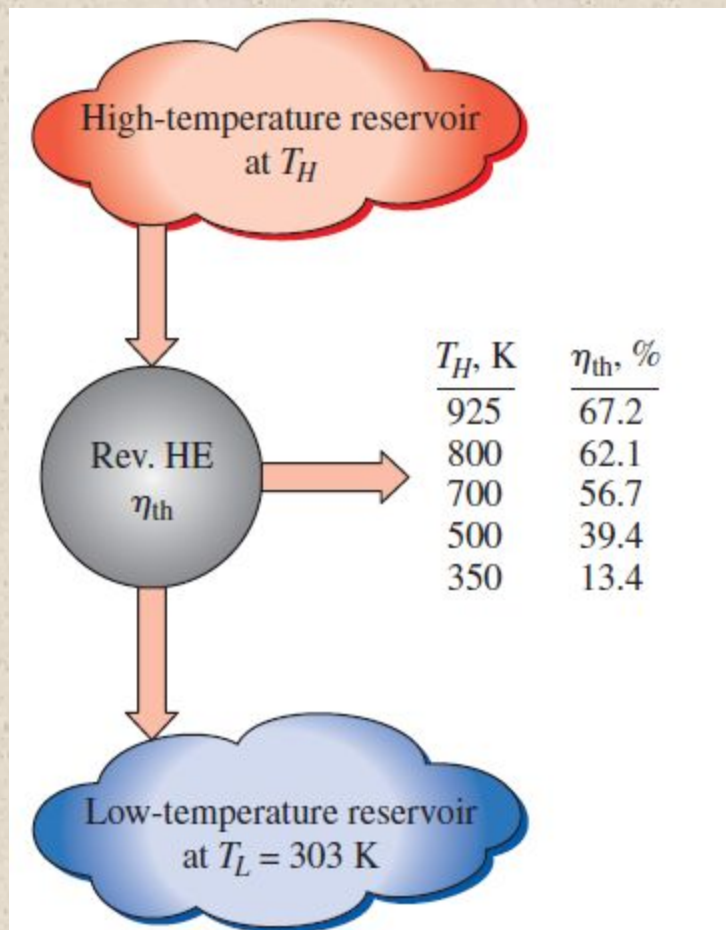


FIGURE 6–48

The fraction of heat that can be converted to work as a function of source temperature (for $T_L = 303 \text{ K}$).

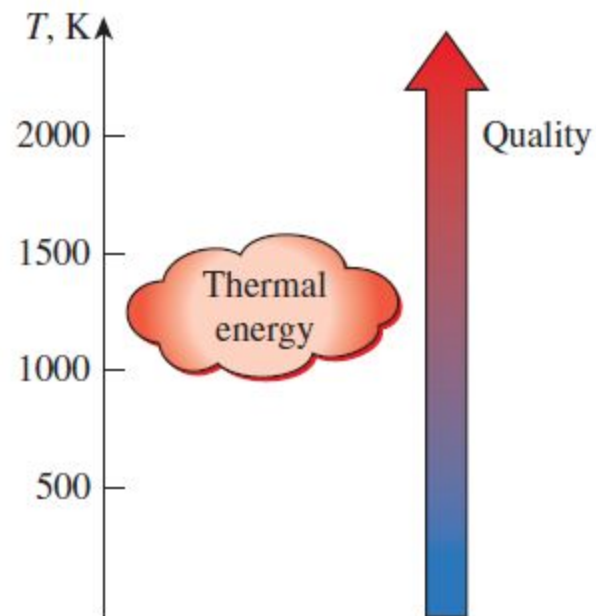


FIGURE 6–49

The higher the temperature of the thermal energy, the higher its quality.

$$\eta_{th,rev} = 1 - \frac{T_L}{T_H}$$

Can we use $^{\circ}\text{C}$ unit for temperature here?

How do you increase the thermal efficiency of a Carnot heat engine?
How about for actual heat engines?

THE CARNOT REFRIGERATOR AND HEAT PUMP

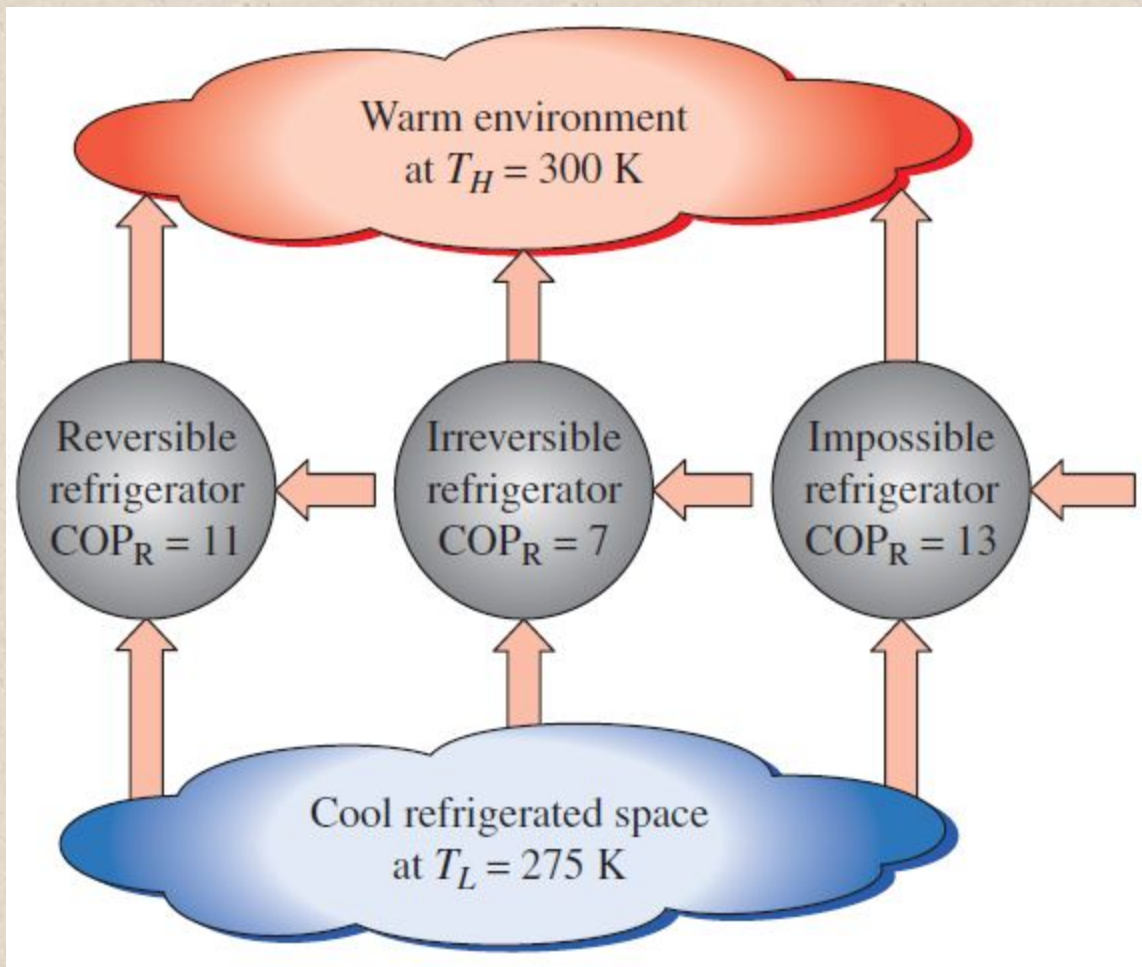


FIGURE 6-50

No refrigerator can have a higher COP than a reversible refrigerator operating between the same temperature limits.

Any refrigerator or heat pump

$$\text{COP}_R = \frac{1}{Q_H/Q_L - 1}$$

$$\text{COP}_{\text{HP}} = \frac{1}{1 - Q_L/Q_H}$$

Carnot refrigerator or heat pump

$$\text{COP}_{R,\text{rev}} = \frac{1}{T_H/T_L - 1}$$

$$\text{COP}_{\text{HP},\text{rev}} = \frac{1}{1 - T_L/T_H}$$

How do you increase the COP of a Carnot refrigerator or heat pump? How about for actual ones?

$$\text{COP}_R \begin{cases} < \text{COP}_{R,\text{rev}} & \text{irreversible refrigerator} \\ = \text{COP}_{R,\text{rev}} & \text{reversible refrigerator} \\ > \text{COP}_{R,\text{rev}} & \text{impossible refrigerator} \end{cases}$$

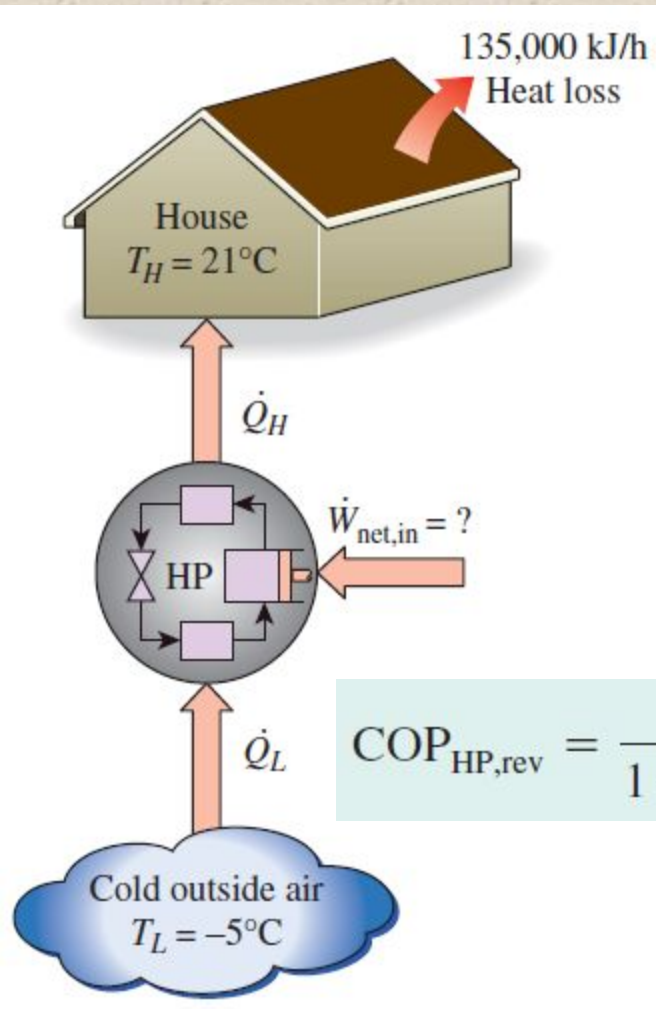
The COP of a reversible refrigerator or heat pump is the maximum theoretical value for the specified temperature limits.

Actual refrigerators or heat pumps may approach these values as their designs are improved, but they can never reach them.

The COPs of both the refrigerators and the heat pumps decrease as T_L decreases.

That is, it requires more work to absorb heat from lower-temperature media.

Heating a House by a Carnot Heat Pump



$$\text{COP}_{\text{HP,rev}} = \frac{1}{1 - T_L/T_H} = \frac{1}{1 - (-5 + 273 \text{ K})/(21 + 273 \text{ K})} = 11.3$$

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP}}} = \frac{37.5 \text{ kW}}{11.3} = \mathbf{3.32 \text{ kW}}$$

Summary

- Introduction to the second law
- Thermal energy reservoirs
- Heat engines
 - ✓ Thermal efficiency
 - ✓ The 2nd law: Kelvin-Planck statement
- Refrigerators and heat pumps
 - ✓ Coefficient of performance (COP)
 - ✓ The 2nd law: Clausius statement
- Perpetual motion machines
- Reversible and irreversible processes
 - ✓ Irreversibilities, Internally and externally reversible processes
- The Carnot cycle
 - ✓ The reversed Carnot cycle
- The Carnot principles
- The thermodynamic temperature scale
- The Carnot heat engine
 - ✓ The quality of energy
- The Carnot refrigerator and heat pump

Thermodynamics: An Engineering Approach

8th Edition

Yunus A. Çengel, Michael A. Boles
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CHAPTERS 1-6

Few Examples

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Control Mass Problem

EXAMPLE 4–9 Heating of a Gas by a Resistance Heater

A piston–cylinder device initially contains 0.5 m^3 of nitrogen gas at 400 kPa and 27°C . An electric heater within the device is turned on and is allowed to pass a current of 2 A for 5 min from a 120-V source. Nitrogen expands at constant pressure, and a heat loss of 2800 J occurs during the process. Determine the final temperature of nitrogen.

SOLUTION Nitrogen gas in a piston–cylinder device is heated by an electric resistance heater. Nitrogen expands at constant pressure while some heat is lost. The final temperature of nitrogen is to be determined.

Assumptions 1 Nitrogen is an ideal gas since it is at a high temperature and low pressure relative to its critical-point values of -147°C , and 3.39 MPa . 2 The system is stationary and thus the kinetic and potential energy changes are zero, $\Delta KE = \Delta PE = 0$ and $\Delta E = \Delta U$. 3 The pressure remains constant during the process and thus $P_2 = P_1$. 4 Nitrogen has constant specific heats at room temperature.

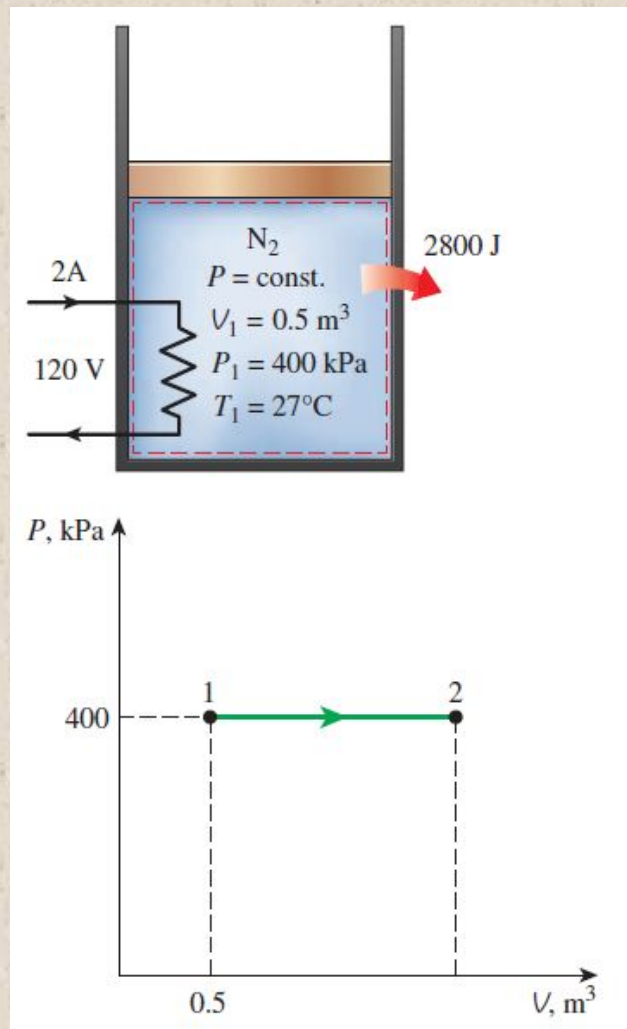
Analysis We take the contents of the cylinder as the *system* (Fig. 4–31). This is a *closed system* since no mass crosses the system boundary during the process. We observe that a piston–cylinder device typically involves a moving boundary and thus boundary work, W_b . Also, heat is lost from the system and electrical work W_e is done on the system.

First, let us determine the electrical work done on the nitrogen:

$$W_e = VI \Delta t = (120 \text{ V})(2 \text{ A})(5 \times 60 \text{ s}) \left(\frac{1 \text{ kJ/s}}{1000 \text{ VA}} \right) = 72 \text{ kJ}$$

The mass of nitrogen is determined from the ideal-gas relation:

$$m = \frac{P_1 V_1}{RT_1} = \frac{(400 \text{ kPa})(0.5 \text{ m}^3)}{(0.297 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300 \text{ K})} = 2.245 \text{ kg}$$



Under the stated assumptions and observations, the energy balance on the system can be expressed as

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

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

$$W_{e,\text{in}} - Q_{\text{out}} = \Delta H = m(h_2 - h_1) = mc_p(T_2 - T_1)$$

since $\Delta U + W_b = \Delta H$ for a closed system undergoing a quasi-equilibrium expansion or compression process at constant pressure. From Table A-2a, $c_p = 1.039 \text{ kJ/kg}\cdot\text{K}$ for nitrogen at room temperature. The only unknown quantity in the previous equation is T_2 , and it is found to be

$$72 \text{ kJ} - 2.8 \text{ kJ} = (2.245 \text{ kg})(1.039 \text{ kJ/kg}\cdot\text{K})(T_2 - 27^\circ\text{C})$$

$$T_2 = 56.7^\circ\text{C}$$

Discussion Note that we could also solve this problem by determining the boundary work and the internal energy change rather than the enthalpy change.

Control Volume Problem (Steady)

Problem 1

EXAMPLE 5–7 Power Generation by a Steam Turbine

The power output of an adiabatic steam turbine is 5 MW, and the inlet and the exit conditions of the steam are as indicated in Fig. 5–31.

- (a) Compare the magnitudes of Δh , Δke , and Δpe .
- (b) Determine the work done per unit mass of the steam flowing through the turbine.
- (c) Calculate the mass flow rate of the steam.

SOLUTION The inlet and exit conditions of a steam turbine and its power output are given. The changes in kinetic energy, potential energy, and enthalpy of steam, as well as the work done per unit mass and the mass flow rate of steam are to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time at any point and thus $\Delta m_{cv} = 0$ and $\Delta E_{cv} = 0$. 2 The system is adiabatic and thus there is no heat transfer.

Analysis We take the *turbine* 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 $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Also, work is done by the system. The inlet and exit velocities and elevations are given, and thus the kinetic and potential energies are to be considered.

(a) At the inlet, steam is in a superheated vapor state, and its enthalpy is

$$\left. \begin{array}{l} P_1 = 2 \text{ MPa} \\ T_1 = 400^\circ\text{C} \end{array} \right\} h_1 = 3248.4 \text{ kJ/kg} \quad (\text{Table A-6})$$

At the turbine exit, we obviously have a saturated liquid–vapor mixture at 15-kPa pressure. The enthalpy at this state is

$$h_2 = h_f + x_2 h_{fg} = [225.94 + (0.9)(2372.3)] \text{ kJ/kg} = 2361.01 \text{ kJ/kg}$$

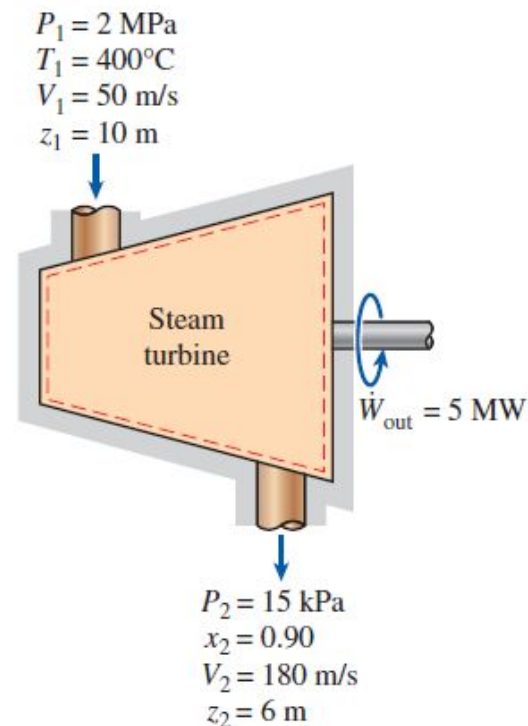


FIGURE 5–31
Schematic for Example 5–7.

Then

$$\Delta h = h_2 - h_1 = (2361.01 - 3248.4) \text{ kJ/kg} = -\mathbf{887.39 \text{ kJ/kg}}$$

$$\Delta \text{ke} = \frac{V_2^2 - V_1^2}{2} = \frac{(180 \text{ m/s})^2 - (50 \text{ m/s})^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = \mathbf{14.95 \text{ kJ/kg}}$$

$$\Delta \text{pe} = g(z_2 - z_1) = (9.81 \text{ m/s}^2)[(6 - 10) \text{ m}] \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = -\mathbf{0.04 \text{ kJ/kg}}$$

(b) The energy balance for this steady-flow system can be expressed in the rate form as

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

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

$$\dot{m} \left(h_1 + \frac{V_1^2}{2} + gz_1 \right) = \dot{W}_{\text{out}} + \dot{m} \left(h_2 + \frac{V_2^2}{2} + gz_2 \right) \quad (\text{since } \dot{Q} = 0)$$

Dividing by the mass flow rate \dot{m} and substituting, the work done by the turbine per unit mass of the steam is determined to be

$$\begin{aligned} w_{\text{out}} &= -\left[(h_2 - h_1) + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1) \right] = -(\Delta h + \Delta \text{ke} + \Delta \text{pe}) \\ &= -[-887.39 + 14.95 - 0.04] \text{ kJ/kg} = \mathbf{872.48 \text{ kJ/kg}} \end{aligned}$$

(c) The required mass flow rate for a 5-MW power output is

$$\dot{m} = \frac{\dot{W}_{\text{out}}}{w_{\text{out}}} = \frac{5000 \text{ kJ/s}}{872.48 \text{ kJ/kg}} = \mathbf{5.73 \text{ kg/s}}$$

Discussion Two observations can be made from these results. First, the change in potential energy is insignificant in comparison to the changes in enthalpy and kinetic energy. This is typical for most engineering devices. Second, as a result of low pressure and thus high specific volume, the steam velocity at the turbine exit can be very high. Yet the change in kinetic energy is a small fraction of the change in enthalpy (less than 2 percent in our case) and is therefore often neglected.

Control Volume Problem (Steady)

Problem 2

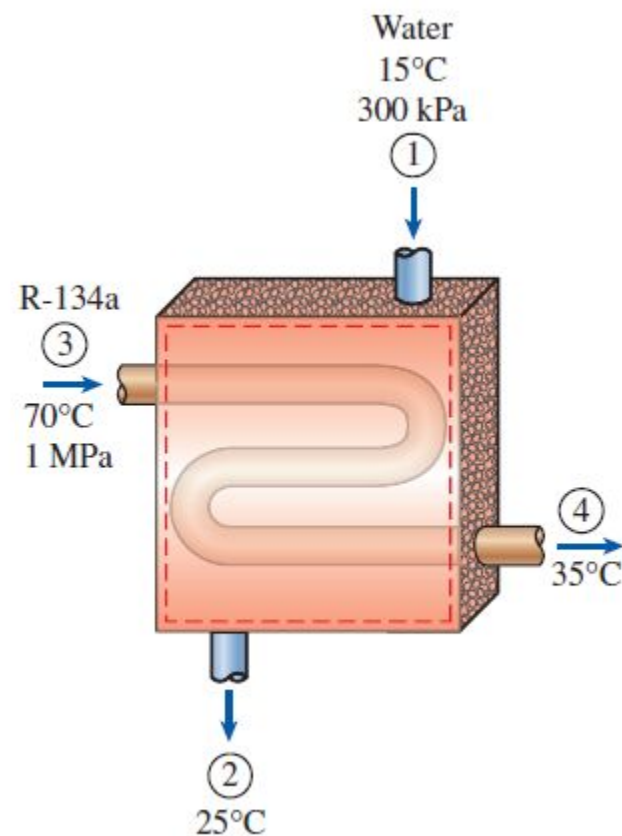
EXAMPLE 5-10 Cooling of Refrigerant-134a by Water

Refrigerant-134a is to be cooled by water in a condenser. The refrigerant enters the condenser with a mass flow rate of 6 kg/min at 1 MPa and 70°C and leaves at 35°C. The cooling water enters at 300 kPa and 15°C and leaves at 25°C. Neglecting any pressure drops, determine (a) the mass flow rate of the cooling water required and (b) the heat transfer rate from the refrigerant to water.

SOLUTION Refrigerant-134a is cooled by water in a condenser. The mass flow rate of the cooling water and the rate of heat transfer from the refrigerant to the water are to be determined.

Assumptions 1 This is a steady-flow process since there is no change with time at any point and thus $\Delta m_{cv} = 0$ and $\Delta E_{cv} = 0$. 2 The kinetic and potential energies are negligible, $ke \equiv pe \equiv 0$. 3 Heat losses from the system are negligible and thus $\dot{Q} \equiv 0$. 4 There is no work interaction.

Analysis We take the *entire heat exchanger* as the system (Fig. 5-40). This is a *control volume* since mass crosses the system boundary during the process. In general, there are several possibilities for selecting the control volume for multiple-stream steady-flow devices, and the proper choice depends on the situation at hand. We observe that there are two fluid streams (and thus two inlets and two exits) but no mixing.



(a) Under the stated assumptions and observations, the mass and energy balances for this steady-flow system can be expressed in the rate form as follows:

Mass balance: $\dot{m}_{\text{in}} = \dot{m}_{\text{out}}$

for each fluid stream since there is no mixing. Thus,

$$\dot{m}_1 = \dot{m}_2 = \dot{m}_w$$

$$\dot{m}_3 = \dot{m}_4 = \dot{m}_R$$

Energy balance:

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

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

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

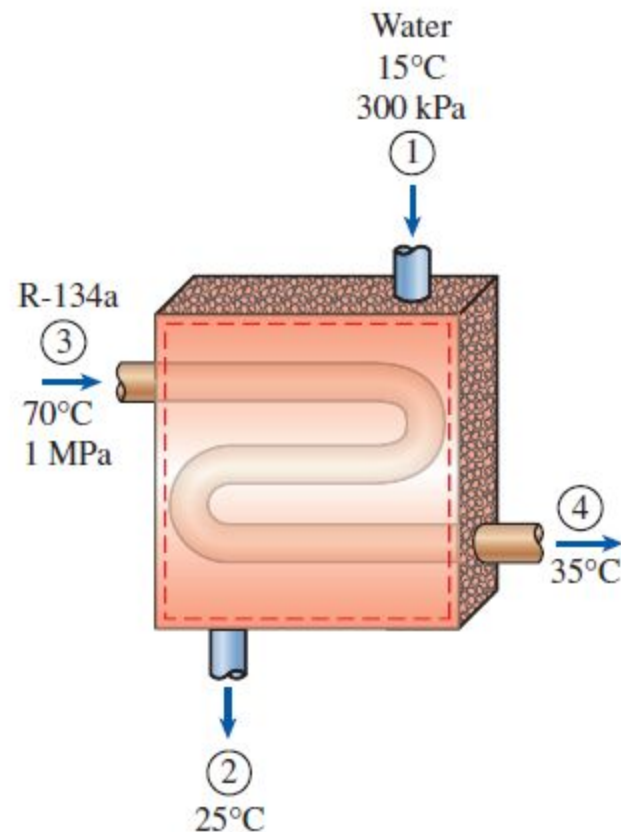
Combining the mass and energy balances and rearranging give

$$\dot{m}_w(h_1 - h_2) = \dot{m}_R(h_4 - h_3)$$

Now we need to determine the enthalpies at all four states. Water exists as a compressed liquid at both the inlet and the exit since the temperatures at both locations are below the saturation temperature of water at 300 kPa (133.52°C). Approximating the compressed liquid as a saturated liquid at the given temperatures, we have

$$h_1 \cong h_{f@15^\circ\text{C}} = 62.982 \text{ kJ/kg} \quad (\text{Table A-4})$$

$$h_2 \cong h_{f@25^\circ\text{C}} = 104.83 \text{ kJ/kg}$$



The refrigerant enters the condenser as a superheated vapor and leaves as a compressed liquid at 35°C. From refrigerant-134a tables,

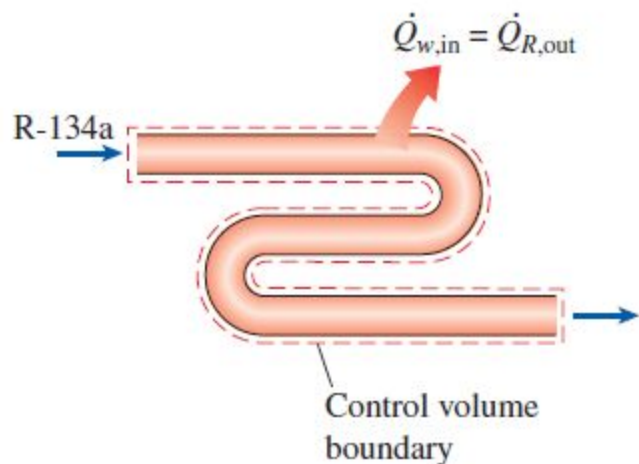
$$\left. \begin{array}{l} P_3 = 1 \text{ MPa} \\ T_3 = 70^\circ\text{C} \end{array} \right\} h_3 = 303.87 \text{ kJ/kg} \quad (\text{Table A-13})$$

$$\left. \begin{array}{l} P_4 = 1 \text{ MPa} \\ T_4 = 35^\circ\text{C} \end{array} \right\} h_4 \cong h_{f@35^\circ\text{C}} = 100.88 \text{ kJ/kg} \quad (\text{Table A-11})$$

Substituting, we find

$$\dot{m}_w(62.982 - 104.83) \text{ kJ/kg} = (6 \text{ kg/min})[(100.88 - 303.87) \text{ kJ/kg}]$$

$$\dot{m}_w = \mathbf{29.1 \text{ kg/min}}$$



(b) To determine the heat transfer from the refrigerant to the water, we have to choose a control volume whose boundary lies on the path of heat transfer. We can choose the volume occupied by either fluid as our control volume. For no particular reason, we choose the volume occupied by the water. All the assumptions stated earlier apply, except that the heat transfer is no longer zero. Then assuming heat to be transferred to water, the energy balance for this single-stream steady-flow system reduces to

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

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

$$\dot{Q}_{w, \text{in}} + \dot{m}_w h_1 = \dot{m}_w h_2$$

Rearranging and substituting,

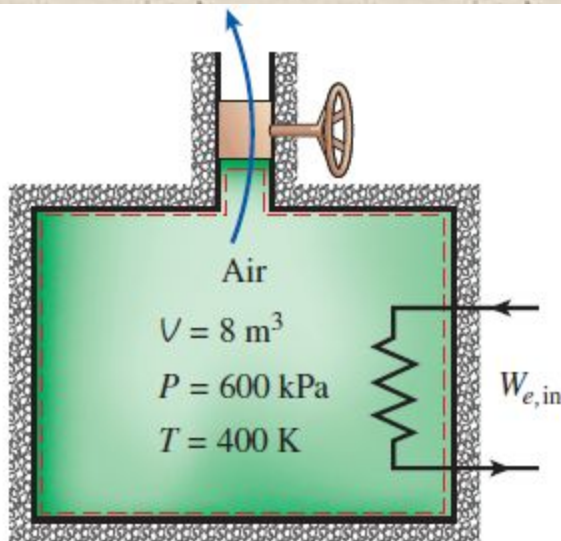
$$\begin{aligned} \dot{Q}_{w, \text{in}} &= \dot{m}_w (h_2 - h_1) = (29.1 \text{ kg/min})[(104.83 - 62.982) \text{ kJ/kg}] \\ &= \mathbf{1218 \text{ kJ/min}} \end{aligned}$$

Discussion Had we chosen the volume occupied by the refrigerant as the control volume (Fig. 5–41), we would have obtained the same result for $\dot{Q}_{R, \text{out}}$ since the heat gained by the water is equal to the heat lost by the refrigerant.

Control Volume Problem (Unsteady)

EXAMPLE 5–13**Discharge of Heated Air at Constant Temperature**

An insulated 8-m³ rigid tank contains air at 600 kPa and 400 K. A valve connected to the tank is now opened, and air is allowed to escape until the pressure inside drops to 200 kPa. The air temperature during the process is maintained constant by an electric resistance heater placed in the tank. Determine the electrical energy supplied to air during this process.



SOLUTION Pressurized air in an insulated rigid tank equipped with an electric heater is allowed to escape at constant temperature until the pressure inside drops to a specified value. The amount of electrical energy supplied to air 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 conditions remain 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.

Analysis We take the contents of the tank as the system, which is a control volume since mass crosses the boundary (Fig. 5–52). 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

$$\text{Mass balance:} \quad m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$$

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

$$W_{e,\text{in}} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong \text{ke} \cong \text{pe} \cong 0)$$

The gas constant of air is $R = 0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}$ (Table A-1). The initial and final masses of air in the tank and the discharged amount are determined from the ideal gas relation to be

$$m_1 = \frac{P_1 V_1}{RT_1} = \frac{(600 \text{ kPa})(8 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(400 \text{ K})} = 41.81 \text{ kg}$$

$$m_2 = \frac{P_2 V_2}{RT_2} = \frac{(200 \text{ kPa})(8 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(400 \text{ K})} = 13.94 \text{ kg}$$

$$m_e = m_1 - m_2 = 41.81 - 13.94 = 27.87 \text{ kg}$$

The enthalpy and internal energy of air at 400 K are $h_e = 400.98 \text{ kJ/kg}$ and $u_1 = u_2 = 286.16 \text{ kJ/kg}$ (Table A-17). The the electrical energy supplied to air is determined from the energy balance to be

$$\begin{aligned} W_{e,\text{in}} &= m_e h_e + m_2 u_2 - m_1 u_1 \\ &= (27.87 \text{ kg})(400.98 \text{ kJ/kg}) + (13.94 \text{ kg})(286.16 \text{ kJ/kg}) \\ &\quad - (41.81 \text{ kg})(286.16 \text{ kJ/kg}) \\ &= 3200 \text{ kJ} = \mathbf{0.889 \text{ kWh}} \end{aligned}$$

since $1 \text{ kWh} = 3600 \text{ kJ}$.

Discussion If the temperature of discharged air changes during the process, the problem can be solved with reasonable accuracy by evaluating h_e at the average discharge temperature $T_e = (T_2 + T_1)/2$, and treating it as constant.

Second Law Analysis (Problem 1)

EXAMPLE 6–3 Heat Rejection by a Refrigerator

The food compartment of a refrigerator, shown in Fig. 6–23, is maintained at 4°C by removing heat from it at a rate of 360 kJ/min. If the required power input to the refrigerator is 2 kW, determine (a) the coefficient of performance of the refrigerator and (b) the rate of heat rejection to the room that houses the refrigerator.

SOLUTION The power consumption of a refrigerator is given. The COP and the rate of heat rejection are to be determined.

Assumptions Steady operating conditions exist.

Analysis (a) The coefficient of performance of the refrigerator is

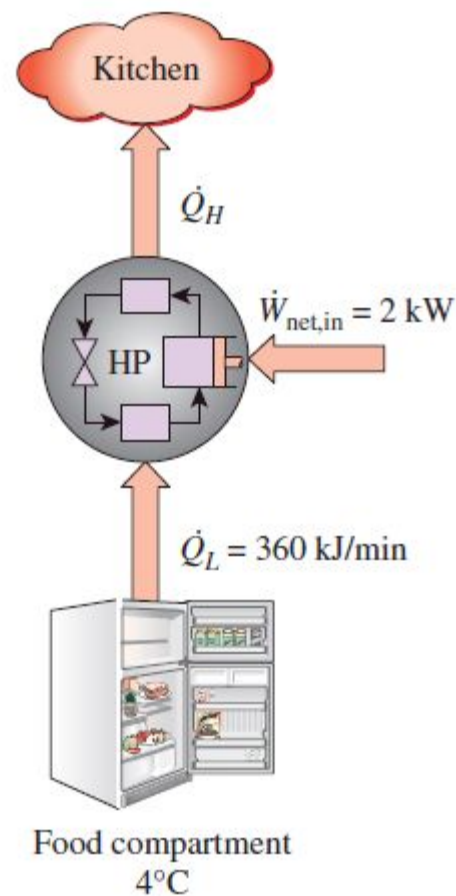
$$\text{COP}_R = \frac{\dot{Q}_L}{\dot{W}_{\text{net,in}}} = \frac{360 \text{ kJ/min}}{2 \text{ kW}} \left(\frac{1 \text{ kW}}{60 \text{ kJ/min}} \right) = 3$$

That is, 3 kJ of heat is removed from the refrigerated space for each kJ of work supplied.

(b) The rate at which heat is rejected to the room that houses the refrigerator is determined from the conservation of energy relation for cyclic devices,

$$\dot{Q}_H = \dot{Q}_L + \dot{W}_{\text{net,in}} = 360 \text{ kJ/min} + (2 \text{ kW}) \left(\frac{60 \text{ kJ/min}}{1 \text{ kW}} \right) = 480 \text{ kJ/min}$$

Discussion Notice that both the energy removed from the refrigerated space as heat and the energy supplied to the refrigerator as electrical work eventually show up in the room air and become part of the internal energy of the air. This demonstrates that energy can change from one form to another, can move from one place to another, but is never destroyed during a process.



Second Law Analysis (Problem 2)

EXAMPLE 6–6**A Carnot Refrigeration Cycle Operating in the Saturation Dome**

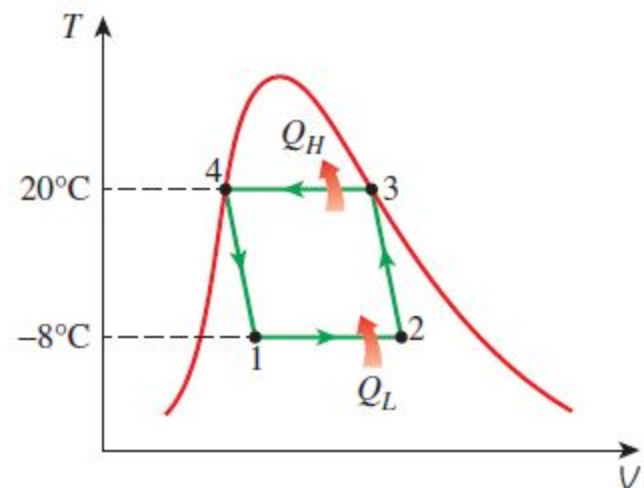
A Carnot refrigeration cycle is executed in a closed system in the saturated liquid–vapor mixture region using 0.8 kg of refrigerant-134a as the working fluid (Fig. 6–51). The maximum and the minimum temperatures in the cycle are 20 and -8°C , respectively. It is known that the refrigerant is saturated liquid at the end of the heat rejection process, and the net work input to the cycle is 15 kJ. Determine the fraction of the mass of the refrigerant that vaporizes during the heat addition process, and the pressure at the end of the heat rejection process.

SOLUTION A Carnot refrigeration cycle is executed in a closed system. The mass fraction of the refrigerant that vaporizes during the heat addition process and the pressure at the end of the heat rejection process are to be determined.

Assumptions The refrigerator operates on the ideal Carnot cycle.

Analysis Knowing the high and low temperatures, the coefficient of performance of the cycle is

$$\text{COP}_R = \frac{1}{T_H/T_L - 1} = \frac{1}{(20 + 273 \text{ K})/(-8 + 273 \text{ K}) - 1} = 9.464$$



The amount of cooling is determined from the definition of the coefficient of performance to be

$$Q_L = \text{COP}_R \times W_{\text{in}} = (9.464)(15 \text{ kJ}) = 142 \text{ kJ}$$

The enthalpy of vaporization R-134a at -8°C is $h_{fg} = 204.59 \text{ kJ/kg}$ (Table A-11). Then the amount of refrigerant that vaporizes during heat absorption becomes

$$Q_L = m_{\text{evap}} h_{fg@-8^\circ\text{C}} \rightarrow m_{\text{evap}} = \frac{142 \text{ kJ}}{204.59 \text{ kJ/kg}} = 0.694 \text{ kg}$$

Therefore, the fraction of mass that vaporized during heat addition process to the refrigerant is

$$\text{Mass fraction} = \frac{m_{\text{evap}}}{m_{\text{total}}} = \frac{0.694 \text{ kg}}{0.8 \text{ kg}} = \mathbf{0.868} \text{ or } \mathbf{86.8\%}$$

The pressure at the end of heat rejection process is simply the saturation pressure at heat rejection temperature,

$$P_4 = P_{\text{sat}@20^\circ\text{C}} = \mathbf{572.1 \text{ kPa}}$$

Discussion Carnot cycle is an idealized refrigeration cycle, thus it cannot be achieved in practice. Practical refrigeration cycles are analyzed in Chap. 11.