Chapter 8

EXERGY: A MEASURE OF WORK POTENTIAL

The increased awareness that the world's energy resources are limited has caused many countries to reexamine their energy policies and take drastic measures in eliminating waste. It has also sparked interest in the scientific community to take a closer look at the energy conversion devices and to develop new techniques to better utilize the existing limited resources. The first law of thermodynamics deals with the *quantity* of energy and asserts that energy cannot be created or destroyed. This law merely serves as a necessary tool for the bookkeeping of energy during a process and offers no challenges to the engineer. The second law, however, deals with the *quality* of energy. More specifically, it is concerned with the degradation of energy during a process, the entropy generation, and the lost opportunities to do work; and it offers plenty of room for improvement.

The second law of thermodynamics has proved to be a very powerful tool in the optimization of complex thermodynamic systems. In this chapter, we examine the performance of engineering devices in light of the second law of thermodynamics. We start our discussions with the introduction of *exergy* (also called *availability*), which is the maximum useful work that could be obtained from the system at a given state in a specified environment, and we continue with the *reversible work*, which is the maximum useful work that can be obtained as a system undergoes a process between two specified states. Next we discuss the *irreversibility* (also called the *exergy destruction* or *lost work*), which is the wasted work potential during a process as a result of irreversibilities, and we define a *second-law efficiency*. We then develop the *exergy balance* relation and apply it to closed systems and control volumes.

Objectives

The objectives of Chapter 8 are to:

- Examine the performance of engineering devices in light of the second law of thermodynamics.
- Define *exergy*, which is the maximum useful work that could be obtained from the system at a given state in a specified environment.
- Define *reversible work*, which is the maximum useful work that can be obtained as a system undergoes a process between two specified states.
- Define the exergy destruction, which is the wasted work potential during a process as a result of irreversibilities.
- Define the *second-law efficiency*.
- Develop the exergy balance relation.
- Apply exergy balance to closed systems and control volumes.
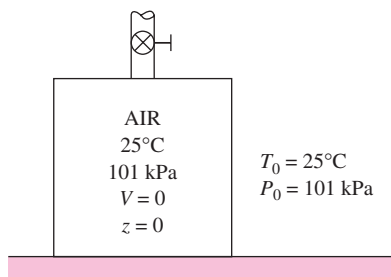
FIGURE 8–1

A system that is in equilibrium with its environment is said to be at the dead state.

FIGURE 8–2

At the dead state, the useful work potential (exergy) of a system is zero.

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8–1 ▪ EXERGY: WORK POTENTIAL OF ENERGY

When a new energy source, such as a geothermal well, is discovered, the first thing the explorers do is estimate the amount of energy contained in the source. This information alone, however, is of little value in deciding whether to build a power plant on that site. What we really need to know is the *work potential* of the source—that is, the amount of energy we can extract as useful work. The rest of the energy is eventually discarded as waste energy and is not worthy of our consideration. Thus, it would be very desirable to have a property to enable us to determine the useful work potential of a given amount of energy at some specified state. This property is *exergy,* which is also called the *availability* or *available energy*.

The work potential of the energy contained in a system at a specified state is simply the maximum useful work that can be obtained from the system. You will recall that the work done during a process depends on the initial state, the final state, and the process path. That is,

$$\text{Work} = f(\text{initial state, process path, final state})$$

In an exergy analysis, the *initial state* is specified, and thus it is not a variable. The work output is maximized when the process between two specified states is executed in a *reversible manner,* as shown in Chap. 7. Therefore, all the irreversibilities are disregarded in determining the work potential. Finally, the system must be in the *dead state* at the end of the process to maximize the work output.

A system is said to be in the **dead state** when it is in thermodynamic equilibrium with the environment it is in (Fig. 8–1). At the dead state, a system is at the temperature and pressure of its environment (in thermal and mechanical equilibrium); it has no kinetic or potential energy relative to the environment (zero velocity and zero elevation above a reference level); and it does not react with the environment (chemically inert). Also, there are no unbalanced magnetic, electrical, and surface tension effects between the system and its surroundings, if these are relevant to the situation at hand. The properties of a system at the dead state are denoted by subscript zero, for example, P_0, T_0, h_0, u_0, and s_0. Unless specified otherwise, the dead-state temperature and pressure are taken to be $T_0 = 25°C$ (77°F) and $P_0 = 1$ atm (101.325 kPa or 14.7 psia). A system has zero exergy at the dead state (Fig. 8–2).

Distinction should be made between the *surroundings, immediate surroundings,* and the *environment*. By definition, **surroundings** are everything outside the system boundaries. The **immediate surroundings** refer to the portion of the surroundings that is affected by the process, and **environment** refers to the region beyond the immediate surroundings whose properties are not affected by the process at any point. Therefore, any irreversibilities during a process occur within the system and its immediate surroundings, and the environment is free of any irreversibilities. When analyzing the cooling of a hot baked potato in a room at 25°C, for example, the warm air that surrounds the potato is the immediate surroundings, and the remaining part of the room air at 25°C is the environment. Note that the temperature of the immediate surroundings changes from the temperature of the potato at the boundary to the environment temperature of 25°C (Fig. 8–3).

The notion that a system must go to the dead state at the end of the process to maximize the work output can be explained as follows: If the system temperature at the final state is greater than (or less than) the temperature of the environment it is in, we can always produce additional work by running a heat engine between these two temperature levels. If the final pressure is greater than (or less than) the pressure of the environment, we can still obtain work by letting the system expand to the pressure of the environment. If the final velocity of the system is not zero, we can catch that extra kinetic energy by a turbine and convert it to rotating shaft work, and so on. No work can be produced from a system that is initially at the dead state. The atmosphere around us contains a tremendous amount of energy. However, the atmosphere is in the dead state, and the energy it contains has no work potential (Fig. 8–4).

Therefore, we conclude that a *system delivers the maximum possible work as it undergoes a reversible process from the specified initial state to the state of its environment, that is, the dead state*. This represents the *useful work potential* of the system at the specified state and is called **exergy.** It is important to realize that exergy does not represent the amount of work that a work-producing device will actually deliver upon installation. Rather, it represents the *upper limit on the amount of work a device can deliver without violating any thermodynamic laws*. There will always be a difference, large or small, between exergy and the actual work delivered by a device. This difference represents the room engineers have for improvement.

Note that the exergy of a system at a specified state depends on the conditions of the environment (the dead state) as well as the properties of the system. Therefore, exergy is a property of the *system–environment combination* and not of the system alone. Altering the environment is another way of increasing exergy, but it is definitely not an easy alternative.

The term *availability* was made popular in the United States by the M.I.T. School of Engineering in the 1940s. Today, an equivalent term, *exergy,* introduced in Europe in the 1950s, has found global acceptance partly because it is shorter, it rhymes with energy and entropy, and it can be adapted without requiring translation. In this text the preferred term is *exergy.*
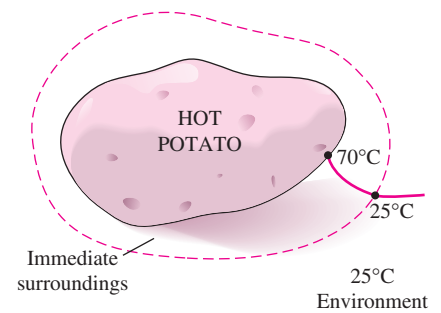

FIGURE 8–3
The immediate surroundings of a hot potato are simply the temperature gradient zone of the air next to the potato.

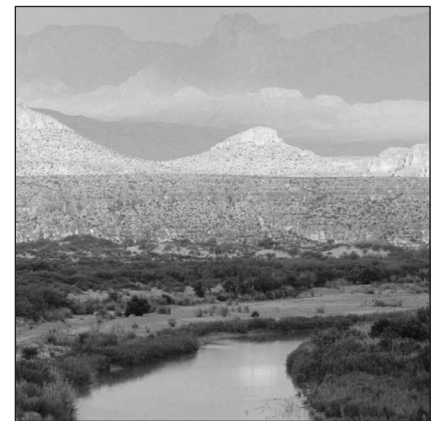**FIGURE 8–4**
The atmosphere contains a tremendous amount of energy, but no exergy.
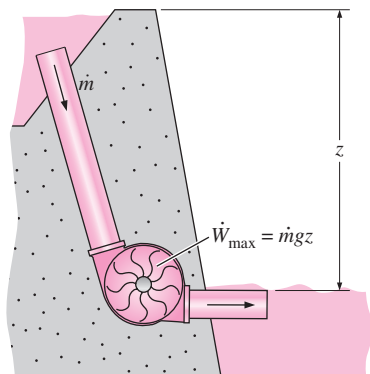
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Exergy (Work Potential) Associated with Kinetic and Potential Energy

Kinetic energy is a form of *mechanical energy*, and thus it can be converted to work entirely. Therefore, the *work potential* or *exergy* of the kinetic energy of a system is equal to the kinetic energy itself regardless of the temperature and pressure of the environment. That is,

Exergy of kinetic energy: $x_{ke} = ke = \dfrac{V^2}{2}$ (kJ/kg)   **(8–1)**

where V is the velocity of the system relative to the environment.

FIGURE 8–5

The *work potential* or *exergy* of potential energy is equal to the potential energy itself.
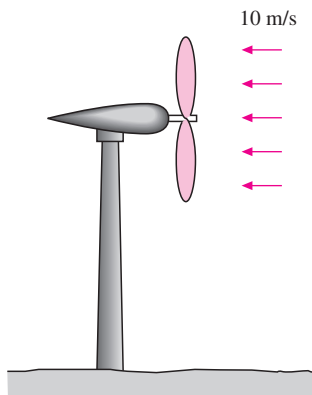
Potential energy is also a form of *mechanical energy*, and thus it can be converted to work entirely. Therefore, the *exergy* of the potential energy of a system is equal to the potential energy itself regardless of the temperature and pressure of the environment (Fig. 8–5). That is,

Exergy of potential energy: $\quad\quad x_{pe} = pe = gz \quad$ (kJ/kg) $\quad\quad$ **(8–2)**

where g is the gravitational acceleration and z is the elevation of the system relative to a reference level in the environment.

Therefore, the exergies of kinetic and potential energies are equal to themselves, and they are entirely available for work. However, the internal energy u and enthalpy h of a system are not entirely available for work, as shown later.

FIGURE 8–6

Schematic for Example 8–1.

EXAMPLE 8–1   **Maximum Power Generation by a Wind Turbine**

A wind turbine with a 12-m-diameter rotor, as shown in Fig. 8–6, is to be installed at a location where the wind is blowing steadily at an average velocity of 10 m/s. Determine the maximum power that can be generated by the wind turbine.

Solution   A wind turbine is being considered for a specified location. The maximum power that can be generated by the wind turbine is to be determined.

Assumptions   Air is at standard conditions of 1 atm and 25°C, and thus its density is 1.18 kg/m³.

Analysis   The air flowing with the wind has the same properties as the stagnant atmospheric air except that it possesses a velocity and thus some kinetic energy. This air will reach the dead state when it is brought to a complete stop. Therefore, the exergy of the blowing air is simply the kinetic energy it possesses:

$$ke = \frac{V^2}{2} = \frac{(10 \text{ m/s})^2}{2} \left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2} \right) = 0.05 \text{ kJ/kg}$$

That is, every unit mass of air flowing at a velocity of 10 m/s has a work potential of 0.05 kJ/kg. In other words, a perfect wind turbine will bring the air to a complete stop and capture that 0.05 kJ/kg of work potential. To determine the maximum power, we need to know the amount of air passing through the rotor of the wind turbine per unit time, that is, the mass flow rate, which is determined to be

$$\dot{m} = \rho AV = \rho \frac{\pi D^2}{4} V = (1.18 \text{ kg/m}^3) \frac{\pi (12 \text{ m})^2}{4} (10 \text{ m/s}) = 1335 \text{ kg/s}$$

Thus,

$$\text{Maximum power} = \dot{m}(ke) = (1335 \text{ kg/s})(0.05 \text{ kJ/kg}) = \textbf{66.8 kW}$$

This is the maximum power available to the wind turbine. Assuming a conversion efficiency of 30 percent, an actual wind turbine will convert 20.0 kW to electricity. Notice that the work potential for this case is equal to the entire kinetic energy of the air.

Discussion   It should be noted that although the entire kinetic energy of the wind is available for power production, Betz's law states that the power output of a wind machine is at maximum when the wind is slowed to one-third of its initial velocity. Therefore, for maximum power (and thus minimum cost per

installed power), the highest efficiency of a wind turbine is about 59 percent. In practice, the actual efficiency ranges between 20 and 40 percent and is about 35 percent for many wind turbines.

Wind power is suitable for harvesting when there are steady winds with an average velocity of at least 6 m/s (or 13 mph). Recent improvements in wind turbine design have brought the cost of generating wind power to about 5 cents per kWh, which is competitive with electricity generated from other resources.

EXAMPLE 8–2   Exergy Transfer from a Furnace

Consider a large furnace that can transfer heat at a temperature of 2000 R at a steady rate of 3000 Btu/s. Determine the rate of exergy flow associated with this heat transfer. Assume an environment temperature of 77°F.

Solution   Heat is being supplied by a large furnace at a specified temperature. The rate of exergy flow is to be determined.

Analysis   The furnace in this example can be modeled as a heat reservoir that supplies heat indefinitely at a constant temperature. The exergy of this heat energy is its useful work potential, that is, the maximum possible amount of work that can be extracted from it. This corresponds to the amount of work that a reversible heat engine operating between the furnace and the environment can produce.

The thermal efficiency of this reversible heat engine is

$$\eta_{th,max} = \eta_{th,rev} = 1 - \frac{T_L}{T_H} = 1 - \frac{T_0}{T_H} = 1 - \frac{537\,\text{R}}{2000\,\text{R}} = 0.732\ (\text{or } 73.2\%)$$

That is, a heat engine can convert, at best, 73.2 percent of the heat received from this furnace to work. Thus, the exergy of this furnace is equivalent to the power produced by the reversible heat engine:

$$\dot{W}_{max} = \dot{W}_{rev} = \eta_{th,rev}\,\dot{Q}_{in} = (0.732)(3000\,\text{Btu/s}) = \textbf{2196 Btu/s}$$

Discussion   Notice that 26.8 percent of the heat transferred from the furnace is not available for doing work. The portion of energy that cannot be converted to work is called **unavailable energy** (Fig. 8–7). Unavailable energy is simply the difference between the total energy of a system at a specified state and the exergy of that energy.
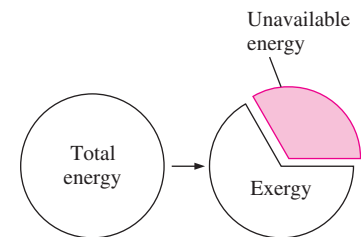

FIGURE 8–7
Unavailable energy is the portion of energy that cannot be converted to work by even a reversible heat engine.

8–2 ▪ REVERSIBLE WORK AND IRREVERSIBILITY

The property exergy serves as a valuable tool in determining the quality of energy and comparing the work potentials of different energy sources or systems. The evaluation of exergy alone, however, is not sufficient for studying engineering devices operating between two fixed states. This is because when evaluating exergy, the final state is always assumed to be the *dead state,* which is hardly ever the case for actual engineering systems. The isentropic efficiencies discussed in Chap. 7 are also of limited use because the exit state
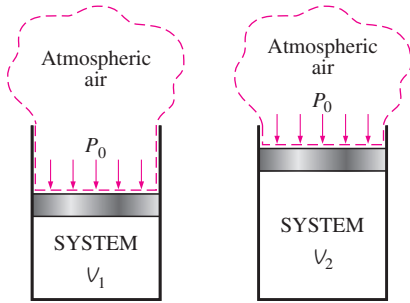
FIGURE 8–8

As a closed system expands, some work needs to be done to push the atmospheric air out of the way (W_{surr}).

FIGURE 8–9
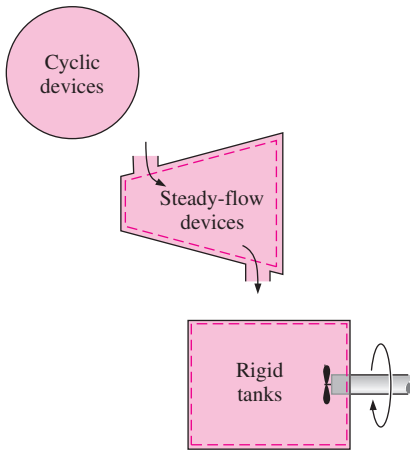
For constant-volume systems, the total actual and useful works are identical ($W_u = W$).
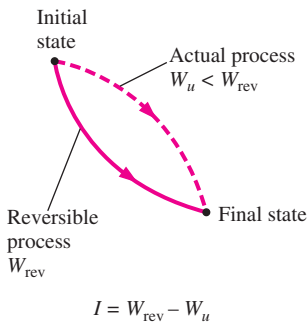

$$I = W_{rev} - W_u$$

FIGURE 8–10

The difference between reversible work and actual useful work is the irreversibility.

of the model (isentropic) process is not the same as the actual exit state and it is limited to adiabatic processes.

In this section, we describe two quantities that are related to the actual initial and final states of processes and serve as valuable tools in the thermodynamic analysis of components or systems. These two quantities are the *reversible work* and *irreversibility* (or *exergy destruction*). But first we examine the **surroundings work,** which is the work done by or against the surroundings during a process.

The work done by work-producing devices is not always entirely in a usable form. For example, when a gas in a piston–cylinder device expands, part of the work done by the gas is used to push the atmospheric air out of the way of the piston (Fig. 8–8). This work, which cannot be recovered and utilized for any useful purpose, is equal to the atmospheric pressure P_0 times the volume change of the system,

$$W_{surr} = P_0(V_2 - V_1) \tag{8–3}$$

The difference between the actual work W and the surroundings work W_{surr} is called the **useful work** W_u:

$$W_u = W - W_{surr} = W - P_0(V_2 - V_1) \tag{8–4}$$

When a system is expanding and doing work, part of the work done is used to overcome the atmospheric pressure, and thus W_{surr} represents a loss. When a system is compressed, however, the atmospheric pressure helps the compression process, and thus W_{surr} represents a gain.

Note that the work done by or against the atmospheric pressure has significance only for systems whose volume changes during the process (i.e., systems that involve moving boundary work). It has no significance for cyclic devices and systems whose boundaries remain fixed during a process such as rigid tanks and steady-flow devices (turbines, compressors, nozzles, heat exchangers, etc.), as shown in Fig. 8–9.

Reversible work W_{rev} is defined as *the maximum amount of useful work that can be produced (or the minimum work that needs to be supplied) as a system undergoes a process between the specified initial and final states.* This is the useful work output (or input) obtained (or expended) when the process between the initial and final states is executed in a totally reversible manner. When the final state is the dead state, the reversible work equals exergy. For processes that require work, reversible work represents the minimum amount of work necessary to carry out that process. For convenience in presentation, the term *work* is used to denote both work and power throughout this chapter.

Any difference between the reversible work W_{rev} and the useful work W_u is due to the irreversibilities present during the process, and this difference is called **irreversibility** *I*. It is expressed as (Fig. 8–10)

$$I = W_{rev,out} - W_{u,out} \quad \text{or} \quad I = W_{u,in} - W_{rev,in} \tag{8–5}$$

The irreversibility is equivalent to the *exergy destroyed,* discussed in Sec. 8–4. For a totally reversible process, the actual and reversible work terms are identical, and thus the irreversibility is zero. This is expected since totally reversible processes generate no entropy. Irreversibility is a *positive quantity* for all actual (irreversible) processes since $W_{rev} \geq W_u$ for work-producing devices and $W_{rev} \leq W_u$ for work-consuming devices.

Irreversibility can be viewed as the *wasted work potential* or the *lost opportunity* to do work. It represents the energy that could have been converted to work but was not. The smaller the irreversibility associated with a process, the greater the work that is produced (or the smaller the work that is consumed). The performance of a system can be improved by minimizing the irreversibility associated with it.

EXAMPLE 8–3   The Rate of Irreversibility of a Heat Engine

A heat engine receives heat from a source at 1200 K at a rate of 500 kJ/s and rejects the waste heat to a medium at 300 K (Fig. 8–11). The power output of the heat engine is 180 kW. Determine the reversible power and the irreversibility rate for this process.

Solution   The operation of a heat engine is considered. The reversible power and the irreversibility rate associated with this operation are to be determined.
Analysis   The reversible power for this process is the amount of power that a reversible heat engine, such as a Carnot heat engine, would produce when operating between the same temperature limits, and is determined to be:

$$\dot{W}_{rev} = \eta_{th,rev}\,\dot{Q}_{in} = \left(1 - \frac{T_{sink}}{T_{source}}\right)\dot{Q}_{in} = \left(1 - \frac{300\ \text{K}}{1200\ \text{K}}\right)(500\ \text{kW}) = \mathbf{375\ kW}$$

This is the maximum power that can be produced by a heat engine operating between the specified temperature limits and receiving heat at the specified rate. This would also represent the *available power* if 300 K were the lowest temperature available for heat rejection.
    The irreversibility rate is the difference between the reversible power (maximum power that could have been produced) and the useful power output:

$$\dot{I} = \dot{W}_{rev,out} - \dot{W}_{u,out} = 375 - 180 = \mathbf{195\ kW}$$

Discussion   Note that 195 kW of power potential is wasted during this process as a result of irreversibilities. Also, the $500 - 375 = 125$ kW of heat rejected to the sink is not available for converting to work and thus is not part of the irreversibility.
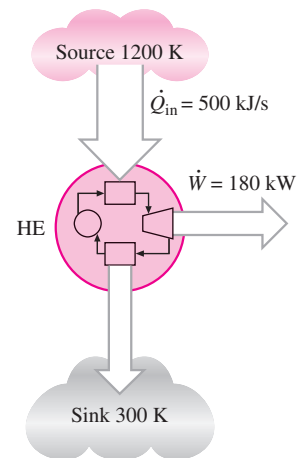

FIGURE 8–11
Schematic for Example 8–3.

EXAMPLE 8–4   Irreversibility during the Cooling of an Iron Block

A 500-kg iron block shown in Fig. 8–12 is initially at 200°C and is allowed to cool to 27°C by transferring heat to the surrounding air at 27°C. Determine the reversible work and the irreversibility for this process.

Solution   A hot iron block is allowed to cool in air. The reversible work and irreversibility associated with this process are to be determined.
Assumptions   **1** The kinetic and potential energies are negligible. **2** The process involves no work interactions.
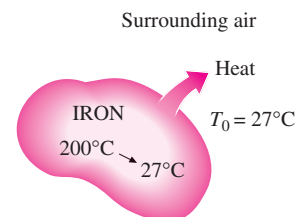

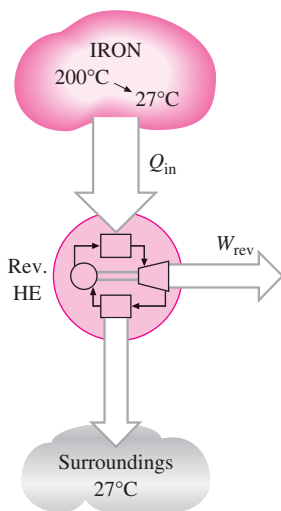**FIGURE 8–12**
Schematic for Example 8–4.

FIGURE 8–13

An irreversible heat transfer process can be made reversible by the use of a reversible heat engine.

Analysis We take the *iron block* as the system. This is a *closed system* since no mass crosses the system boundary. We note that heat is lost from the system.

It probably came as a surprise to you that we are asking to find the "reversible work" for a process that does not involve any work interactions. Well, even if no attempt is made to produce work during this process, the potential to do work still exists, and the reversible work is a quantitative measure of this potential.

The reversible work in this case is determined by considering a series of imaginary reversible heat engines operating between the source (at a variable temperature T) and the sink (at a constant temperature T_0), as shown in Fig. 8–13. Summing their work output:

$$\delta W_{rev} = \eta_{th,rev}\,\delta Q_{in} = \left(1 - \frac{T_{sink}}{T_{source}}\right)\delta Q_{in} = \left(1 - \frac{T_0}{T}\right)\delta Q_{in}$$

and

$$W_{rev} = \int\left(1 - \frac{T_0}{T}\right)\delta Q_{in}$$

The source temperature T changes from $T_1 = 200°C = 473$ K to $T_0 = 27°C = 300$ K during this process. A relation for the differential heat transfer from the iron block can be obtained from the differential form of the energy balance applied on the iron block,

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

$$-\delta Q_{out} = dU = mc_{avg}\,dT$$

Then,

$$\delta Q_{in,heat\,engine} = \delta Q_{out,system} = -mc_{avg}\,dT$$

since heat transfers from the iron and to the heat engine are equal in magnitude and opposite in direction. Substituting and performing the integration, the reversible work is determined to be

$$W_{rev} = \int_{T_1}^{T_0}\left(1 - \frac{T_0}{T}\right)(-mc_{avg}\,dT) = mc_{avg}(T_1 - T_0) - mc_{avg}T_0\ln\frac{T_1}{T_0}$$

$$= (500\text{ kg})(0.45\text{ kJ/kg}\cdot\text{K})\left[(473 - 300)\text{ K} - (300\text{ K})\ln\frac{473\text{ K}}{300\text{ K}}\right]$$

$$= \mathbf{8191\text{ kJ}}$$

where the specific heat value is obtained from Table A–3. The first term in the above equation [$Q = mc_{avg}(T_1 - T_0) = 38{,}925$ kJ] is the total heat transfer from the iron block to the heat engine. The reversible work for this problem is found to be 8191 kJ, which means that 8191 (21 percent) of the 38,925 kJ of heat transferred from the iron block to the ambient air *could* have been converted to work. If the specified ambient temperature of 27°C is the lowest available environment temperature, the reversible work determined above also represents the exergy, which is the maximum work potential of the sensible energy contained in the iron block.

The irreversibility for this process is determined from its definition,

$$I = W_{rev} - W_u = 8191 - 0 = \textbf{8191 kJ}$$

Discussion Notice that the reversible work and irreversibility (the wasted work potential) are the same for this case since the entire work potential is wasted. The source of irreversibility in this process is the heat transfer through a finite temperature difference.

EXAMPLE 8–5   **Heating Potential of a Hot Iron Block**

The iron block discussed in Example 8–4 is to be used to maintain a house at 27°C when the outdoor temperature is 5°C. Determine the maximum amount of heat that can be supplied to the house as the iron cools to 27°C.

Solution The iron block is now reconsidered for heating a house. The maximum amount of heating this block can provide is to be determined.

Analysis Probably the first thought that comes to mind to make the most use of the energy stored in the iron block is to take it inside and let it cool in the house, as shown in Fig. 8–14, transferring its sensible energy as heat to the indoors air (provided that it meets the approval of the household, of course). The iron block can keep "losing" heat until its temperature drops to the indoor temperature of 27°C, transferring a total of 38,925 kJ of heat. Since we utilized the entire energy of the iron block available for heating without wasting a single kilojoule, it seems like we have a 100-percent-efficient operation, and nothing can beat this, right? Well, not quite.

In Example 8–4 we determined that this process has an irreversibility of 8191 kJ, which implies that things are not as "perfect" as they seem. A "perfect" process is one that involves "zero" irreversibility. The irreversibility in this process is associated with the heat transfer through a finite temperature difference that can be eliminated by running a reversible heat engine between the iron block and the indoor air. This heat engine produces (as determined in Example 8–4) 8191 kJ of work and reject the remaining 38,925 − 8191 = 30,734 kJ of heat to the house. Now we managed to eliminate the irreversibility and ended up with 8191 kJ of work. What can we do with this work? Well, at worst we can convert it to heat by running a paddle wheel, for example, creating an equal amount of irreversibility. Or we can supply this work to a heat pump that transports heat from the outdoors at 5°C to the indoors at 27°C. Such a heat pump, if reversible, has a coefficient of performance of

$$COP_{HP} = \frac{1}{1 - T_L/T_H} = \frac{1}{1 - (278\ \text{K})/(300\ \text{K})} = 13.6$$

That is, this heat pump can supply the house with 13.6 times the energy it consumes as work. In our case, it will consume the 8191 kJ of work and deliver 8191 × 13.6 = 111,398 kJ of heat to the house. Therefore, the hot iron block has the potential to supply

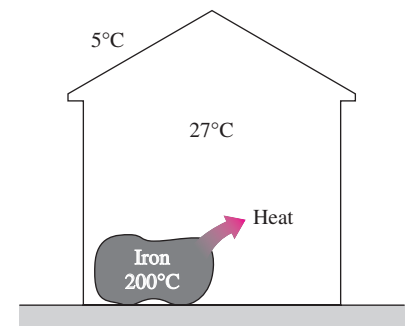$$(30,734 + 111,398)\ \text{kJ} = 142,132\ \text{kJ} \cong \textbf{142 MJ}$$

FIGURE 8–14
Schematic for Example 8–5.

of heat to the house. The irreversibility for this process is zero, and this is *the best* we can do under the specified conditions. A similar argument can be given for the electric heating of residential or commercial buildings.

Discussion Now try to answer the following question: What would happen if the heat engine were operated between the iron block and the outside air instead of the house until the temperature of the iron block fell to 27°C? Would the amount of heat supplied to the house still be 142 MJ? Here is a hint: The initial and final states in both cases are the same, and the irreversibility for both cases is zero.

8–3 · SECOND-LAW EFFICIENCY, η_{II}

In Chap. 6 we defined the *thermal efficiency* and the *coefficient of performance* for devices as a measure of their performance. They are defined on the basis of the first law only, and they are sometimes referred to as the *first-law efficiencies*. The first law efficiency, however, makes no reference to the best possible performance, and thus it may be misleading.

Consider two heat engines, both having a thermal efficiency of 30 percent, as shown in Fig. 8–15. One of the engines (engine A) is supplied with heat from a source at 600 K, and the other one (engine B) from a source at 1000 K. Both engines reject heat to a medium at 300 K. At first glance, both engines seem to convert to work the same fraction of heat that they receive; thus they are performing equally well. When we take a second look at these engines in light of the second law of thermodynamics, however, we see a totally different picture. These engines, at best, can perform as reversible engines, in which case their efficiencies would be

$$\eta_{rev,A} = \left(1 - \frac{T_L}{T_H}\right)_A = 1 - \frac{300\text{ K}}{600\text{ K}} = 50\%$$

$$\eta_{rev,B} = \left(1 - \frac{T_L}{T_H}\right)_B = 1 - \frac{300\text{ K}}{1000\text{ K}} = 70\%$$

Now it is becoming apparent that engine B has a greater work potential available to it (70 percent of the heat supplied as compared to 50 percent for engine A), and thus should do a lot better than engine A. Therefore, we can say that engine B is performing poorly relative to engine A even though both have the same thermal efficiency.

It is obvious from this example that the first-law efficiency alone is not a realistic measure of performance of engineering devices. To overcome this deficiency, we define a **second-law efficiency** η_{II} as the ratio of the actual thermal efficiency to the maximum possible (reversible) thermal efficiency under the same conditions (Fig. 8–16):

$$\eta_{II} = \frac{\eta_{th}}{\eta_{th,rev}} \qquad \text{(heat engines)} \qquad \textbf{(8–6)}$$

Based on this definition, the second-law efficiencies of the two heat engines discussed above are

$$\eta_{II,A} = \frac{0.30}{0.50} = 0.60 \quad \text{and} \quad \eta_{II,B} = \frac{0.30}{0.70} = 0.43$$
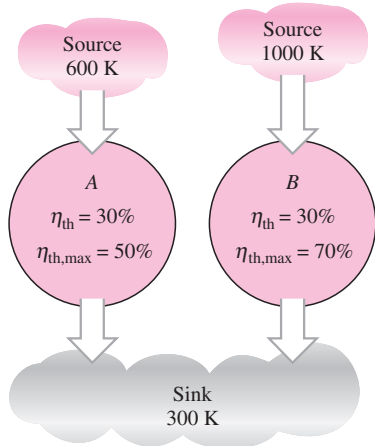

FIGURE 8–15

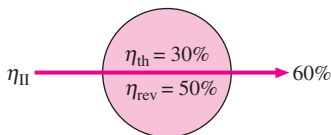Two heat engines that have the same thermal efficiency, but different maximum thermal efficiencies.

FIGURE 8–16

Second-law efficiency is a measure of the performance of a device relative to its performance under reversible conditions.

That is, engine *A* is converting 60 percent of the available work potential to useful work. This ratio is only 43 percent for engine *B*.

The second-law efficiency can also be expressed as the ratio of the useful work output and the maximum possible (reversible) work output:

$$\eta_{II} = \frac{W_u}{W_{rev}} \qquad \text{(work-producing devices)} \qquad \text{(8–7)}$$

This definition is more general since it can be applied to processes (in turbines, piston–cylinder devices, etc.) as well as to cycles. Note that the second-law efficiency cannot exceed 100 percent (Fig. 8–17).

We can also define a second-law efficiency for work-consuming noncyclic (such as compressors) and cyclic (such as refrigerators) devices as the ratio of the minimum (reversible) work input to the useful work input:

$$\eta_{II} = \frac{W_{rev}}{W_u} \qquad \text{(work-consuming devices)} \qquad \text{(8–8)}$$

For cyclic devices such as refrigerators and heat pumps, it can also be expressed in terms of the coefficients of performance as

$$\eta_{II} = \frac{COP}{COP_{rev}} \qquad \text{(refrigerators and heat pumps)} \qquad \text{(8–9)}$$

Again, because of the way we defined the second-law efficiency, its value cannot exceed 100 percent. In the above relations, the reversible work W_{rev} should be determined by using the same initial and final states as in the actual process.

The definitions above for the second-law efficiency do not apply to devices that are not intended to produce or consume work. Therefore, we need a more general definition. However, there is some disagreement on a general definition of the second-law efficiency, and thus a person may encounter different definitions for the same device. The second-law efficiency is intended to serve as a measure of approximation to reversible operation, and thus its value should range from zero in the worst case (complete destruction of exergy) to one in the best case (no destruction of exergy). With this in mind, we define the second-law efficiency of a system during a process as (Fig. 8–18)

$$\eta_{II} = \frac{\text{Exergy recovered}}{\text{Exergy supplied}} = 1 - \frac{\text{Exergy destroyed}}{\text{Exergy supplied}} \qquad \text{(8–10)}$$

Therefore, when determining the second-law efficiency, the first thing we need to do is determine how much exergy or work potential is consumed during a process. In a reversible operation, we should be able to recover entirely the exergy supplied during the process, and the irreversibility in this case should be zero. The second-law efficiency is zero when we recover none of the exergy supplied to the system. Note that the exergy can be supplied or recovered at various amounts in various forms such as heat, work, kinetic energy, potential energy, internal energy, and enthalpy. Sometimes there are differing (though valid) opinions on what constitutes supplied exergy, and this causes differing definitions for second-law efficiency. At all times, however, the exergy recovered and the exergy destroyed (the irreversibility) must add up to the exergy supplied. Also, we need to define the system precisely in order to identify correctly any interactions between the system and its surroundings.
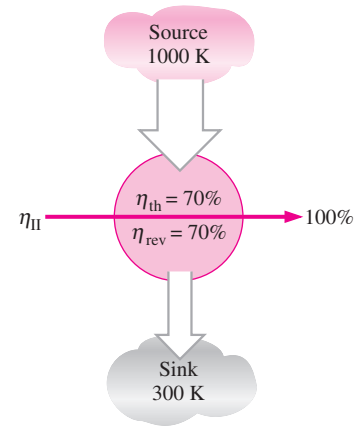

FIGURE 8–17

Second-law efficiency of all reversible devices is 100 percent.
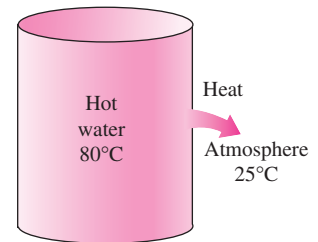

FIGURE 8–18

The second-law efficiency of naturally occurring processes is zero if none of the work potential is recovered.

For a *heat engine,* the exergy supplied is the decrease in the exergy of the heat transferred to the engine, which is the difference between the exergy of the heat supplied and the exergy of the heat rejected. (The exergy of the heat rejected at the temperature of the surroundings is zero.) The net work output is the recovered exergy.

For a *refrigerator* or *heat pump,* the exergy supplied is the work input since the work supplied to a cyclic device is entirely available. The recovered exergy is the exergy of the heat transferred to the high-temperature medium (which is the reversible work) for a heat pump, and the exergy of the heat transferred from the low-temperature medium for a refrigerator.

For a heat exchanger with two unmixed fluid streams, normally the exergy supplied is the decrease in the exergy of the higher-temperature fluid stream, and the exergy recovered is the increase in the exergy of the lower-temperature fluid stream. This is discussed further in Sec. 8–8.
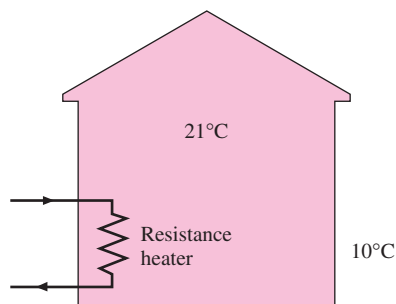

FIGURE 8–19

Schematic for Example 8–6.

EXAMPLE 8–6   **Second-Law Efficiency of Resistance Heaters**

A dealer advertises that he has just received a shipment of electric resistance heaters for residential buildings that have an efficiency of 100 percent (Fig. 8–19). Assuming an indoor temperature of 21°C and outdoor temperature of 10°C, determine the second-law efficiency of these heaters.

Solution   Electric resistance heaters are being considered for residential buildings. The second-law efficiency of these heaters is to be determined.
Analysis   Obviously the efficiency that the dealer is referring to is the first-law efficiency, meaning that for each unit of electric energy (work) consumed, the heater will supply the house with 1 unit of energy (heat). That is, the advertised heater has a COP of 1.

At the specified conditions, a reversible heat pump would have a coefficient of the performance of

$$\text{COP}_{\text{HP,rev}} = \frac{1}{1 - T_L/T_H} = \frac{1}{1 - (283 \text{ K})/(294 \text{ K})} = 26.7$$

That is, it would supply the house with 26.7 units of heat (extracted from the cold outside air) for each unit of electric energy it consumes.

The second-law efficiency of this resistance heater is

$$\eta_{\text{II}} = \frac{\text{COP}}{\text{COP}_{\text{rev}}} = \frac{1.0}{26.7} = \textbf{0.037 or 3.7\%}$$

which does not look so impressive. The dealer will not be happy to see this value. Considering the high price of electricity, a consumer will probably be better off with a "less" efficient gas heater.

INTERACTIVE TUTORIAL

SEE TUTORIAL CH. 8, SEC. 4 ON THE DVD.

8–4 ▪ EXERGY CHANGE OF A SYSTEM

The property *exergy* is the work potential of a system in a specified environment and represents the maximum amount of useful work that can be obtained as the system is brought to equilibrium with the environment.

Unlike energy, the value of exergy depends on the state of the environment as well as the state of the system. Therefore, exergy is a combination property. The exergy of a system that is in equilibrium with its environment is zero. The state of the environment is referred to as the "dead state" since the system is practically "dead" (cannot do any work) from a thermodynamic point of view when it reaches that state.

In this section we limit the discussion to **thermo-mechanical exergy,** and thus disregard any mixing and chemical reactions. Therefore, a system at this "restricted dead state" is at the temperature and pressure of the environment and it has no kinetic or potential energies relative to the environment. However, it may have a different chemical composition than the environment. Exergy associated with different chemical compositions and chemical reactions is discussed in later chapters.

Below we develop relations for the exergies and exergy changes for a fixed mass and a flow stream.

Exergy of a Fixed Mass:
Nonflow (or Closed System) Exergy

In general, internal energy consists of *sensible, latent, chemical,* and *nuclear* energies. However, in the absence of any chemical or nuclear reactions, the chemical and nuclear energies can be disregarded and the internal energy can be considered to consist of only sensible and latent energies that can be transferred to or from a system as *heat* whenever there is a temperature difference across the system boundary. The second law of thermodynamics states that heat cannot be converted to work entirely, and thus the work potential of internal energy must be less than the internal energy itself. But how much less?

To answer that question, we need to consider a stationary closed system at a specified state that undergoes a *reversible* process to the state of the environment (that is, the final temperature and pressure of the system should be T_0 and P_0, respectively). The useful work delivered during this process is the exergy of the system at its initial state (Fig. 8–20).

Consider a piston–cylinder device that contains a fluid of mass m at temperature T and pressure P. The system (the mass inside the cylinder) has a volume V, internal energy U, and entropy S. The system is now allowed to undergo a differential change of state during which the volume changes by a differential amount dV and heat is transferred in the differential amount of δQ. Taking the direction of heat and work transfers to be *from* the system (heat and work outputs), the energy balance for the system during this differential process can be expressed as

$$\underbrace{\delta E_{in} - \delta E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{dE_{system}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc., energies}}} \quad \textbf{(8–11)}$$

$$-\delta Q - \delta W = dU$$

since the only form of energy the system contains is *internal energy,* and the only forms of energy transfer a fixed mass can involve are heat and work. Also, the only form of work a simple compressible system can involve during a reversible process is the boundary work, which is given to be $\delta W = P\,dV$
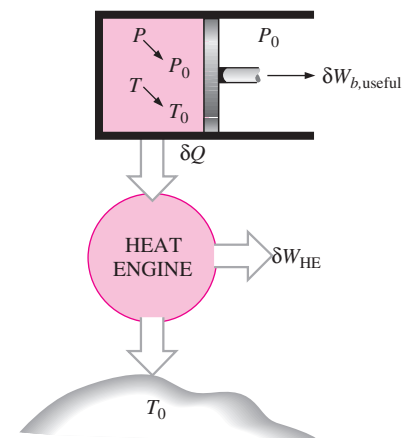

FIGURE 8–20

The *exergy* of a specified mass at a specified state is the useful work that can be produced as the mass undergoes a reversible process to the state of the environment.

when the direction of work is taken to be from the system (otherwise it would be $-P\,dV$). The pressure P in the $P\,dV$ expression is the absolute pressure, which is measured from absolute zero. Any useful work delivered by a piston–cylinder device is due to the pressure above the atmospheric level. Therefore,

$$\delta W = P\,dV = (P - P_0)\,dV + P_0\,dV = \delta W_{b,\text{useful}} + P_0\,dV \qquad \textbf{(8–12)}$$

A reversible process cannot involve any heat transfer through a finite temperature difference, and thus any heat transfer between the system at temperature T and its surroundings at T_0 must occur through a reversible heat engine. Noting that $dS = \delta Q/T$ for a reversible process, and the thermal efficiency of a reversible heat engine operating between the temperatures of T and T_0 is $\eta_{\text{th}} = 1 - T_0/T$, the differential work produced by the engine as a result of this heat transfer is

$$\delta W_{\text{HE}} = \left(1 - \frac{T_0}{T}\right)\delta Q = \delta Q - \frac{T_0}{T}\,\delta Q = \delta Q - (-T_0\,dS) \rightarrow$$

$$\delta Q = \delta W_{\text{HE}} - T_0\,dS \qquad \textbf{(8–13)}$$

Substituting the δW and δQ expressions in Eqs. 8–12 and 8–13 into the energy balance relation (Eq. 8–11) gives, after rearranging,

$$\delta W_{\text{total useful}} = \delta W_{\text{HE}} + \delta W_{b,\text{useful}} = -dU - P_0\,dV + T_0\,dS$$

Integrating from the given state (no subscript) to the dead state (0 subscript) we obtain

$$W_{\text{total useful}} = (U - U_0) + P_0(V - V_0) - T_0(S - S_0) \qquad \textbf{(8–14)}$$

where $W_{\text{total useful}}$ is the total useful work delivered as the system undergoes a reversible process from the given state to the dead state, which is *exergy* by definition.

A closed system, in general, may possess kinetic and potential energies, and the total energy of a closed system is equal to the sum of its internal, kinetic, and potential energies. Noting that kinetic and potential energies themselves are forms of exergy, the exergy of a closed system of mass m is

$$X = (U - U_0) + P_0(V - V_0) - T_0(S - S_0) + m\frac{V^2}{2} + mgz \qquad \textbf{(8–15)}$$

On a unit mass basis, the **closed system** (or **nonflow**) **exergy** ϕ is expressed as

$$\phi = (u - u_0) + P_0(v - v_0) - T_0(s - s_0) + \frac{V^2}{2} + gz \qquad \textbf{(8–16)}$$

$$= (e - e_0) + P_0(v - v_0) - T_0(s - s_0)$$

where u_0, v_0, and s_0 are the properties of the *system* evaluated at the dead state. Note that the exergy of a system is zero at the dead state since $e = e_0$, $v = v_0$, and $s = s_0$ at that state.

The exergy change of a closed system during a process is simply the difference between the final and initial exergies of the system,

$$\Delta X = X_2 - X_1 = m(\phi_2 - \phi_1) = (E_2 - E_1) + P_0(V_2 - V_1) - T_0(S_2 - S_1) \qquad \textbf{(8–17)}$$

$$= (U_2 - U_1) + P_0(V_2 - V_1) - T_0(S_2 - S_1) + m\frac{V_2^2 - V_1^2}{2} + mg(z_2 - z_1)$$

or, on a unit mass basis,

$$\Delta\phi = \phi_2 - \phi_1 = (u_2 - u_1) + P_0(v_2 - v_1) - T_0(s_2 - s_1) + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1)$$

$$= (e_2 - e_1) + P_0(v_2 - v_1) - T_0(s_2 - s_1) \tag{8-18}$$

For *stationary* closed systems, the kinetic and potential energy terms drop out.

When the properties of a system are not uniform, the exergy of the system can be determined by integration from

$$X_{\text{system}} = \int \phi \, \delta m = \int_V \phi \rho \, dV \tag{8-19}$$

where V is the volume of the system and ρ is density.

Note that exergy is a property, and the value of a property does not change unless the *state* changes. Therefore, the *exergy change* of a system is zero if the state of the system or the environment does not change during the process. For example, the exergy change of steady flow devices such as nozzles, compressors, turbines, pumps, and heat exchangers in a given environment is zero during steady operation.

The exergy of a closed system is either *positive* or *zero*. It is never negative. Even a medium at *low temperature* ($T < T_0$) and/or *low pressure* ($P < P_0$) contains exergy since a cold medium can serve as the heat sink to a heat engine that absorbs heat from the environment at T_0, and an evacuated space makes it possible for the atmospheric pressure to move a piston and do useful work (Fig. 8–21).
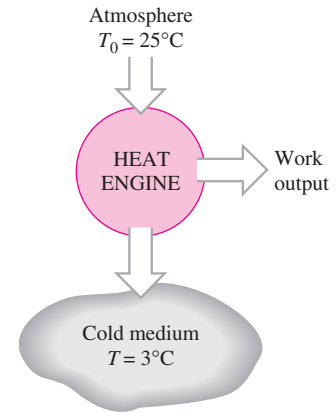

FIGURE 8–21

The *exergy* of a cold medium is also a *positive* quantity since work can be produced by transferring heat to it.

Exergy of a Flow Stream: Flow (or Stream) Exergy

In Chap. 5 it was shown that a flowing fluid has an additional form of energy, called the *flow energy*, which is the energy needed to maintain flow in a pipe or duct, and was expressed as $w_{\text{flow}} = Pv$ where v is the specific volume of the fluid, which is equivalent to the *volume change* of a unit mass of the fluid as it is displaced during flow. The flow work is essentially the boundary work done by a fluid on the fluid downstream, and thus the exergy associated with flow work is equivalent to the exergy associated with the boundary work, which is the boundary work in excess of the work done against the atmospheric air at P_0 to displace it by a volume v (Fig. 8–22). Noting that the flow work is Pv and the work done against the atmosphere is P_0v, the *exergy* associated with flow energy can be expressed as

$$x_{\text{flow}} = Pv - P_0v = (P - P_0)v \tag{8-20}$$

Therefore, the exergy associated with flow energy is obtained by replacing the pressure P in the flow work relation by the pressure in excess of the atmospheric pressure, $P - P_0$. Then the exergy of a flow stream is determined by simply adding the flow exergy relation above to the exergy relation in Eq. 8–16 for a nonflowing fluid,

$$x_{\text{flowing fluid}} = x_{\text{nonflowing fluid}} + x_{\text{flow}} \tag{8-21}$$

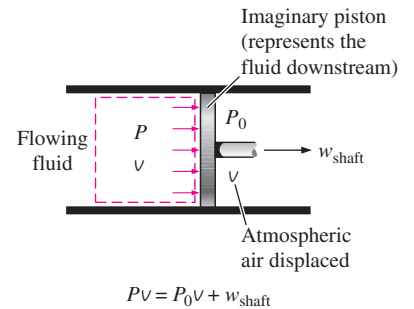$$= (u - u_0) + P_0(v - v_0) - T_0(s - s_0) + \frac{V^2}{2} + gz + (P - P_0)v$$

FIGURE 8–22

The *exergy* associated with *flow energy* is the useful work that would be delivered by an imaginary piston in the flow section.
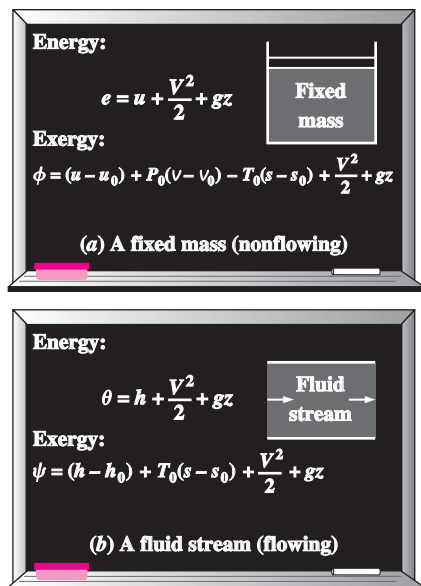
Energy:

$$e = u + \frac{V^2}{2} + gz$$

Fixed mass

Exergy:

$$\phi = (u - u_0) + P_0(v - v_0) - T_0(s - s_0) + \frac{V^2}{2} + gz$$

(a) A fixed mass (nonflowing)

Energy:

$$\theta = h + \frac{V^2}{2} + gz$$

Fluid stream

Exergy:

$$\psi = (h - h_0) + T_0(s - s_0) + \frac{V^2}{2} + gz$$

(b) A fluid stream (flowing)

FIGURE 8–23

The *energy* and *exergy* contents of (a) a fixed mass and (b) a fluid stream.

$$= (u + Pv) - (u_0 + P_0 v_0) - T_0(s - s_0) + \frac{V^2}{2} + gz$$

$$= (h - h_0) - T_0(s - s_0) + \frac{V^2}{2} + gz$$

The final expression is called **flow** (or **stream**) **exergy,** and is denoted by ψ (Fig. 8–23).

Flow exergy:
$$\psi = (h - h_0) - T_0(s - s_0) + \frac{V^2}{2} + gz \qquad \text{(8–22)}$$

Then the *exergy change* of a fluid stream as it undergoes a process from state 1 to state 2 becomes

$$\Delta\psi = \psi_2 - \psi_1 = (h_2 - h_1) + T_0(s_2 - s_1) + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1) \qquad \text{(8–23)}$$

For fluid streams with negligible kinetic and potential energies, the kinetic and potential energy terms drop out.

Note that the *exergy change* of a closed system or a fluid stream represents the *maximum* amount of useful work that can be done (or the *minimum* amount of useful work that needs to be supplied if it is negative) as the system changes from state 1 to state 2 in a specified environment, and represents the *reversible work* W_{rev}. It is independent of the type of process executed, the kind of system used, and the nature of energy interactions with the surroundings. Also note that the exergy of a closed system cannot be negative, but the exergy of a flow stream can at pressures below the environment pressure P_0.

COMPRESSED AIR

1 MPa

300 K

FIGURE 8–24

Schematic for Example 8–7.

EXAMPLE 8–7   **Work Potential of Compressed Air in a Tank**

A 200-m³ rigid tank contains compressed air at 1 MPa and 300 K. Determine how much work can be obtained from this air if the environment conditions are 100 kPa and 300 K.

Solution   Compressed air stored in a large tank is considered. The work potential of this air is to be determined.

Assumptions   **1** Air is an ideal gas. **2** The kinetic and potential energies are negligible.

Analysis   We take the air in the rigid tank as the system (Fig. 8–24). This is a *closed system* since no mass crosses the system boundary during the process. Here the question is the work potential of a fixed mass, which is the nonflow exergy by definition.

Taking the state of the air in the tank to be state 1 and noting that $T_1 = T_0 = 300$ K, the mass of air in the tank is

$$m_1 = \frac{P_1 V}{R T_1} = \frac{(1000 \text{ kPa})(200 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(300 \text{ K})} = 2323 \text{ kg}$$

The exergy content of the compressed air can be determined from

$$X_1 = m\phi_1$$

$$= m\left[(u_1 - u_0)^{\nearrow 0} + P_0(v_1 - v_0) - T_0(s_1 - s_0) + \frac{V_1^2}{2}^{\nearrow 0} + gz_1^{\nearrow 0} \right]$$

$$= m[P_0(v_1 - v_0) - T_0(s_1 - s_0)]$$

We note that

$$P_0(v_1 - v_0) = P_0\left(\frac{RT_1}{P_1} - \frac{RT_0}{P_0} \right) = RT_0\left(\frac{P_0}{P_1} - 1 \right) \qquad \text{(since } T_1 = T_0\text{)}$$

$$T_0(s_1 - s_0) = T_0\left(c_p \ln \frac{T_1}{T_0} - R \ln \frac{P_1}{P_0} \right) = -RT_0 \ln \frac{P_1}{P_0} \qquad \text{(since } T_1 = T_0\text{)}$$

Therefore,

$$\phi_1 = RT_0\left(\frac{P_0}{P_1} - 1 \right) + RT_0 \ln \frac{P_1}{P_0} = RT_0\left(\ln \frac{P_1}{P_0} + \frac{P_0}{P_1} - 1 \right)$$

$$= (0.287 \text{ kJ/kg} \cdot \text{K})(300 \text{ K})\left(\ln \frac{1000 \text{ kPa}}{100 \text{ kPa}} + \frac{100 \text{ kPa}}{1000 \text{ kPa}} - 1 \right)$$

$$= 120.76 \text{ kJ/kg}$$

and

$$X_1 = m_1\phi_1 = (2323 \text{ kg})(120.76 \text{ kJ/kg}) = 280{,}525 \text{ kJ} \cong \textbf{281 MJ}$$

Discussion The work potential of the system is 281 MJ, and thus a maximum of 281 MJ of useful work can be obtained from the compressed air stored in the tank in the specified environment.

EXAMPLE 8–8 Exergy Change during a Compression Process

Refrigerant-134a is to be compressed from 0.14 MPa and $-10°C$ to 0.8 MPa and 50°C steadily by a compressor. Taking the environment conditions to be 20°C and 95 kPa, determine the exergy change of the refrigerant during this process and the minimum work input that needs to be supplied to the compressor per unit mass of the refrigerant.

Solution Refrigerant-134a is being compressed from a specified inlet state to a specified exit state. The exergy change of the refrigerant and the minimum compression work per unit mass are to be determined.
Assumptions 1 Steady operating conditions exist. 2 The kinetic and potential energies are negligible.
Analysis We take the *compressor* as the system (Fig. 8–25). This is a *control volume* since mass crosses the system boundary during the process. Here the question is the exergy change of a fluid stream, which is the change in the flow exergy ψ.
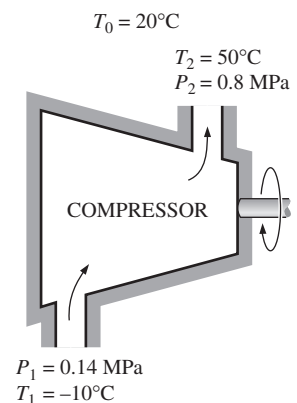
$T_0 = 20°C$

$T_2 = 50°C$
$P_2 = 0.8 \text{ MPa}$

COMPRESSOR

$P_1 = 0.14 \text{ MPa}$
$T_1 = -10°C$

FIGURE 8–25
Schematic for Example 8–8.

The properties of the refrigerant at the inlet and the exit states are

Inlet state: $\left.\begin{array}{l} P_1 = 0.14 \text{ MPa} \\ T_1 = -10°C \end{array}\right\}$ $\begin{array}{l} h_1 = 246.36 \text{ kJ/kg} \\ s_1 = 0.9724 \text{ kJ/kg·K} \end{array}$

Exit state: $\left.\begin{array}{l} P_2 = 0.8 \text{ MPa} \\ T_2 = 50°C \end{array}\right\}$ $\begin{array}{l} h_2 = 286.69 \text{ kJ/kg} \\ s_2 = 0.9802 \text{ kJ/kg·K} \end{array}$

The exergy change of the refrigerant during this compression process is determined directly from Eq. 8–23 to be

$$\Delta\psi = \psi_2 - \psi_1 = (h_2 - h_1) - T_0(s_2 - s_1) + \frac{V_2^2 - V_1^{2\nearrow 0}}{2} + g(z_2 - z_1)^{\nearrow 0}$$

$$= (h_2 - h_1) - T_0(s_2 - s_1)$$

$$= (286.69 - 246.36) \text{ kJ/kg} - (293 \text{ K})[(0.9802 - 0.9724)\text{kJ/kg·K}]$$

$$= \mathbf{38.0 \text{ kJ/kg}}$$

Therefore, the exergy of the refrigerant increases during compression by 38.0 kJ/kg.

The exergy change of a system in a specified environment represents the reversible work in that environment, which is the minimum work input required for work-consuming devices such as compressors. Therefore, the increase in exergy of the refrigerant is equal to the minimum work that needs to be supplied to the compressor:

$$w_{\text{in,min}} = \psi_2 - \psi_1 = \mathbf{38.0 \text{ kJ/kg}}$$

Discussion   Note that if the compressed refrigerant at 0.8 MPa and 50°C were to be expanded to 0.14 MPa and −10°C in a turbine in the same environment in a reversible manner, 38.0 kJ/kg of work would be produced.

INTERACTIVE TUTORIAL

SEE TUTORIAL CH. 8, SEC. 5 ON THE DVD.

8–5 · EXERGY TRANSFER BY HEAT, WORK, AND MASS

Exergy, like energy, can be transferred to or from a system in three forms: *heat, work,* and *mass flow.* Exergy transfer is recognized at the system boundary as exergy crosses it, and it represents the exergy gained or lost by a system during a process. The only two forms of exergy interactions associated with a fixed mass or closed system are *heat transfer* and *work.*

Exergy by Heat Transfer, *Q*

Recall from Chap. 6 that the work potential of the energy transferred from a heat source at temperature T is the maximum work that can be obtained from that energy in an environment at temperature T_0 and is equivalent to the work produced by a Carnot heat engine operating between the source and the environment. Therefore, the Carnot efficiency $\eta_c = 1 - T_0/T$ represents the fraction of energy of a heat source at temperature T that can be converted to work (Fig. 8–26). For example, only 70 percent of the energy transferred from a heat source at $T = 1000$ K can be converted to work in an environment at $T_0 = 300$ K.

Heat is a form of disorganized energy, and thus only a portion of it can be converted to work, which is a form of organized energy (the second law). We can always produce work from heat at a temperature above the environment temperature by transferring it to a heat engine that rejects the waste heat to the environment. Therefore, heat transfer is always accompanied by exergy transfer. Heat transfer Q at a location at thermodynamic temperature T is always accompanied by *exergy transfer* X_{heat} in the amount of

Exergy transfer by heat: $$X_{\text{heat}} = \left(1 - \frac{T_0}{T}\right)Q \qquad \text{(kJ)} \qquad \textbf{(8–24)}$$

This relation gives the exergy transfer accompanying heat transfer Q whether T is greater than or less than T_0. When $T > T_0$, heat transfer to a system increases the exergy of that system and heat transfer from a system decreases it. But the opposite is true when $T < T_0$. In this case, the heat transfer Q is the heat rejected to the cold medium (the waste heat), and it should not be confused with the heat supplied by the environment at T_0. The exergy transferred with heat is zero when $T = T_0$ at the point of transfer.

Perhaps you are wondering what happens when $T < T_0$. That is, what if we have a medium that is at a lower temperature than the environment? In this case it is conceivable that we can run a heat engine between the environment and the "cold" medium, and thus a cold medium offers us an opportunity to produce work. However, this time the environment serves as the heat source and the cold medium as the heat sink. In this case, the relation above gives the negative of the exergy transfer associated with the heat Q transferred to the cold medium. For example, for $T = 100$ K and a heat transfer of $Q = 1$ kJ to the medium, Eq. 8–24 gives $X_{\text{heat}} = (1 - 300/100)(1\ \text{kJ}) = -2$ kJ, which means that the exergy of the cold medium decreases by 2 kJ. It also means that this exergy can be recovered, and the cold medium–environment combination has the potential to produce 2 units of work for each unit of heat rejected to the cold medium at 100 K. That is, a Carnot heat engine operating between $T_0 = 300$ K and $T = 100$ K produces 2 units of work while rejecting 1 unit of heat for each 3 units of heat it receives from the environment.

When $T > T_0$, the exergy and heat transfer are in the same direction. That is, both the exergy and energy content of the medium to which heat is transferred increase. When $T < T_0$ (cold medium), however, the exergy and heat transfer are in opposite directions. That is, the energy of the cold medium increases as a result of heat transfer, but its exergy decreases. The exergy of the cold medium eventually becomes zero when its temperature reaches T_0. Equation 8–24 can also be viewed as the *exergy associated with thermal energy Q at temperature T*.

When the temperature T at the location where heat transfer is taking place is not constant, the exergy transfer accompanying heat transfer is determined by integration to be

$$X_{\text{heat}} = \int \left(1 - \frac{T_0}{T}\right)\delta Q \qquad \textbf{(8–25)}$$

HEAT SOURCE
Temperature: T
Energy transferred: E

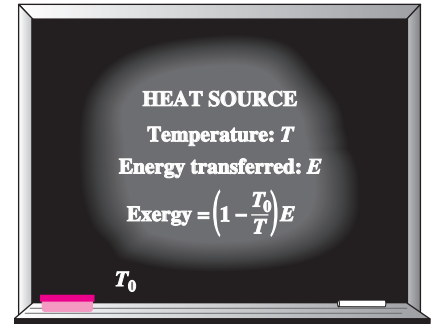Exergy $= \left(1 - \frac{T_0}{T}\right)E$

T_0

FIGURE 8–26
The Carnot efficiency $\eta_c = 1 - T_0/T$ represents the fraction of the energy transferred from a heat source at temperature T that can be converted to work in an environment at temperature T_0.
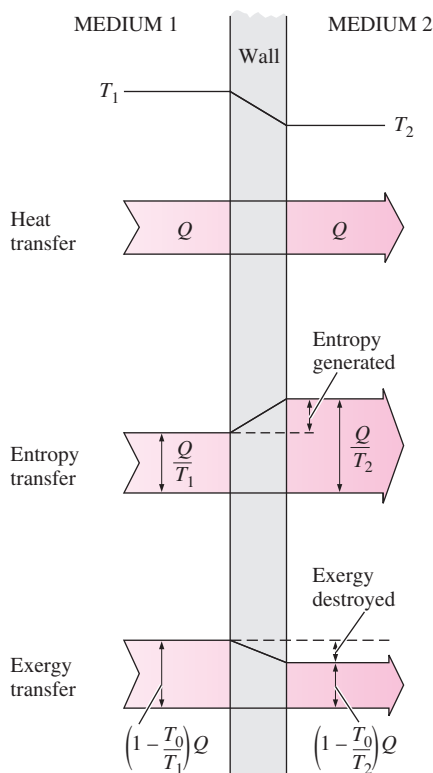
FIGURE 8–27

The transfer and destruction of exergy during a heat transfer process through a finite temperature difference.
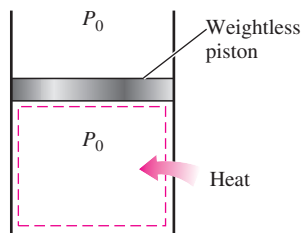

FIGURE 8–28

There is no useful work transfer associated with boundary work when the pressure of the system is maintained constant at atmospheric pressure.

Note that heat transfer through a finite temperature difference is irreversible, and some entropy is generated as a result. The entropy generation is always accompanied by exergy destruction, as illustrated in Fig. 8–27. Also note that *heat transfer Q at a location at temperature T* is always accompanied by *entropy transfer* in the amount of Q/T and *exergy transfer* in the amount of $(1 - T_0/T)Q$.

Exergy Transfer by Work, *W*

Exergy is the useful work potential, and the exergy transfer by work can simply be expressed as

Exergy transfer by work: $\quad X_{work} = \begin{cases} W - W_{surr} & \text{(for boundary work)} \\ W & \text{(for other forms of work)} \end{cases}$ **(8–26)**

where $W_{surr} = P_0(V_2 - V_1)$, P_0 is atmospheric pressure, and V_1 and V_2 are the initial and final volumes of the system. Therefore, the exergy transfer with work such as shaft work and electrical work is equal to the work W itself. In the case of a system that involves boundary work, such as a piston–cylinder device, the work done to push the atmospheric air out of the way during expansion cannot be transferred, and thus it must be subtracted. Also, during a compression process, part of the work is done by the atmospheric air, and thus we need to supply less useful work from an external source.

To clarify this point further, consider a vertical cylinder fitted with a weightless and frictionless piston (Fig. 8–28). The cylinder is filled with a gas that is maintained at the atmospheric pressure P_0 at all times. Heat is now transferred to the system and the gas in the cylinder expands. As a result, the piston rises and boundary work is done. However, this work cannot be used for any useful purpose since it is just enough to push the atmospheric air aside. (If we connect the piston to an external load to extract some useful work, the pressure in the cylinder will have to rise above P_0 to beat the resistance offered by the load.) When the gas is cooled, the piston moves down, compressing the gas. Again, no work is needed from an external source to accomplish this compression process. Thus we conclude that the work done by or against the atmosphere is not available for any useful purpose, and should be excluded from available work.

Exergy Transfer by Mass, *m*

Mass contains *exergy* as well as energy and entropy, and the exergy, energy, and entropy contents of a system are proportional to mass. Also, the rates of exergy, entropy, and energy transport into or out of a system are proportional to the mass flow rate. Mass flow is a mechanism to transport exergy, entropy, and energy into or out of a system. When mass in the amount of m enters or leaves a system, exergy in the amount of $m\psi$, where $\psi = (h - h_0) - T_0(s - s_0) + V^2/2 + gz$, accompanies it. That is,

Exergy transfer by mass: $\quad\quad\quad\quad\quad X_{mass} = m\psi$ **(8–27)**

Therefore, the exergy of a system increases by $m\psi$ when mass in the amount of m enters, and decreases by the same amount when the same amount of mass at the same state leaves the system (Fig. 8–29).

Exergy flow associated with a fluid stream when the fluid properties are variable can be determined by integration from

$$\dot{X}_{\text{mass}} = \int_{A_c} \psi \rho V_n \, dA_c \quad \text{and} \quad X_{\text{mass}} = \int \psi \, \delta m = \int_{\Delta t} \dot{X}_{\text{mass}} \, dt \quad \text{(8–28)}$$

where A_c is the cross-sectional area of the flow and V_n is the local velocity normal to dA_c.

Note that exergy transfer by heat X_{heat} is zero for adiabatic systems, and the exergy transfer by mass X_{mass} is zero for systems that involve no mass flow across their boundaries (i.e., closed systems). The total exergy transfer is zero for isolated systems since they involve no heat, work, or mass transfer.
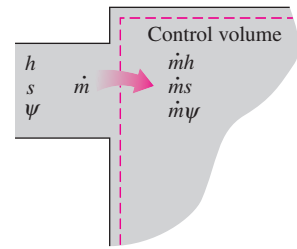

FIGURE 8–29

Mass contains energy, entropy, and exergy, and thus mass flow into or out of a system is accompanied by energy, entropy, and exergy transfer.

8–6 ▪ THE DECREASE OF EXERGY PRINCIPLE AND EXERGY DESTRUCTION

In Chap. 2 we presented the *conservation of energy principle* and indicated that energy cannot be created or destroyed during a process. In Chap. 7 we established the *increase of entropy principle,* which can be regarded as one of the statements of the second law, and indicated that entropy can be created but cannot be destroyed. That is, entropy generation S_{gen} must be positive (actual processes) or zero (reversible processes), but it cannot be negative. Now we are about to establish an alternative statement of the second law of thermodynamics, called the *decrease of exergy principle,* which is the counterpart of the increase of entropy principle.

Consider an *isolated system* shown in Fig. 8–30. By definition, no heat, work, or mass can cross the boundaries of an isolated system, and thus there is no energy and entropy transfer. Then the *energy* and *entropy* balances for an isolated system can be expressed as

Energy balance: $\qquad E_{\text{in}}^{\nearrow 0} - E_{\text{out}}^{\nearrow 0} = \Delta E_{\text{system}} \rightarrow 0 = E_2 - E_1$

Entropy balance: $\qquad S_{\text{in}}^{\nearrow 0} - S_{\text{out}}^{\nearrow 0} + S_{\text{gen}} = \Delta S_{\text{system}} \rightarrow S_{\text{gen}} = S_2 - S_1$

Multiplying the second relation by T_0 and subtracting it from the first one gives

$$-T_0 S_{\text{gen}} = E_2 - E_1 - T_0(S_2 - S_1) \quad \text{(8–29)}$$

From Eq. 8–17 we have

$$X_2 - X_1 = (E_2 - E_1) + P_0(V_2 - V_1)^{\nearrow 0} - T_0(S_2 - S_1) \quad \text{(8–30)}$$
$$= (E_2 - E_1) - T_0(S_2 - S_1)$$

since $V_2 = V_1$ for an isolated system (it cannot involve any moving boundary and thus any boundary work). Combining Eqs. 8–29 and 8–30 gives

$$-T_0 S_{\text{gen}} = X_2 - X_1 \leq 0 \quad \text{(8–31)}$$

since T_0 is the thermodynamic temperature of the environment and thus a positive quantity, $S_{\text{gen}} \geq 0$, and thus $T_0 S_{\text{gen}} \geq 0$. Then we conclude that

$$\Delta X_{\text{isolated}} = (X_2 - X_1)_{\text{isolated}} \leq 0 \quad \text{(8–32)}$$

INTERACTIVE TUTORIAL

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No heat, work or mass transfer

Isolated system

$\Delta X_{\text{isolated}} \leq 0$
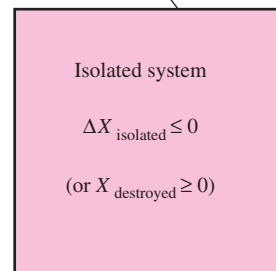
(or $X_{\text{destroyed}} \geq 0$)

FIGURE 8–30

The isolated system considered in the development of the decrease of exergy principle.

This equation can be expressed as *the exergy of an isolated system during a process always decreases or, in the limiting case of a reversible process, remains constant.* In other words, it *never* increases and *exergy is destroyed* during an actual process. This is known as the **decrease of exergy principle.** For an isolated system, the decrease in exergy equals exergy destroyed.

Exergy Destruction

Irreversibilities such as friction, mixing, chemical reactions, heat transfer through a finite temperature difference, unrestrained expansion, nonquasi-equilibrium compression or expansion always *generate entropy,* and anything that generates entropy always *destroys exergy.* The **exergy destroyed** is proportional to the entropy generated, as can be seen from Eq. 8–31, and is expressed as

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} \geq 0 \qquad (8\text{–}33)$$

Note that exergy destroyed is a *positive quantity* for any actual process and becomes *zero* for a reversible process. Exergy destroyed represents the lost work potential and is also called the *irreversibility* or *lost work.*

Equations 8–32 and 8–33 for the decrease of exergy and the exergy destruction are applicable to *any kind of system* undergoing *any kind of process* since any system and its surroundings can be enclosed by a sufficiently large arbitrary boundary across which there is no heat, work, and mass transfer, and thus any system and its surroundings constitute an *isolated system.*

No actual process is truly reversible, and thus some exergy is destroyed during a process. Therefore, the exergy of the universe, which can be considered to be an isolated system, is continuously decreasing. The more irreversible a process is, the larger the exergy destruction during that process. No exergy is destroyed during a reversible process ($X_{\text{destroyed,rev}} = 0$).

The decrease of exergy principle does not imply that the exergy of a system cannot increase. The exergy change of a system *can* be positive or negative during a process (Fig. 8–31), but exergy destroyed cannot be negative. The decrease of exergy principle can be summarized as follows:

$$X_{\text{destroyed}} \begin{cases} > 0 & \text{Irreversible process} \\ = 0 & \text{Reversible process} \\ < 0 & \text{Impossible process} \end{cases} \qquad (8\text{–}34)$$

This relation serves as an alternative criterion to determine whether a process is reversible, irreversible, or impossible.

8–7 · EXERGY BALANCE: CLOSED SYSTEMS

The nature of exergy is opposite to that of entropy in that exergy can be *destroyed,* but it cannot be created. Therefore, the *exergy change* of a system during a process is less than the *exergy transfer* by an amount equal to the *exergy destroyed* during the process within the system boundaries. Then the *decrease of exergy principle* can be expressed as (Fig. 8–32)

$$\begin{pmatrix} \text{Total} \\ \text{exergy} \\ \text{entering} \end{pmatrix} - \begin{pmatrix} \text{Total} \\ \text{exergy} \\ \text{leaving} \end{pmatrix} - \begin{pmatrix} \text{Total} \\ \text{exergy} \\ \text{destroyed} \end{pmatrix} = \begin{pmatrix} \text{Change in the} \\ \text{total exergy} \\ \text{of the system} \end{pmatrix}$$
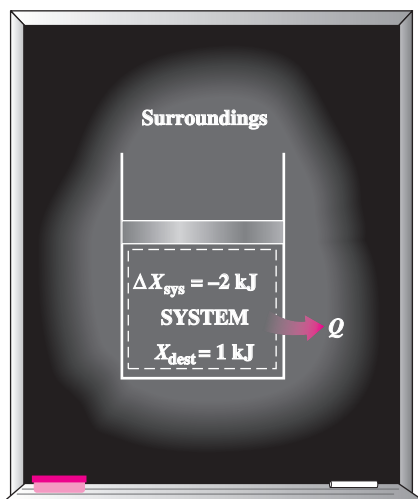

FIGURE 8–31

The exergy change of a system can be negative, but the exergy destruction cannot.

INTERACTIVE TUTORIAL

SEE TUTORIAL CH. 8, SEC. 7 ON THE DVD.

or

$$X_{in} - X_{out} - X_{destroyed} = \Delta X_{system} \qquad \textbf{(8–35)}$$

This relation is referred to as the **exergy balance** and can be stated as *the exergy change of a system during a process is equal to the difference between the net exergy transfer through the system boundary and the exergy destroyed within the system boundaries as a result of irreversibilities.*

We mentioned earlier that exergy can be transferred to or from a system by heat, work, and mass transfer. Then the exergy balance for *any system* undergoing *any process* can be expressed more explicitly as

General: $\qquad \underbrace{X_{in} - X_{out}}_{\substack{\text{Net exergy transfer} \\ \text{by heat, work, and mass}}} - \underbrace{X_{destroyed}}_{\substack{\text{Exergy} \\ \text{destruction}}} = \underbrace{\Delta X_{system}}_{\substack{\text{Change} \\ \text{in exergy}}} \quad \text{(kJ)} \qquad \textbf{(8–36)}$

or, in the **rate form,** as

General, rate form: $\qquad \underbrace{\dot{X}_{in} - \dot{X}_{out}}_{\substack{\text{Rate of net exergy transfer} \\ \text{by heat, work, and mass}}} - \underbrace{\dot{X}_{destroyed}}_{\substack{\text{Rate of exergy} \\ \text{destruction}}} = \underbrace{dX_{system}/dt}_{\substack{\text{Rate of change} \\ \text{in exergy}}} \quad \text{(kW)} \qquad \textbf{(8–37)}$

where the rates of exergy transfer by heat, work, and mass are expressed as $\dot{X}_{heat} = (1 - T_0/T)\dot{Q}$, $\dot{X}_{work} = \dot{W}_{useful}$, and $\dot{X}_{mass} = \dot{m}\psi$, respectively. The exergy balance can also be expressed per unit mass as

General, unit-mass basis: $\quad (x_{in} - x_{out}) - x_{destroyed} = \Delta x_{system} \qquad \text{(kJ/kg)} \quad \textbf{(8–38)}$

where all the quantities are expressed per unit mass of the system. Note that for a *reversible process,* the exergy destruction term $X_{destroyed}$ drops out from all of the relations above. Also, it is usually more convenient to find the entropy generation S_{gen} first, and then to evaluate the exergy destroyed directly from Eq. 8–33. That is,

$$X_{destroyed} = T_0 S_{gen} \quad \text{or} \quad \dot{X}_{destroyed} = T_0 \dot{S}_{gen} \qquad \textbf{(8–39)}$$

When the environment conditions P_0 and T_0 and the end states of the system are specified, the exergy change of the system $\Delta X_{system} = X_2 - X_1$ can be determined directly from Eq. 8–17 regardless of how the process is executed. However, the determination of the exergy transfers by heat, work, and mass requires a knowledge of these interactions.

A *closed system* does not involve any mass flow and thus any exergy transfer associated with mass flow. Taking the positive direction of heat transfer to be to the system and the positive direction of work transfer to be from the system, the exergy balance for a closed system can be expressed more explicitly as (Fig. 8–33)

Closed system: $\qquad X_{heat} - X_{work} - X_{destroyed} = \Delta X_{system} \qquad \textbf{(8–40)}$

or

Closed system: $\sum \left(1 - \dfrac{T_0}{T_k}\right)Q_k - [W - P_0(V_2 - V_1)] - T_0 S_{gen} = X_2 - X_1$ **(8–41)**
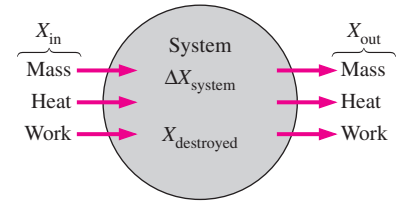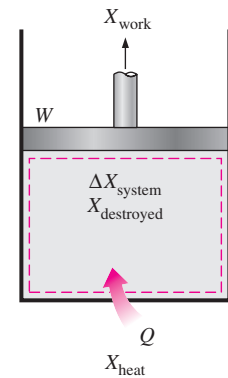

FIGURE 8–32

Mechanisms of exergy transfer.

$X_{heat} - X_{work} - X_{destroyed} = \Delta X_{system}$

FIGURE 8–33

Exergy balance for a closed system when the direction of heat transfer is taken to be to the system and the direction of work from the system.
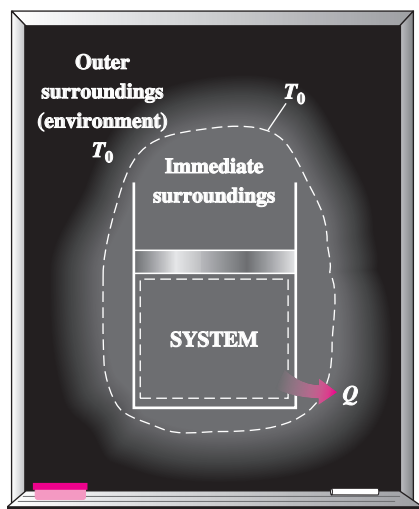
where Q_k is the heat transfer through the boundary at temperature T_k at location k. Dividing the previous equation by the time interval Δt and taking the limit as $\Delta t \to 0$ gives the *rate form* of the exergy balance for a closed system,

Rate form: $$\sum \left(1 - \frac{T_0}{T_k} \right) \dot{Q}_k - \left(\dot{W} - P_0 \frac{d V_{\text{system}}}{dt} \right) - T_0 \dot{S}_{\text{gen}} = \frac{dX_{\text{system}}}{dt} \quad \text{(8–42)}$$

Note that the relations above for a closed system are developed by taking the heat transfer to a system and work done by the system to be positive quantities. Therefore, heat transfer from the system and work done on the system should be taken to be negative quantities when using those relations.

The exergy balance relations presented above can be used to determine the *reversible work* W_{rev} by setting the exergy destruction term equal to zero. The work W in that case becomes the reversible work. That is, $W = W_{\text{rev}}$ when $X_{\text{destroyed}} = T_0 S_{\text{gen}} = 0$.

Note that $X_{\text{destroyed}}$ represents the exergy destroyed *within the system boundary* only, and not the exergy destruction that may occur outside the system boundary during the process as a result of external irreversibilities. Therefore, a process for which $X_{\text{destroyed}} = 0$ is *internally reversible* but not necessarily *totally* reversible. The *total* exergy destroyed during a process can be determined by applying the exergy balance to an *extended system* that includes the system itself and its immediate surroundings where external irreversibilities might be occurring (Fig. 8–34). Also, the exergy change in this case is equal to the sum of the exergy changes of the system and the *exergy change* of the immediate surroundings. Note that under steady conditions, the state and thus the exergy of the immediate surroundings (the "buffer zone") at any point does not change during the process, and thus the exergy change of the immediate surroundings is zero. When evaluating the exergy transfer between an extended system and the environment, the boundary temperature of the extended system is simply taken to be the environment temperature T_0.

For a *reversible process,* the *entropy generation* and thus the *exergy destruction* are *zero,* and the exergy balance relation in this case becomes analogous to the energy balance relation. That is, the exergy change of the system becomes equal to the exergy transfer.

Note that the *energy change* of a system equals the *energy transfer* for *any* process, but the *exergy change* of a system equals the *exergy transfer* only for a *reversible* process. The *quantity* of energy is always preserved during an actual process (the first law), but the *quality* is bound to decrease (the second law). This decrease in quality is always accompanied by an increase in entropy and a decrease in exergy. When 10 kJ of heat is transferred from a hot medium to a cold one, for example, we still have 10 kJ of energy at the end of the process, but at a lower temperature, and thus at a lower quality and at a lower potential to do work.

FIGURE 8–34

Exergy destroyed outside system boundaries can be accounted for by writing an exergy balance on the extended system that includes the system and its immediate surroundings.

EXAMPLE 8–9   General Exergy Balance for Closed Systems

Starting with energy and entropy balances, derive the general exergy balance relation for a closed system (Eq. 8–41).

Solution   Starting with energy and entropy balance relations, a general relation for exergy balance for a closed system is to be obtained.

Analysis We consider a general closed system (a fixed mass) that is free to exchange heat and work with its surroundings (Fig. 8–35). The system undergoes a process from state 1 to state 2. Taking the positive direction of heat transfer to be *to* the system and the positive direction of work transfer to be *from* the system, the energy and entropy balances for this closed system can be expressed as

Energy balance: $E_{in} - E_{out} = \Delta E_{system} \rightarrow Q - W = E_2 - E_1$

Entropy balance: $S_{in} - S_{out} + S_{gen} = \Delta S_{system} \rightarrow \int_1^2 \left(\frac{\delta Q}{T}\right)_{boundary} + S_{gen} = S_2 - S_1$

Multiplying the second relation by T_0 and subtracting it from the first one gives

$$Q - T_0 \int_1^2 \left(\frac{\delta Q}{T}\right)_{boundary} - W - T_0 S_{gen} = E_2 - E_1 - T_0(S_2 - S_1)$$

However, the heat transfer for the process 1-2 can be expressed as $Q = \int_1^2 \delta Q$ and the right side of the above equation is, from Eq. 8–17, $(X_2 - X_1) - P_0(V_2 - V_1)$. Thus,

$$\int_1^2 \delta Q - T_0 \int_1^2 \left(\frac{\delta Q}{T}\right)_{boundary} - W - T_0 S_{gen} = X_2 - X_1 - P_0(V_2 - V_1)$$

Letting T_b denote the boundary temperature and rearranging give

$$\int_1^2 \left(1 - \frac{T_0}{T_b}\right)\delta Q - [W - P_0(V_2 - V_1)] - T_0 S_{gen} = X_2 - X_1 \quad \textbf{(8–43)}$$

which is equivalent to Eq. 8–41 for the exergy balance except that the integration is replaced by summation in that equation for convenience. This completes the proof.

Discussion Note that the exergy balance relation above is obtained by adding the energy and entropy balance relations, and thus it is not an independent equation. However, it can be used in place of the entropy balance relation as an alternative second law expression in exergy analysis.
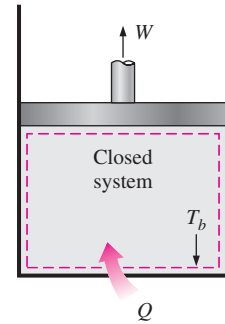

FIGURE 8–35
A general closed system considered in Example 8–9.

EXAMPLE 8–10 **Exergy Destruction during Heat Conduction**

Consider steady heat transfer through a 5-m × 6-m brick wall of a house of thickness 30 cm. On a day when the temperature of the outdoors is 0°C, the house is maintained at 27°C. The temperatures of the inner and outer surfaces of the brick wall are measured to be 20°C and 5°C, respectively, and the rate of heat transfer through the wall is 1035 W. Determine the rate of exergy destruction in the wall, and the rate of total exergy destruction associated with this heat transfer process.

Solution Steady heat transfer through a wall is considered. For specified heat transfer rate, wall surface temperatures, and environment conditions, the rate of exergy destruction within the wall and the rate of total exergy destruction are to be determined.

Assumptions **1** The process is steady, and thus the rate of heat transfer through the wall is constant. **2** The exergy change of the wall is zero during
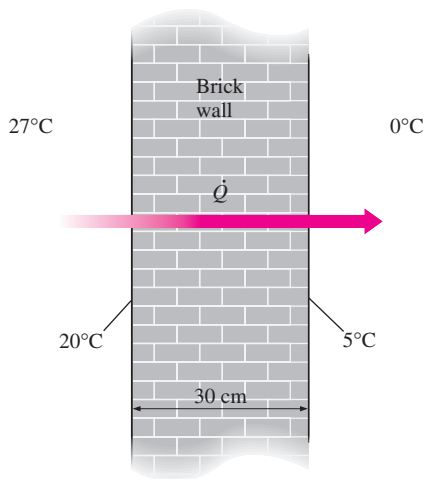
FIGURE 8–36

Schematic for Example 8–10.

this process since the state and thus the exergy of the wall do not change anywhere in the wall. **3** Heat transfer through the wall is one-dimensional.

Analysis We first take the *wall* as the system (Fig. 8–36). This is a *closed system* since no mass crosses the system boundary during the process. We note that heat and exergy are entering from one side of the wall and leaving from the other side.

Applying the rate form of the exergy balance to the wall gives

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

$$\dot{Q}\left(1 - \frac{T_0}{T}\right)_{in} - \dot{Q}\left(1 - \frac{T_0}{T}\right)_{out} - \dot{X}_{destroyed} = 0$$

$$(1035 \text{ W})\left(1 - \frac{273 \text{ K}}{293 \text{ K}}\right) - (1035 \text{ W})\left(1 - \frac{273 \text{ K}}{278 \text{ K}}\right) - \dot{X}_{destroyed} = 0$$

Solving, the rate of exergy destruction in the wall is determined to be

$$\dot{X}_{destroyed} = \textbf{52.0 W}$$

Note that exergy transfer with heat at any location is $(1 - T_0/T)Q$ at that location, and the direction of exergy transfer is the same as the direction of heat transfer.

To determine the rate of total exergy destruction during this heat transfer process, we extend the system to include the regions on both sides of the wall that experience a temperature change. Then one side of the system boundary becomes room temperature while the other side, the temperature of the outdoors. The exergy balance for this *extended system* (system + immediate surroundings) is the same as that given above, except the two boundary temperatures are 300 and 273 K instead of 293 and 278 K, respectively. Then the rate of total exergy destruction becomes

$$\dot{X}_{destroyed,total} = (1035 \text{ W})\left(1 - \frac{273 \text{ K}}{300 \text{ K}}\right) - (1035 \text{ W})\left(1 - \frac{273 \text{ K}}{273 \text{ K}}\right) = \textbf{93.2 W}$$

The difference between the two exergy destructions is 41.2 W and represents the exergy destroyed in the air layers on both sides of the wall. The exergy destruction in this case is entirely due to irreversible heat transfer through a finite temperature difference.

Discussion This problem was solved in Chap. 7 for entropy generation. We could have determined the exergy destroyed by simply multiplying the entropy generations by the environment temperature of $T_0 = 273$ K.

EXAMPLE 8–11 **Exergy Destruction during Expansion of Steam**

A piston–cylinder device contains 0.05 kg of steam at 1 MPa and 300°C. Steam now expands to a final state of 200 kPa and 150°C, doing work. Heat losses from the system to the surroundings are estimated to be 2 kJ during this process. Assuming the surroundings to be at $T_0 = 25$°C and $P_0 = 100$ kPa,

determine (a) the exergy of the steam at the initial and the final states, (b) the exergy change of the steam, (c) the exergy destroyed, and (d) the second-law efficiency for the process.

Solution   Steam in a piston–cylinder device expands to a specified state. The exergies of steam at the initial and final states, the exergy change, the exergy destroyed, and the second-law efficiency for this process are to be determined.
Assumptions   The kinetic and potential energies are negligible.
Analysis   We take the *steam* contained within the piston–cylinder device as the system (Fig. 8–37). This is a *closed system* since no mass crosses the system boundary during the process. We note that boundary work is done by the system and heat is lost from the system during the process.
(a) First we determine the properties of the steam at the initial and final states as well as the state of the surroundings:

State 1: $\left.\begin{array}{l} P_1 = 1\ \text{MPa} \\ T_1 = 300°\text{C} \end{array}\right\}$ $\begin{array}{l} u_1 = 2793.7\ \text{kJ/kg} \\ v_1 = 0.25799\ \text{m}^3/\text{kg} \\ s_1 = 7.1246\ \text{kJ/kg} \cdot \text{K} \end{array}$ (Table A–6)

State 2: $\left.\begin{array}{l} P_2 = 200\ \text{kPa} \\ T_2 = 150°\text{C} \end{array}\right\}$ $\begin{array}{l} u_2 = 2577.1\ \text{kJ/kg} \\ v_2 = 0.95986\ \text{m}^3/\text{kg} \\ s_2 = 7.2810\ \text{kJ/kg} \cdot \text{K} \end{array}$ (Table A–6)

Dead state: $\left.\begin{array}{l} P_0 = 100\ \text{kPa} \\ T_0 = 25°\text{C} \end{array}\right\}$ $\begin{array}{l} u_0 \cong u_{f\,@\,25°\text{C}} = 104.83\ \text{kJ/kg} \\ v_0 \cong v_{f\,@\,25°\text{C}} = 0.00103\ \text{m}^3/\text{kg} \\ s_0 \cong s_{f\,@\,25°\text{C}} = 0.3672\ \text{kJ/kg} \cdot \text{K} \end{array}$ (Table A–4)

The exergies of the system at the initial state X_1 and the final state X_2 are determined from Eq. 8–15 to be

$$X_1 = m[(u_1 - u_0) - T_0(s_1 - s_0) + P_0(v_1 - v_0)]$$
$$= (0.05\ \text{kg})\{(2793.7 - 104.83)\ \text{kJ/kg}$$
$$- (298\ \text{K})[(7.1246 - 0.3672)\ \text{kJ/kg} \cdot \text{K}]$$
$$+ (100\ \text{kPa})[(0.25799 - 0.00103)\ \text{m}^3/\text{kg}]\}(\text{kJ/kPa} \cdot \text{m}^3)$$
$$= \mathbf{35.0\ kJ}$$

and

$$X_2 = m[(u_2 - u_0) - T_0(s_2 - s_0) + P_0(v_2 - v_0)]$$
$$= (0.05\ \text{kg})\{(2577.1 - 104.83)\ \text{kJ/kg}$$
$$- (298\ \text{K})[(7.2810 - 0.3672)\ \text{kJ/kg} \cdot \text{K}]$$
$$+ (100\ \text{kPa})[(0.95986 - 0.00103)\ \text{m}^3/\text{kg}]\}(\text{kJ/kPa} \cdot \text{m}^3)$$
$$= \mathbf{25.4\ kJ}$$

That is, steam initially has an exergy content of 35 kJ, which drops to 25.4 kJ at the end of the process. In other words, if the steam were allowed to undergo a reversible process from the initial state to the state of the environment, it would produce 35 kJ of useful work.

(b) The exergy change for a process is simply the difference between the exergy at the initial and final states of the process,

$$\Delta X = X_2 - X_1 = 25.4 - 35.0 = \mathbf{-9.6\ kJ}$$

$P_0 = 100\ \text{kPa}$
$T_0 = 25°\text{C}$

$P_1 = 1\ \text{MP}$
$T_1 = 300°\text{C}$

2 kJ

Steam
$P_2 = 200\ \text{kPa}$
$T_2 = 150°\text{C}$
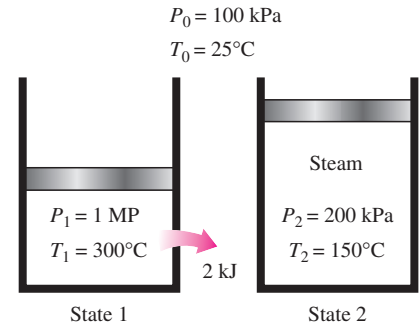
State 1   State 2

FIGURE 8–37
Schematic for Example 8–11.

That is, if the process between states 1 and 2 were executed in a reversible manner, the system would deliver 9.6 kJ of useful work.

(c) The total exergy destroyed during this process can be determined from the exergy balance applied on the *extended system* (system + immediate surroundings) whose boundary is at the environment temperature of T_0 (so that there is no exergy transfer accompanying heat transfer to or from the environment),

$$\underbrace{X_{in} - X_{out}}_{\substack{\text{Net exergy transfer} \\ \text{by heat, work, and mass}}} - \underbrace{X_{destroyed}}_{\substack{\text{Exergy} \\ \text{destruction}}} = \underbrace{\Delta X_{system}}_{\substack{\text{Change} \\ \text{in exergy}}}$$

$$-X_{work,out} - X_{heat,out} \nearrow^{0} - X_{destroyed} = X_2 - X_1$$

$$X_{destroyed} = X_1 - X_2 - W_{u,out}$$

where $W_{u,out}$ is the useful boundary work delivered as the system expands. By writing an energy balance on the system, the total boundary work done during the process is determined to be

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

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

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

$$= -(2 \text{ kJ}) - (0.05 \text{ kg})(2577.1 - 2793.7) \text{ kJ/kg}$$

$$= 8.8 \text{ kJ}$$

This is the total boundary work done by the system, including the work done against the atmosphere to push the atmospheric air out of the way during the expansion process. The useful work is the difference between the two:

$$W_u = W - W_{surr} = W_{b,out} - P_0(V_2 - V_1) = W_{b,out} - P_0 m(v_2 - v_1)$$

$$= 8.8 \text{ kJ} - (100 \text{ kPa})(0.05 \text{ kg})[(0.9599 - 0.25799) \text{ m}^3/\text{kg}]\left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right)$$

$$= 5.3 \text{ kJ}$$

Substituting, the exergy destroyed is determined to be

$$X_{destroyed} = X_1 - X_2 - W_{u,out} = 35.0 - 25.4 - 5.3 = \textbf{4.3 kJ}$$

That is, 4.3 kJ of work potential is wasted during this process. In other words, an additional 4.3 kJ of energy *could have been* converted to work during this process, but was not.

The exergy destroyed could also be determined from

$$X_{destroyed} = T_0 S_{gen} = T_0\left[m(s_2 - s_1) + \frac{Q_{surr}}{T_0}\right]$$

$$= (298 \text{ K})\left\{(0.05 \text{ kg})[(7.2810 - 7.1246) \text{ kJ/kg} \cdot \text{K}] + \frac{2 \text{ kJ}}{298 \text{ K}}\right\}$$

$$= 4.3 \text{ kJ}$$

which is the same result obtained before.

(d) Noting that the decrease in the exergy of the steam is the exergy supplied and the useful work output is the exergy recovered, the second-law efficiency for this process can be determined from

$$\eta_{II} = \frac{\text{Exergy recovered}}{\text{Exergy supplied}} = \frac{W_u}{X_1 - X_2} = \frac{5.3}{35.0 - 25.4} = \textbf{0.552 or 55.2\%}$$

That is, 44.8 percent of the work potential of the steam is wasted during this process.

EXAMPLE 8–12 Exergy Destroyed during Stirring of a Gas

An insulated rigid tank contains 2 lbm of air at 20 psia and 70°F. A paddle wheel inside the tank is now rotated by an external power source until the temperature in the tank rises to 130°F (Fig. 8–38). If the surrounding air is at $T_0 = 70°F$, determine (a) the exergy destroyed and (b) the reversible work for this process.

Solution The air in an adiabatic rigid tank is heated by stirring it by a paddle wheel. The exergy destroyed and the reversible work for this process are to be determined.

Assumptions **1** Air at about atmospheric conditions can be treated as an ideal gas with constant specific heats at room temperature. **2** The kinetic and potential energies are negligible. **3** The volume of a rigid tank is constant, and thus there is no boundary work. **4** The tank is well insulated and thus there is no heat transfer.

Analysis We take the *air* contained within the tank as the system. This is a *closed system* since no mass crosses the system boundary during the process. We note that shaft work is done on the system.

(a) The exergy destroyed during a process can be determined from an exergy balance, or directly from $X_{\text{destroyed}} = T_0 S_{\text{gen}}$. We will use the second approach since it is usually easier. But first we determine the entropy generated from an entropy balance,

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change} \\ \text{in entropy}}}$$

$$0 + S_{\text{gen}} = \Delta S_{\text{system}} = m\left(c_v \ln \frac{T_2}{T_1} + R \ln \cancelto{0}{\frac{V_2}{V_1}} \right)$$

$$S_{\text{gen}} = m c_v \ln \frac{T_2}{T_1}$$

Taking $c_v = 0.172$ Btu/lbm · °F and substituting, the exergy destroyed becomes

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = T_0 m c_v \ln \frac{T_2}{T_1}$$

$$= (530 \text{ R})(2 \text{ lbm})(0.172 \text{ Btu/lbm} \cdot °\text{F}) \ln \frac{590 \text{ R}}{530 \text{ R}}$$

$$= \textbf{19.6 Btu}$$

$T_0 = 70°F$

AIR
$m = 2$ lbm
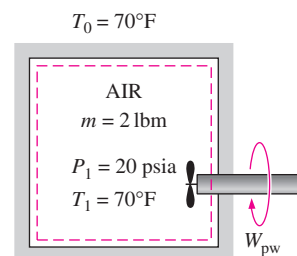$P_1 = 20$ psia
$T_1 = 70°F$

W_{pw}

FIGURE 8–38
Schematic for Example 8–12.

(*b*) The reversible work, which represents the minimum work input $W_{\text{rev,in}}$ in this case, can be determined from the exergy balance by setting the exergy destruction equal to zero,

$$\underbrace{X_{\text{in}} - X_{\text{out}}}_{\substack{\text{Net exergy transfer} \\ \text{by heat, work, and mass}}} - \underbrace{X_{\text{destroyed}}\nearrow^{0 \, \text{(reversible)}}}_{\substack{\text{Exergy} \\ \text{destruction}}} = \underbrace{\Delta X_{\text{system}}}_{\substack{\text{Change} \\ \text{in exergy}}}$$

$$W_{\text{rev,in}} = X_2 - X_1$$

$$= (E_2 - E_1) + P_0(V_2 - V_1)\nearrow^{0} - T_0(S_2 - S_1)$$

$$= (U_2 - U_1) - T_0(S_2 - S_1)$$

since $\Delta \text{KE} = \Delta \text{PE} = 0$ and $V_2 = V_1$. Noting that $T_0(S_2 - S_1) = T_0 \Delta S_{\text{system}}$ = 19.6 Btu, the reversible work becomes

$$W_{\text{rev,in}} = mc_v(T_2 - T_1) - T_0(S_2 - S_1)$$

$$= (2 \text{ lbm})(0.172 \text{ Btu/lbm} \cdot {}^\circ\text{F})(130 - 70){}^\circ\text{F} - 19.6 \text{ Btu}$$

$$= (20.6 - 19.6) \text{ Btu}$$

$$= \textbf{1.0 Btu}$$

Therefore, a work input of just 1.0 Btu would be sufficient to accomplish this process (raise the temperature of air in the tank from 70 to 130°F) if all the irreversibilities were eliminated.

Discussion The solution is complete at this point. However, to gain some physical insight, we will set the stage for a discussion. First, let us determine the actual work (the paddle-wheel work W_{pw}) done during this process. Applying the energy balance on the system,

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

$$W_{\text{pw,in}} = \Delta U = 20.6 \text{ Btu} \qquad [\text{from part } (b)]$$

since the system is adiabatic ($Q = 0$) and involves no moving boundaries ($W_b = 0$).

To put the information into perspective, 20.6 Btu of work is consumed during the process, 19.6 Btu of exergy is destroyed, and the reversible work input for the process is 1.0 Btu. What does all this mean? It simply means that we could have created the same effect on the closed system (raising its temperature to 130°F at constant volume) by consuming 1.0 Btu of work only instead of 20.6 Btu, and thus saving 19.6 Btu of work from going to waste. This would have been accomplished by a reversible heat pump.

To prove what we have just said, consider a Carnot heat pump that absorbs heat from the surroundings at $T_0 = 530$ R and transfers it to the air in the rigid tank until the air temperature T rises from 530 to 590 R, as shown in Fig. 8–39. The system involves no direct work interactions in this case, and the heat supplied to the system can be expressed in differential form as

$$\delta Q_H = dU = mc_v \, dT$$

The coefficient of performance of a reversible heat pump is given by

$$\text{COP}_{\text{HP}} = \frac{\delta Q_H}{\delta W_{\text{net,in}}} = \frac{1}{1 - T_0/T}$$
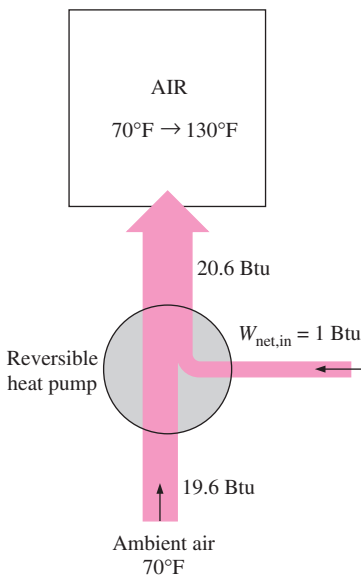

FIGURE 8–39

The same effect on the system can be accomplished by a reversible heat pump that consumes only 1 Btu of work.

Thus,

$$\delta W_{\text{net,in}} = \frac{\delta Q_H}{\text{COP}_{\text{HP}}} = \left(1 - \frac{T_0}{T}\right) mc_v \, dT$$

Integrating, we get

$$W_{\text{net,in}} = \int_1^2 \left(1 - \frac{T_0}{T}\right) mc_v \, dT$$

$$= mc_{v,\text{avg}}(T_2 - T_1) - T_0 \, mc_{v,\text{avg}} \ln \frac{T_2}{T_1}$$

$$= (20.6 - 19.6) \text{ Btu} = 1.0 \text{ Btu}$$

The first term on the right-hand side of the final expression above is recognized as ΔU and the second term as the exergy destroyed, whose values were determined earlier. By substituting those values, the total work input to the heat pump is determined to be 1.0 Btu, proving our claim. Notice that the system is still supplied with 20.6 Btu of energy; all we did in the latter case is replace the 19.6 Btu of valuable work by an equal amount of "useless" energy captured from the surroundings.

Discussion It is also worth mentioning that the exergy of the system as a result of 20.6 Btu of paddle-wheel work done on it has increased by 1.0 Btu only, that is, by the amount of the reversible work. In other words, if the system were returned to its initial state, it would produce, at most, 1.0 Btu of work.

EXAMPLE 8–13   Dropping a Hot Iron Block into Water

A 5-kg block initially at 350°C is quenched in an insulated tank that contains 100 kg of water at 30°C (Fig. 8–40). Assuming the water that vaporizes during the process condenses back in the tank and the surroundings are at 20°C and 100 kPa, determine (a) the final equilibrium temperature, (b) the exergy of the combined system at the initial and the final states, and (c) the wasted work potential during this process.

Solution A hot iron block is quenched in an insulated tank by water. The final equilibrium temperature, the initial and final exergies, and the wasted work potential are to be determined.

Assumptions 1 Both water and the iron block are incompressible substances. 2 Constant specific heats at room temperature can be used for both the water and the iron. 3 The system is stationary and thus the kinetic and potential energy changes are zero, $\Delta KE = \Delta PE = 0$. 4 There are no electrical, shaft, or other forms of work involved. 5 The system is well-insulated and thus there is no heat transfer.

Analysis We take the entire contents of the tank, *water + iron block,* as the *system*. This is a *closed system* since no mass crosses the system boundary during the process. We note that the volume of a rigid tank is constant, and thus there is no boundary work.
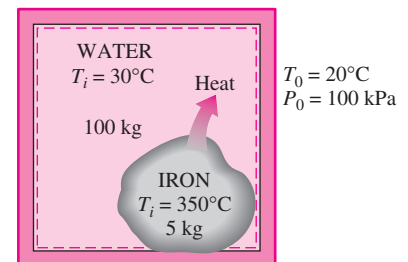

FIGURE 8–40

Schematic for Example 8–13.

(*a*) Noting that no energy enters or leaves the system during the process, the application of the energy balance gives

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

$$0 = \Delta U$$

$$0 = (\Delta U)_{iron} + (\Delta U)_{water}$$

$$0 = [mc(T_f - T_i)]_{iron} + [mc(T_f - T_i)]_{water}$$

By using the specific-heat values for water and iron at room temperature (from Table A–3), the final equilibrium temperature T_f becomes

$$0 = (5 \text{ kg})(0.45 \text{ kJ/kg} \cdot °\text{C})(T_f - 350°\text{C})$$
$$+ (100 \text{ kg})(4.18 \text{ kJ/kg} \cdot °\text{C})(T_f - 30°\text{C})$$

which yields

$$T_f = \mathbf{31.7°C}$$

(*b*) Exergy X is an extensive property, and the exergy of a composite system at a specified state is the sum of the exergies of the components of that system at that state. It is determined from Eq. 8–15, which for an incompressible substance reduces to

$$X = (U - U_0) - T_0(S - S_0) + P_0(\overset{0}{\cancel{V - V_0}})$$

$$= mc(T - T_0) - T_0 mc \ln \frac{T}{T_0} + 0$$

$$= mc\left(T - T_0 - T_0 \ln \frac{T}{T_0}\right)$$

where T is the temperature at the specified state and T_0 is the temperature of the surroundings. At the initial state,

$$X_{1,iron} = (5 \text{ kg})(0.45 \text{ kJ/kg} \cdot \text{K})\left[(623 - 293) \text{ K} - (293 \text{ K}) \ln \frac{623 \text{ K}}{293 \text{ K}}\right]$$

$$= 245.2 \text{ kJ}$$

$$X_{1,water} = (100 \text{ kg})(4.18 \text{ kJ/kg} \cdot \text{K})\left[(303 - 293) \text{ K} - (293 \text{ K}) \ln \frac{303 \text{ K}}{293 \text{ K}}\right]$$

$$= 69.8 \text{ kJ}$$

$$X_{1,total} = X_{1,iron} + X_{1,water} = (245.2 + 69.8)\text{kJ} = \mathbf{315 \text{ kJ}}$$

Similarly, the total exergy at the final state is

$$X_{2,iron} = 0.5 \text{ kJ}$$

$$X_{2,water} = 95.1 \text{ kJ}$$

$$X_{2,total} = X_{2,iron} + X_{2,water} = 0.5 + 95.1 = \mathbf{95.6 \text{ kJ}}$$

That is, the exergy of the combined system (water + iron) decreased from 315 to 95.6 kJ as a result of this irreversible heat transfer process.

(c) The wasted work potential is equivalent to the exergy destroyed, which can be determined from $X_{destroyed} = T_0 S_{gen}$ or by performing an exergy balance on the system. The second approach is more convenient in this case since the initial and final exergies of the system are already evaluated.

$$\underbrace{X_{in} - X_{out}}_{\substack{\text{Net exergy transfer} \\ \text{by heat, work, and mass}}} - \underbrace{X_{destroyed}}_{\substack{\text{Exergy} \\ \text{destruction}}} = \underbrace{\Delta X_{system}}_{\substack{\text{Change} \\ \text{in exergy}}}$$

$$0 - X_{destroyed} = X_2 - X_1$$

$$X_{destroyed} = X_1 - X_2 = 315 - 95.6 = \textbf{219.4 kJ}$$

Discussion   Note that 219.4 kJ of work could have been produced as the iron was cooled from 350 to 31.7°C and water was heated from 30 to 31.7°C, but was not.

EXAMPLE 8–14   **Exergy Destruction during Heat Transfer to a Gas**

A frictionless piston–cylinder device, shown in Fig. 8–41, initially contains 0.01 m³ of argon gas at 400 K and 350 kPa. Heat is now transferred to the argon from a furnace at 1200 K, and the argon expands isothermally until its volume is doubled. No heat transfer takes place between the argon and the surrounding atmospheric air, which is at $T_0 = 300$ K and $P_0 = 100$ kPa. Determine (a) the useful work output, (b) the exergy destroyed, and (c) the reversible work for this process.

Solution   Argon gas in a piston–cylinder device expands isothermally as a result of heat transfer from a furnace. The useful work output, the exergy destroyed, and the reversible work are to be determined.

Assumptions   **1** Argon at specified conditions can be treated as an ideal gas since it is well above its critical temperature of 151 K. **2** The kinetic and potential energies are negligible.

Analysis   We take the *argon gas* contained within the piston–cylinder device as the system. This is a *closed system* since no mass crosses the system boundary during the process. We note that heat is transferred to the system from a source at 1200 K, but there is no heat exchange with the environment at 300 K. Also, the temperature of the system remains constant during the expansion process, and its volume doubles, that is, $T_2 = T_1$ and $V_2 = 2V_1$.

(a) The only work interaction involved during this isothermal process is the quasi-equilibrium boundary work, which is determined from

$$W = W_b = \int_1^2 P \, dV = P_1 V_1 \ln \frac{V_2}{V_1} = (350 \text{ kPa})(0.01 \text{ m}^3) \ln \frac{0.02 \text{ m}^3}{0.01 \text{ m}^3}$$

$$= 2.43 \text{ kPa} \cdot \text{m}^3 = 2.43 \text{ kJ}$$

This is the total boundary work done by the argon gas. Part of this work is done against the atmospheric pressure P_0 to push the air out of the way, and it cannot be used for any useful purpose. It is determined from Eq. 8–3:

$$W_{surr} = P_0(V_2 - V_1) = (100 \text{ kPa})[(0.02 - 0.01) \text{ m}^3]\left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right) = 1 \text{ kJ}$$
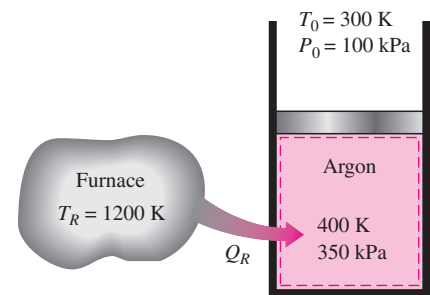

$T_0 = 300$ K
$P_0 = 100$ kPa

Furnace
$T_R = 1200$ K

Q_R

Argon

400 K
350 kPa

FIGURE 8–41
Schematic for Example 8–14.

The useful work is the difference between these two:

$$W_u = W - W_{surr} = 2.43 - 1 = \textbf{1.43 kJ}$$

That is, 1.43 kJ of the work done is available for creating a useful effect such as rotating a shaft.

Also, the heat transfer from the furnace to the system is determined from an energy balance on the system to be

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

$$Q_{in} - W_{b,out} = \Delta U = mc_V \Delta T \nearrow^{0} = 0$$

$$Q_{in} = W_{b,out} = 2.43 \text{ kJ}$$

(b) The exergy destroyed during a process can be determined from an exergy balance, or directly from $X_{destroyed} = T_0 S_{gen}$. We will use the second approach since it is usually easier. But first we determine the entropy generation by applying an entropy balance on an *extended system* (system + immediate surroundings), which includes the temperature gradient zone between the cylinder and the furnace so that the temperature at the boundary where heat transfer occurs is $T_R = 1200$ K. This way, the entropy generation associated with the heat transfer is included. Also, the entropy change of the argon gas can be determined from Q/T_{sys} since its temperature remains constant.

$$\underbrace{S_{in} - S_{out}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{gen}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{system}}_{\substack{\text{Change} \\ \text{in entropy}}}$$

$$\frac{Q}{T_R} + S_{gen} = \Delta S_{system} = \frac{Q}{T_{sys}}$$

Therefore,

$$S_{gen} = \frac{Q}{T_{sys}} - \frac{Q}{T_R} = \frac{2.43 \text{ kJ}}{400 \text{ K}} - \frac{2.43 \text{ kJ}}{1200 \text{ K}} = 0.00405 \text{ kJ}$$

and

$$X_{destroyed} = T_0 S_{gen} = (300 \text{ K})(0.00405 \text{ kJ/K}) = \textbf{1.22 kJ/K}$$

(c) The reversible work, which represents the maximum useful work that could be produced $W_{rev,out}$, can be determined from the exergy balance by setting the exergy destruction equal to zero,

$$\underbrace{X_{in} - X_{out}}_{\substack{\text{Net exergy transfer} \\ \text{by heat, work, and mass}}} - \underbrace{X_{destroyed}}_{\substack{\text{Exergy} \\ \text{destruction}}} \nearrow^{0 \text{ (reversible)}} = \underbrace{\Delta X_{system}}_{\substack{\text{Change} \\ \text{in exergy}}}$$

$$\left(1 - \frac{T_0}{T_b}\right)Q - W_{rev,out} = X_2 - X_1$$

$$= (U_2 - U_1) + P_0(V_2 - V_1) - T_0(S_2 - S_1)$$

$$= 0 + W_{surr} - T_0\frac{Q}{T_{sys}}$$

since $\Delta KE = \Delta PE = 0$ and $\Delta U = 0$ (the change in internal energy of an ideal gas is zero during an isothermal process), and $\Delta S_{sys} = Q/T_{sys}$ for isothermal processes in the absence of any irreversibilities. Then,

$$W_{rev,out} = T_0 \frac{Q}{T_{sys}} - W_{surr} + \left(1 - \frac{T_0}{T_R} \right) Q$$

$$= (300 \text{ K}) \frac{2.43 \text{ kJ}}{400 \text{ K}} - (1 \text{ kJ}) + \left(1 - \frac{300 \text{ K}}{1200 \text{ K}} \right)(2.43 \text{ kJ})$$

$$= \textbf{2.65 kJ}$$

Therefore, the useful work output would be 2.65 kJ instead of 1.43 kJ if the process were executed in a totally reversible manner.

Alternative Approach The reversible work could also be determined by applying the basics only, without resorting to exergy balance. This is done by replacing the irreversible portions of the process by reversible ones that create the same effect on the system. The useful work output of this idealized process (between the actual end states) is the reversible work.

The only irreversibility the actual process involves is the heat transfer between the system and the furnace through a finite temperature difference. This irreversibility can be eliminated by operating a reversible heat engine between the furnace at 1200 K and the surroundings at 300 K. When 2.43 kJ of heat is supplied to this heat engine, it produces a work output of

$$W_{HE} = \eta_{rev} Q_H = \left(1 - \frac{T_L}{T_H} \right) Q_H = \left(1 - \frac{300 \text{ K}}{1200 \text{ K}} \right)(2.43 \text{ kJ}) = 1.82 \text{ kJ}$$

The 2.43 kJ of heat that was transferred to the system from the source is now extracted from the surrounding air at 300 K by a reversible heat pump that requires a work input of

$$W_{HP,in} = \frac{Q_H}{COP_{HP}} = \left[\frac{Q_H}{T_H/(T_H - T_L)} \right]_{HP} = \frac{2.43 \text{ kJ}}{(400 \text{ K})/[(400 - 300) \text{ K}]} = 0.61 \text{ kJ}$$

Then the net work output of this reversible process (i.e., the reversible work) becomes

$$W_{rev} = W_u + W_{HE} - W_{HP,in} = 1.43 + 1.82 - 0.61 = 2.64 \text{ kJ}$$

which is practically identical to the result obtained before. Also, the exergy destroyed is the difference between the reversible work and the useful work, and is determined to be

$$X_{destroyed} = W_{rev,out} - W_{u,out} = 2.65 - 1.43 = 1.22 \text{ kJ}$$

which is identical to the result obtained before.

8–8 · EXERGY BALANCE: CONTROL VOLUMES

The exergy balance relations for control volumes differ from those for closed systems in that they involve one more mechanism of exergy transfer: *mass flow across the boundaries.* As mentioned earlier, mass possesses exergy as well as energy and entropy, and the amounts of these three extensive properties are

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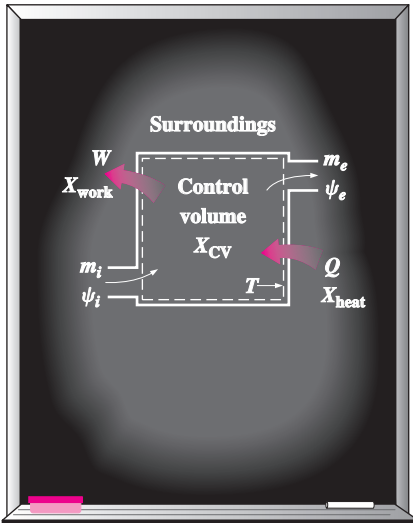
SEE TUTORIAL CH. 8, SEC. 8 ON THE DVD.

FIGURE 8–42

Exergy is transferred into or out of a control volume by mass as well as heat and work transfer.

proportional to the amount of mass (Fig. 8–42). Again taking the positive direction of heat transfer to be to the system and the positive direction of work transfer to be from the system, the general exergy balance relations (Eqs. 8–36 and 8–37) can be expressed for a control volume more explicitly as

$$X_{heat} - X_{work} + X_{mass,in} - X_{mass,out} - X_{destroyed} = (X_2 - X_1)_{CV} \quad \textbf{(8–44)}$$

or

$$\sum \left(1 - \frac{T_0}{T_k}\right) Q_k - [W - P_0(V_2 - V_1)] + \sum_{in} m\psi - \sum_{out} m\psi - X_{destroyed} = (X_2 - X_1)_{CV}$$

$$\textbf{(8–45)}$$

It can also be expressed in the **rate form** as

$$\sum \left(1 - \frac{T_0}{T_k}\right) \dot{Q}_k - \left(\dot{W} - P_0 \frac{dV_{CV}}{dt}\right) + \sum_{in} \dot{m}\psi - \sum_{out} \dot{m}\psi - \dot{X}_{destroyed} = \frac{dX_{CV}}{dt}$$

$$\textbf{(8–46)}$$

The exergy balance relation above can be stated as *the rate of exergy change within the control volume during a process is equal to the rate of net exergy transfer through the control volume boundary by heat, work, and mass flow minus the rate of exergy destruction within the boundaries of the control volume.*

When the initial and final states of the control volume are specified, the exergy change of the control volume is $X_2 - X_1 = m_2\phi_2 - m_1\phi_1$.

Exergy Balance for Steady-Flow Systems

Most control volumes encountered in practice such as turbines, compressors, nozzles, diffusers, heat exchangers, pipes, and ducts operate steadily, and thus they experience no changes in their mass, energy, entropy, and exergy contents as well as their volumes. Therefore, $dV_{CV}/dt = 0$ and $dX_{CV}/dt = 0$ for such systems, and the amount of exergy entering a steady-flow system in all forms (heat, work, mass transfer) must be equal to the amount of exergy leaving plus the exergy destroyed. Then the rate form of the general exergy balance (Eq. 8–46) reduces for a **steady-flow process** to (Fig. 8–43)
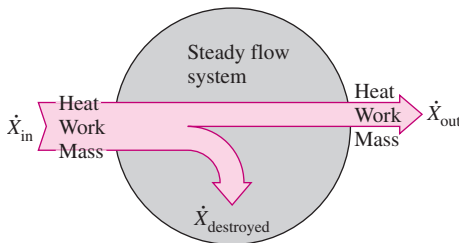

FIGURE 8–43

The exergy transfer to a steady-flow system is equal to the exergy transfer from it plus the exergy destruction within the system.

Steady-flow: $\quad \sum \left(1 - \frac{T_0}{T_k}\right) \dot{Q}_k - \dot{W} + \sum_{in} \dot{m}\psi - \sum_{out} \dot{m}\psi - \dot{X}_{destroyed} = 0 \quad \textbf{(8–47)}$

For a *single-stream* (one-inlet, one-exit) steady-flow device, the relation above further reduces to

Single-stream: $\quad \sum \left(1 - \frac{T_0}{T_k}\right) \dot{Q}_k - \dot{W} + \dot{m}(\psi_1 - \psi_2) - \dot{X}_{destroyed} = 0 \quad \textbf{(8–48)}$

where the subscripts 1 and 2 represent inlet and exit states, $\dot{m}$ is the mass flow rate, and the change in the flow exergy is given by Eq. 8–23 as

$$\psi_1 - \psi_2 = (h_1 - h_2) - T_0(s_1 - s_2) + \frac{V_1^2 - V_2^2}{2} + g(z_1 - z_2)$$

Dividing Eq. 8–48 by $\dot{m}$ gives the exergy balance on a *unit-mass basis* as

Per-unit mass: $\quad \sum \left(1 - \frac{T_0}{T_k} \right) q_k - w + (\psi_1 - \psi_2) - x_{\text{destroyed}} = 0 \qquad \text{(kJ/kg)}$ **(8–49)**

where $q = \dot{Q}/\dot{m}$ and $w = \dot{W}/\dot{m}$ are the heat transfer and work done per unit mass of the working fluid, respectively.

For the case of an *adiabatic* single-stream device with no work interactions, the exergy balance relation further simplifies to $\dot{X}_{\text{destroyed}} = \dot{m}(\psi_1 - \psi_2)$, which indicates that the specific exergy of the fluid must decrease as it flows through a work-free adiabatic device or remain the same ($\psi_2 = \psi_1$) in the limiting case of a reversible process regardless of the changes in other properties of the fluid.

Reversible Work, W_{rev}

The exergy balance relations presented above can be used to determine the reversible work W_{rev} by setting the exergy destroyed equal to zero. The work W in that case becomes the reversible work. That is,

General: $\qquad\qquad W = W_{\text{rev}} \qquad \text{when } X_{\text{destroyed}} = 0$ **(8–50)**

For example, the reversible power for a single-stream steady-flow device is, from Eq. 8–48,

Single stream: $\qquad \dot{W}_{\text{rev}} = \dot{m}(\psi_1 - \psi_2) + \sum \left(1 - \frac{T_0}{T_k} \right) \dot{Q}_k \qquad \text{(kW)}$ **(8–51)**

which reduces for an adiabatic device to

Adiabatic, single stream: $\qquad\qquad \dot{W}_{\text{rev}} = \dot{m}(\psi_1 - \psi_2)$ **(8–52)**

Note that the exergy destroyed is zero only for a reversible process, and reversible work represents the maximum work output for work-producing devices such as turbines and the minimum work input for work-consuming devices such as compressors.

Second-Law Efficiency of Steady-Flow Devices, η_{II}

The *second-law efficiency* of various steady-flow devices can be determined from its general definition, $\eta_{\text{II}} = $ (Exergy recovered)/(Exergy supplied). When the changes in kinetic and potential energies are negligible, the second-law efficiency of an *adiabatic turbine* can be determined from

$$\eta_{\text{II,turb}} = \frac{w}{w_{\text{rev}}} = \frac{h_1 - h_2}{\psi_1 - \psi_2} \quad \text{or} \quad \eta_{\text{II,turb}} = 1 - \frac{T_0 s_{\text{gen}}}{\psi_1 - \psi_2}$$ **(8–53)**

where $s_{\text{gen}} = s_2 - s_1$. For an *adiabatic compressor* with negligible kinetic and potential energies, the second-law efficiency becomes

$$\eta_{\text{II,comp}} = \frac{w_{\text{rev,in}}}{w_{\text{in}}} = \frac{\psi_2 - \psi_1}{h_2 - h_1} \quad \text{or} \quad \eta_{\text{II,comp}} = 1 - \frac{T_0 s_{\text{gen}}}{h_2 - h_1}$$ **(8–54)**

where again $s_{\text{gen}} = s_2 - s_1$.

For an adiabatic *heat exchanger* with two unmixed fluid streams (Fig. 8–44), the exergy supplied is the decrease in the exergy of the hot stream, and the exergy recovered is the increase in the exergy of the
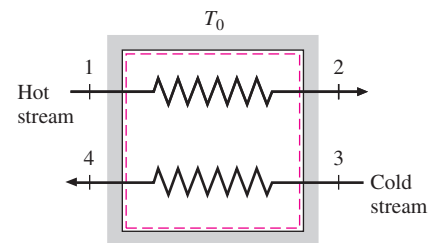

FIGURE 8–44

A heat exchanger with two unmixed fluid streams.

cold stream, provided that the cold stream is not at a lower temperature than the surroundings. Then the second-law efficiency of the heat exchanger becomes

$$\eta_{II,HX} = \frac{\dot{m}_{cold}(\psi_4 - \psi_3)}{\dot{m}_{hot}(\psi_1 - \psi_2)} \quad \text{or} \quad \eta_{II,HX} = 1 - \frac{T_0 \dot{S}_{gen}}{\dot{m}_{hot}(\psi_1 - \psi_2)} \quad \textbf{(8–55)}$$

where $\dot{S}_{gen} = \dot{m}_{hot}(s_2 - s_1) + \dot{m}_{cold}(s_4 - s_3)$. Perhaps you are wondering what happens if the heat exchanger is not adiabatic; that is, it is losing some heat to its surroundings at T_0. If the temperature of the boundary (the outer surface of the heat exchanger) T_b is equal T_0, the definition above still holds (except the entropy generation term needs to be modified if the second definition is used). However, if $T_b > T_0$, then the exergy of the lost heat at the boundary should be included in the recovered exergy. Although no attempt is made in practice to utilize this exergy and it is allowed to be destroyed, the heat exchanger should not be held responsible for this destruction, which occurs outside its boundaries. If we are interested in the exergy destroyed during the process, not just within the boundaries of the device, then it makes sense to consider an *extended system* that includes the immediate surroundings of the device such that the boundaries of the new enlarged system are at T_0. The second-law efficiency of the extended system reflects the effects of the irreversibilities that occur within and just outside the device.

An interesting situation arises when the temperature of the cold stream remains below the temperature of the surroundings at all times. In that case the exergy of the cold stream actually decreases instead of increasing. In such cases it is better to define the second-law efficiency as the ratio of the sum of the exergies of the outgoing streams to the sum of the exergies of the incoming streams.

For an adiabatic *mixing chamber* where a hot stream 1 is mixed with a cold stream 2, forming a mixture 3, the exergy supplied is the sum of the exergies of the hot and cold streams, and the exergy recovered is the exergy of the mixture. Then the second-law efficiency of the mixing chamber becomes

$$\eta_{II,mix} = \frac{\dot{m}_3 \psi_3}{\dot{m}_1 \psi_1 + \dot{m}_2 \psi_2} \quad \text{or} \quad \eta_{II,mix} = 1 - \frac{T_0 \dot{S}_{gen}}{\dot{m}_1 \psi_1 + \dot{m}_2 \psi_2} \quad \textbf{(8–56)}$$

where $\dot{m}_3 = \dot{m}_1 + \dot{m}_2$ and $\dot{S}_{gen} = \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_1 s_1$.

3 MPa
450°C

300 kW

$\dot{W}$

STEAM
TURBINE
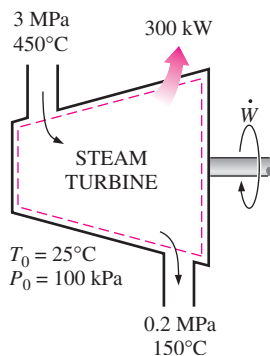
$T_0 = 25°C$
$P_0 = 100$ kPa

0.2 MPa
150°C

FIGURE 8–45

Schematic for Example 8–15.

EXAMPLE 8–15   Second-Law Analysis of a Steam Turbine

Steam enters a turbine steadily at 3 MPa and 450°C at a rate of 8 kg/s and exits at 0.2 MPa and 150°C, (Fig. 8–45). The steam is losing heat to the surrounding air at 100 kPa and 25°C at a rate of 300 kW, and the kinetic and potential energy changes are negligible. Determine (*a*) the actual power output, (*b*) the maximum possible power output, (*c*) the second-law efficiency, (*d*) the exergy destroyed, and (*e*) the exergy of the steam at the inlet conditions.

Solution   A steam turbine operating steadily between specified inlet and exit states is considered. The actual and maximum power outputs, the second-law efficiency, the exergy destroyed, and the inlet exergy are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time at any point and thus $\Delta m_{CV} = 0$, $\Delta E_{CV} = 0$, and $\Delta X_{CV} = 0$. **2** The kinetic and potential energies are negligible.

Analysis We take the *turbine* 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 $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Also, heat is lost to the surrounding air and work is done by the system.

The properties of the steam at the inlet and exit states and the state of the environment are

Inlet state: $\left. \begin{array}{l} P_1 = 3 \text{ MPa} \\ T_1 = 450°C \end{array} \right\}$ $\begin{array}{l} h_1 = 3344.9 \text{ kJ/kg} \\ s_1 = 7.0856 \text{ kJ/kg} \cdot \text{K} \end{array}$ (Table A–6)

Exit state: $\left. \begin{array}{l} P_2 = 0.2 \text{ MPa} \\ T_2 = 150°C \end{array} \right\}$ $\begin{array}{l} h_2 = 2769.1 \text{ kJ/kg} \\ s_2 = 7.2810 \text{ kJ/kg} \cdot \text{K} \end{array}$ (Table A–6)

Dead state: $\left. \begin{array}{l} P_0 = 100 \text{ kPa} \\ T_0 = 25°C \end{array} \right\}$ $\begin{array}{l} h_0 \cong h_{f @ 25°C} = 104.83 \text{ kJ/kg} \\ s_0 \cong s_{f @ 25°C} = 0.3672 \text{ kJ/kg} \cdot \text{K} \end{array}$ (Table A–4)

(*a*) The actual power output of the turbine is determined from the rate form of the energy balance,

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

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

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

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

$$= (8 \text{ kg/s})[(3344.9 - 2769.1) \text{ kJ/kg}] - 300 \text{ kW}$$

$$= \mathbf{4306 \text{ kW}}$$

(*b*) The maximum power output (reversible power) is determined from the rate form of the exergy balance applied on the *extended system* (system + immediate surroundings), whose boundary is at the environment temperature of T_0, and by setting the exergy destruction term equal to zero,

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

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

$$\dot{m}\psi_1 = \dot{W}_{rev,out} + \dot{X}_{heat}^{\nearrow 0} + \dot{m}\psi_2$$

$$\dot{W}_{rev,out} = \dot{m}(\psi_1 - \psi_2)$$

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

Note that exergy transfer with heat is zero when the temperature at the point of transfer is the environment temperature T_0. Substituting,

$$\dot{W}_{rev,out} = (8 \text{ kg/s})[(3344.9 - 2769.1) \text{ kJ/kg}$$

$$- (298 \text{ K})(7.0856 - 7.2810)\text{kJ/kg} \cdot \text{K}]$$

$$= \mathbf{4665 \text{ kW}}$$

(c) The second-law efficiency of a turbine is the ratio of the actual work delivered to the reversible work,

$$\eta_{II} = \frac{\dot{W}_{out}}{\dot{W}_{rev,out}} = \frac{4306 \text{ kW}}{4665 \text{ kW}} = \textbf{0.923 or 92.3\%}$$

That is, 7.7 percent of the work potential is wasted during this process.

(d) The difference between the reversible work and the actual useful work is the exergy destroyed, which is determined to be

$$\dot{X}_{destroyed} = \dot{W}_{rev,out} - \dot{W}_{out} = 4665 - 4306 = \textbf{359 kW}$$

That is, the potential to produce useful work is wasted at a rate of 359 kW during this process. The exergy destroyed could also be determined by first calculating the rate of entropy generation $\dot{S}_{gen}$ during the process.

(e) The exergy (maximum work potential) of the steam at the inlet conditions is simply the stream exergy, and is determined from

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

$$= (h_1 - h_0) - T_0(s_1 - s_0)$$

$$= (3344.9 - 104.83)\text{kJ/kg} - (298 \text{ K})(7.0856 - 0.3672) \text{ kJ/kg} \cdot \text{K}$$

$$= \textbf{1238 kJ/kg}$$

That is, not counting the kinetic and potential energies, every kilogram of the steam entering the turbine has a work potential of 1238 kJ. This corresponds to a power potential of (8 kg/s)(1238 kJ/kg) = 9904 kW. Obviously, the turbine is converting 4306/9904 = 43.5 percent of the available work potential of the steam to work.
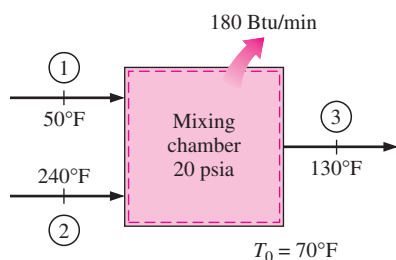
FIGURE 8–46
Schematic for Example 8–16.

EXAMPLE 8–16   **Exergy Destroyed during Mixing of Fluid Streams**

Water at 20 psia and 50°F enters a mixing chamber at a rate of 300 lbm/min, where it is mixed steadily with steam entering at 20 psia and 240°F. The mixture leaves the chamber at 20 psia and 130°F, and heat is being lost to the surrounding air at $T_0 = 70°F$ at a rate of 180 Btu/min (Fig. 8–46). Neglecting the changes in kinetic and potential energies, determine the reversible power and the rate of exergy destruction for this process.

Solution Liquid water and steam are mixed in a chamber that is losing heat at a specified rate. The reversible power and the rate of exergy destruction are to be determined.

Analysis This is a steady-flow process, which was discussed in Example 7–20 with regard to entropy generation. The mass flow rate of the steam was determined in Example 7–20 to be $\dot{m}_2 = 22.7$ lbm/min.

The maximum power output (reversible power) is determined from the rate form of the exergy balance applied on the *extended system* (system + immediate surroundings), whose boundary is at the environment temperature of T_0, and by setting the exergy destruction term equal to zero,

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

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

$$\dot{m}_1\psi_1 + \dot{m}_2\psi_2 = \dot{W}_{rev,out} + \dot{X}_{heat}^{\nearrow 0} + \dot{m}_3\psi_3$$

$$\dot{W}_{rev,out} = \dot{m}_1\psi_1 + \dot{m}_2\psi_2 - \dot{m}_3\psi_3$$

Note that exergy transfer by heat is zero when the temperature at the point of transfer is the environment temperature T_0, and the kinetic and potential energies are negligible. Therefore,

$$\begin{aligned}\dot{W}_{rev,out} &= \dot{m}_1(h_1 - T_0 s_1) + \dot{m}_2(h_2 - T_0 s_2) - \dot{m}_3(h_3 - T_0 s_3)\\
&= (300 \text{ lbm/min})[18.07 \text{ Btu/lbm} - (530 \text{ R})(0.03609 \text{ Btu/lbm}\cdot\text{R})]\\
&\quad + (22.7 \text{ lbm/min})[1162.3 \text{ Btu/lbm} - (530 \text{ R})(1.7406 \text{ Btu/lbm}\cdot\text{R})]\\
&\quad - (322.7 \text{ lbm/min})[97.99 \text{ Btu/lbm} - (530 \text{ R})(0.18174 \text{ Btu/lbm}\cdot\text{R})]\\
&= \textbf{4588 Btu/min}\end{aligned}$$

That is, we could have produced work at a rate of 4588 Btu/min if we ran a heat engine between the hot and the cold fluid streams instead of allowing them to mix directly.

The exergy destroyed is determined from

$$\dot{X}_{destroyed} = \dot{W}_{rev,out} - \dot{W}_u^{\nearrow 0} = T_0\dot{S}_{gen}$$

Thus,

$$\dot{X}_{destroyed} = \dot{W}_{rev,out} = \textbf{4588 Btu/min}$$

since there is no actual work produced during the process (Fig. 8–47).

Discussion The entropy generation rate for this process was determined in Example 7–20 to be $\dot{S}_{gen} = 8.65$ Btu/min · R. Thus the exergy destroyed could also be determined from the second part of the above equation:

$$\dot{X}_{destroyed} = T_0\dot{S}_{gen} = (530 \text{ R})(8.65 \text{ Btu/min}\cdot\text{R}) = 4585 \text{ Btu/min}$$

The slight difference between the two results is due to roundoff error.

EXAMPLE 8–17   Charging a Compressed Air Storage System

A 200-m³ rigid tank initially contains atmospheric air at 100 kPa and 300 K and is to be used as a storage vessel for compressed air at 1 MPa and 300 K (Fig. 8–48). Compressed air is to be supplied by a compressor that takes in atmospheric air at $P_0 = 100$ kPa and $T_0 = 300$ K. Determine the minimum work requirement for this process.

Solution Air is to be compressed and stored at high pressure in a large tank. The minimum work required is to be determined.

FIGURE 8–47

For systems that involve no actual work, the reversible work and irreversibility are identical.

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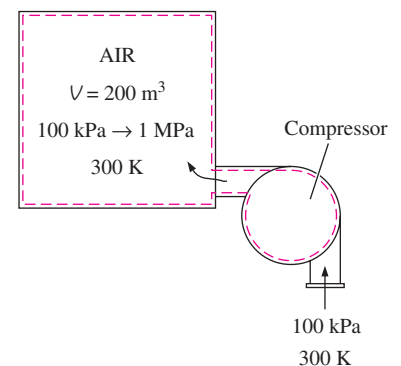

FIGURE 8–48

Schematic for Example 8–17.

Assumptions 1 Air is an ideal gas. 2 The kinetic and potential energies are negligible. 3 The properties of air at the inlet remain constant during the entire charging process.

Analysis We take the *rigid tank combined with the compressor* as the system. This is a *control volume* since mass crosses the system boundary during the process. We note that this is an unsteady-flow process since the mass content of the system changes as the tank is charged. Also, there is only one inlet and no exit.

The minimum work required for a process is the *reversible work,* which can be determined from the exergy balance applied on the *extended system* (system + immediate surroundings) whose boundary is at the environment temperature of T_0 (so that there is no exergy transfer accompanying heat transfer to or from the environment) and by setting the exergy destruction term equal to zero,

$$\underbrace{X_{\text{in}} - X_{\text{out}}}_{\substack{\text{Net exergy transfer} \\ \text{by heat, work, and mass}}} - \underbrace{X_{\text{destroyed}} \nearrow^{0 \text{ (reversible)}}}_{\substack{\text{Exergy} \\ \text{destruction}}} = \underbrace{\Delta X_{\text{system}}}_{\substack{\text{Change} \\ \text{in exergy}}}$$

$$X_{\text{in}} - X_{\text{out}} = X_2 - X_1$$

$$W_{\text{rev,in}} + m_1 \psi_1 \nearrow^0 = m_2 \phi_2 - m_1 \phi_1 \nearrow^0$$

$$W_{\text{rev,in}} = m_2 \phi_2$$

Note that $\phi_1 = \psi_1 = 0$ since the initial air in the tank and the air entering are at the state of the environment, and the exergy of a substance at the state of the environment is zero. The final mass of air and the exergy of the pressurized air in the tank at the end of the process are

$$m_2 = \frac{P_2 V}{RT_2} = \frac{(1000 \text{ kPa})(200 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(300 \text{ K})} = 2323 \text{ kg}$$

$$\phi_2 = (u_2 - u_0) \nearrow^{0 \text{ (since } T_2 = T_0)} + P_0(v_2 - v_0) - T_0(s_2 - s_0) + \frac{V_2^2}{2} \nearrow^0 + gz_2 \nearrow^0$$

$$= P_0(v_2 - v_0) - T_0(s_2 - s_0)$$

We note that

$$P_0(v_2 - v_0) = P_0 \left(\frac{RT_2}{P_2} - \frac{RT_0}{P_0} \right) = RT_0 \left(\frac{P_0}{P_2} - 1 \right) \qquad (\text{since } T_2 = T_0)$$

$$T_0(s_2 - s_0) = T_0 \left(c_p \ln \frac{T_2}{T_0} \nearrow^0 - R \ln \frac{P_2}{P_0} \right) = -RT_0 \ln \frac{P_2}{P_0} \qquad (\text{since } T_2 = T_0)$$

Therefore,

$$\phi_2 = RT_0 \left(\frac{P_0}{P_2} - 1 \right) + RT_0 \ln \frac{P_2}{P_0} = RT_0 \left(\ln \frac{P_2}{P_0} + \frac{P_0}{P_2} - 1 \right)$$

$$= (0.287 \text{ kJ/kg} \cdot \text{K})(300 \text{ K}) \left(\ln \frac{1000 \text{ kPa}}{100 \text{ kPa}} + \frac{100 \text{ kPa}}{1000 \text{ kPa}} - 1 \right)$$

$$= 120.76 \text{ kJ/kg}$$

and

$$W_{\text{rev,in}} = m_2\phi_2 = (2323\text{ kg})(120.76\text{ kJ/kg}) = 280{,}525\text{ kJ} \cong \textbf{281 MJ}$$

Discussion Note that a minimum of 281 MJ of work input is required to fill the tank with compressed air at 300 K and 1 MPa. In reality, the required work input will be greater by an amount equal to the exergy destruction during the process. Compare this to the result of Example 8–7. What can you conclude?

TOPIC OF SPECIAL INTEREST* Second-Law Aspects of Daily Life

Thermodynamics is a fundamental natural science that deals with various aspects of energy, and even nontechnical people have a basic understanding of energy and the first law of thermodynamics since there is hardly any aspect of life that does not involve the transfer or transformation of energy in different forms. All the *dieters,* for example, base their lifestyle on the conservation of energy principle. Although the first-law aspects of thermodynamics are readily understood and easily accepted by most people, there is not a public awareness about the second law of thermodynamics, and the second-law aspects are not fully appreciated even by people with technical backgrounds. This causes some students to view the second law as something that is of theoretical interest rather than an important and practical engineering tool. As a result, students show little interest in a detailed study of the second law of thermodynamics. This is unfortunate because the students end up with a one-sided view of thermodynamics and miss the balanced, complete picture.

Many *ordinary events* that go unnoticed can serve as excellent vehicles to convey important concepts of thermodynamics. Below we attempt to demonstrate the relevance of the second-law concepts such as exergy, reversible work, irreversibility, and the second-law efficiency to various aspects of daily life using examples with which even nontechnical people can identify. Hopefully, this will enhance our understanding and appreciation of the second law of thermodynamics and encourage us to use it more often in technical and even nontechnical areas. The critical reader is reminded that the concepts presented below are *soft* and *difficult to quantize,* and that they are offered here to stimulate interest in the study of the second law of thermodynamics and to enhance our understanding and appreciation of it.

The second-law concepts are implicitly used in various aspects of daily life. Many successful people seem to make extensive use of them without even realizing it. There is growing awareness that quality plays as important a role as quantity in even ordinary daily activities. The following appeared in an article in the *Reno Gazette-Journal* on March 3, 1991:

> Dr. Held considers himself a survivor of the tick-tock conspiracy. About four years ago, right around his 40th birthday, he was putting in 21-hour days—working late, working out, taking care of his three children and getting involved in sports. He got about four or five hours of sleep a night. . . .

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*This section can be skipped without a loss in continuity.

"Now I'm in bed by 9:30 and I'm up by 6," he says. "I get twice as much done as I used to. I don't have to do things twice or read things three times before I understand them."

This statement has a strong relevance to the second-law discussions. It indicates that the problem is not how much time we have (the first law), but, rather, how effectively we use it (the second law). For a person to get *more done in less time* is no different than for a car to go *more miles on less fuel.*

In thermodynamics, *reversible work* for a process is defined as the maximum useful work output (or minimum work input) for that process. It is the useful work that a system would deliver (or consume) during a process between two specified states if that process is executed in a reversible (perfect) manner. The difference between the reversible work and the actual useful work is due to imperfections and is called *irreversibility* (the wasted work potential). For the special case of the final state being the dead state or the state of the surroundings, the reversible work becomes a maximum and is called the *exergy* of the system at the initial state. The irreversibility for a reversible or perfect process is zero.

The *exergy* of a person in daily life can be viewed as the best job that person can do under the most favorable conditions. The *reversible work* in daily life, on the other hand, can be viewed as the best job a person can do under some specified conditions. Then the difference between the reversible work and the actual work done under those conditions can be viewed as the *irreversibility* or the *exergy destroyed.* In engineering systems, we try to identify the major sources of irreversibilities and minimize them in order to maximize performance. In daily life, a person should do just that to maximize his or her performance.

The exergy of a person at a given time and place can be viewed as the maximum amount of work he or she can do at that time and place. Exergy is certainly difficult to quantify because of the interdependence of physical and intellectual capabilities of a person. The ability to perform physical and intellectual tasks simultaneously complicates things even further. *Schooling* and *training* obviously increase the exergy of a person. *Aging* decreases the physical exergy. Unlike most mechanical things, the exergy of human beings is a function of time, and the physical and/or intellectual exergy of a person goes to waste if it is not utilized at the time. A barrel of oil loses nothing from its exergy if left unattended for 40 years. However, a person will lose much of his or her entire exergy during that time period if he or she just sits back.

A hard-working farmer, for example, may make full use of his *physical exergy* but very little use of his *intellectual exergy.* That farmer, for example, could learn a foreign language or a science by listening to some educational CDs at the same time he is doing his physical work. This is also true for people who spend considerable time in the car commuting to work. It is hoped that some day we will be able to do exergy analysis for people and their activities. Such an analysis will point out the way for people to minimize their exergy destruction, and get more done in less time. Computers can perform several tasks at once. Why shouldn't human beings be able to do the same?

Children are born with different levels of *exergies* (talents) in different areas. Giving aptitude tests to children at an early age is simply an attempt to uncover the extent of their "hidden" exergies, or talents. The children are then directed to areas in which they have the greatest exergy. As adults, they are more likely to perform at high levels without stretching the limits if they are naturally fit to be in that area.

We can view the level of *alertness* of a person as his or her *exergy* for intellectual affairs. When a person is well-rested, the degree of alertness, and thus intellectual exergy, is at a maximum and this exergy decreases with time as the person gets tired, as illustrated in Fig. 8–49. Different tasks in daily life require different levels of intellectual exergy, and the difference between available and required alertness can be viewed as the *wasted alertness* or *exergy destruction*. To minimize exergy destruction, there should be a close match between available alertness and required alertness.

Consider a well-rested student who is planning to spend her next 4 h studying and watching a 2-h-long movie. From the *first-law* point of view, it makes no difference in what order these tasks are performed. But from the *second-law* point of view, it makes a lot of difference. Of these two tasks, studying requires more intellectual alertness than watching a movie does, and thus it makes thermodynamic sense to study first when the alertness is high and to watch the movie later when the alertness is lower, as shown in the figure. A student who does it backwards wastes a lot of alertness while watching the movie, as illustrated in Fig. 8–49, and she has to keep going back and forth while studying because of insufficient alertness, thus getting less done in the same time period.

(a) Studying first   (b) Watching a movie first
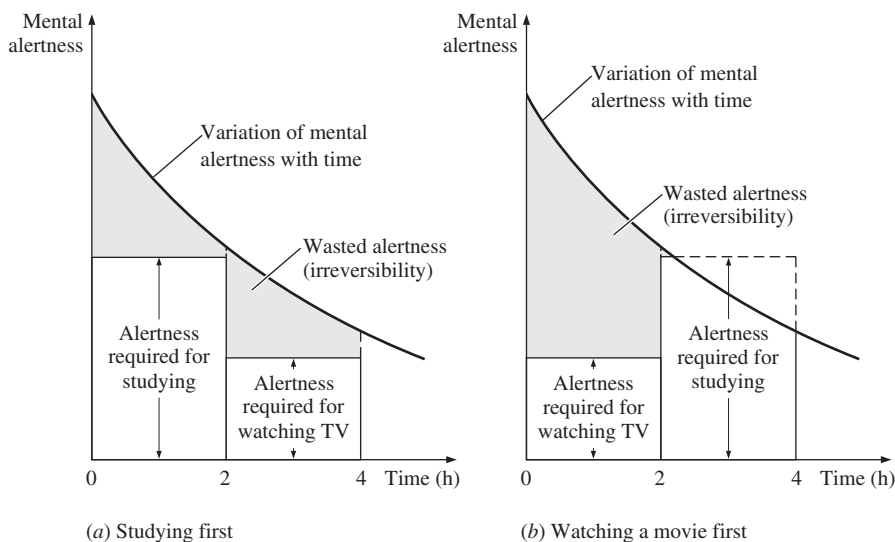
FIGURE 8–49

The irreversibility associated with a student studying and watching a movie on television, each for two hours.

In thermodynamics, *the first-law efficiency* (or thermal efficiency) of a heat engine is defined as the ratio of net work output to total heat input. That is, it is the fraction of the heat supplied that is converted to net work. In general, the first-law efficiency can be viewed as the ratio of the desired output to the required input. The first-law efficiency makes no reference to the *best possible performance,* and thus the first-law efficiency alone is not a realistic measure of performance. To overcome this deficiency, we defined the second-law efficiency, which is a measure of actual performance relative to the best possible performance under the same conditions. For heat engines, the second-law efficiency is defined as the ratio of the actual thermal efficiency to the maximum possible (reversible) thermal efficiency under the same conditions.
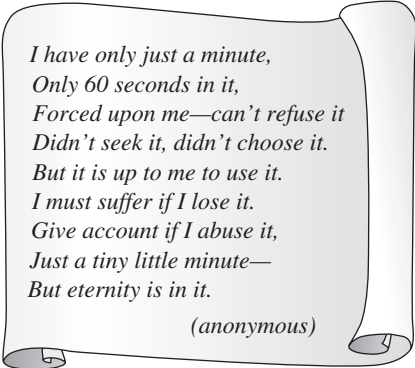
In daily life, the *first-law efficiency* or *performance* of a person can be viewed as the accomplishment of that person relative to the effort he or she puts in. The *second-law efficiency* of a person, on the other hand, can be viewed as the performance of that person relative to the best possible performance under the circumstances.

Happiness is closely related to the *second-law efficiency.* Small children are probably the happiest human beings because there is so little they can do, but they do it so well, considering their limited capabilities. That is, children have very high second-law efficiencies in their daily lives. The term "full life" also refers to second-law efficiency. A person is considered to have a full life, and thus a very high second-law efficiency, if he or she has utilized all of his or her abilities to the limit during a lifetime.

Even a person with some disabilities has to put in considerably more effort to accomplish what a physically fit person accomplishes. Yet, despite accomplishing less with more effort, the person with disabilities who gives an impressive performance often gets more praise. Thus we can say that this person with disabilities had a low first-law efficiency (accomplishing little with a lot of effort) but a very high second-law efficiency (accomplishing as much as possible under the circumstances).

In daily life, exergy can also be viewed as the *opportunities that we have* and the exergy destruction as the *opportunities wasted.* Time is the biggest asset that we have, and the time wasted is the wasted opportunity to do something useful (Fig. 8–50).

The examples above show that several *parallels* can be drawn between the supposedly abstract concepts of thermodynamics related to the second law and daily life, and that the second-law concepts can be used in daily life as frequently and authoritatively as the first-law concepts. Relating the *abstract concepts* of thermodynamics to *ordinary events* of life benefits both engineers and social scientists: it helps *engineers* to have a clearer picture of those concepts and to understand them better, and it enables *social scientists* to use these concepts to describe and formulate some social or psychological phenomena better and with more precision. This is like mathematics and sciences being used in support of each other: abstract mathematical concepts are best understood using examples from sciences, and scientific phenomena are best described and formulated with the help of mathematics.

I have only just a minute,
Only 60 seconds in it,
Forced upon me—can't refuse it
Didn't seek it, didn't choose it.
But it is up to me to use it.
I must suffer if I lose it.
Give account if I abuse it,
Just a tiny little minute—
But eternity is in it.

(anonymous)

FIGURE 8–50

A poetic expression of exergy and exergy destruction.

The arguments presented here are exploratory in nature, and they are hoped to initiate some interesting discussions and research that may lead into better understanding of performance in various aspects of daily life. The second law may eventually be used to determine quantitatively the most effective way to improve the quality of life and performance in daily life, as it is presently used to improve the performance of engineering systems.

SUMMARY

The energy content of the universe is constant, just as its mass content is. Yet at times of crisis we are bombarded with speeches and articles on how to "conserve" energy. As engineers, we know that energy is already conserved. What is not conserved is *exergy,* which is the useful work potential of the energy. Once the exergy is wasted, it can never be recovered. When we use energy (to heat our homes for example), we are not destroying any energy; we are merely converting it to a less useful form, a form of less exergy.

The useful work potential of a system at the specified state is called *exergy.* Exergy is a property and is associated with the state of the system and the environment. A system that is in equilibrium with its surroundings has zero exergy and is said to be at the *dead state.* The exergy of heat supplied by thermal energy reservoirs is equivalent to the work output of a Carnot heat engine operating between the reservoir and the environment.

Reversible work W_{rev} is defined as the maximum amount of useful work that can be produced (or the minimum work that needs to be supplied) as a system undergoes a process between the specified initial and final states. This is the useful work output (or input) obtained when the process between the initial and final states is executed in a totally reversible manner. The difference between the reversible work W_{rev} and the useful work W_u is due to the irreversibilities present during the process and is called the *irreversibility I.* It is equivalent to the *exergy destroyed* and is expressed as

$$I = X_{destroyed} = T_0 S_{gen} = W_{rev,out} - W_{u,out} = W_{u,in} - W_{rev,in}$$

where S_{gen} is the entropy generated during the process. For a totally reversible process, the useful and reversible work terms are identical and thus exergy destruction is zero. Exergy destroyed represents the lost work potential and is also called the *wasted work* or *lost work.*

The *second-law efficiency* is a measure of the performance of a device relative to the performance under reversible conditions for the same end states and is given by

$$\eta_{II} = \frac{\eta_{th}}{\eta_{th,rev}} = \frac{W_u}{W_{rev}}$$

for heat engines and other work-producing devices and

$$\eta_{II} = \frac{COP}{COP_{rev}} = \frac{W_{rev}}{W_u}$$

for refrigerators, heat pumps, and other work-consuming devices. In general, the second-law efficiency is expressed as

$$\eta_{II} = \frac{\text{Exergy recovered}}{\text{Exergy supplied}} = 1 - \frac{\text{Exergy destroyed}}{\text{exergy supplied}}$$

The exergies of a fixed mass (nonflow exergy) and of a flow stream are expressed as

Nonflow exergy:
$$\phi = (u - u_0) + P_0(v - v_0) - T_0(s - s_0) + \frac{V^2}{2} + gz$$

$$= (e - e_0) + P_0(v - v_0) - T_0(s - s_0)$$

Flow exergy:
$$\psi = (h - h_0) - T_0(s - s_0) + \frac{V^2}{2} + gz$$

Then the *exergy change* of a fixed mass or fluid stream as it undergoes a process from state 1 to state 2 is given by

$$\Delta X = X_2 - X_1 = m(\phi_2 - \phi_1)$$
$$= (E_2 - E_1) + P_0(V_2 - V_1) - T_0(S_2 - S_1)$$
$$= (U_2 - U_1) + P_0(V_2 - V_1) - T_0(S_2 - S_1)$$
$$+ m\frac{V_2^2 - V_1^2}{2} + mg(z_2 - z_1)$$

$$\Delta\psi = \psi_2 - \psi_1 = (h_2 - h_1) - T_0(s_2 - s_1)$$
$$+ \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1)$$

Exergy can be transferred by heat, work, and mass flow, and exergy transfer accompanied by heat, work, and mass transfer are given by

Exergy transfer by heat:
$$X_{heat} = \left(1 - \frac{T_0}{T}\right)Q$$

Exergy transfer by work:
$$X_{\text{work}} = \begin{cases} W - W_{\text{surr}} & \text{(for boundary work)} \\ W & \text{(for other forms of work)} \end{cases}$$

Exergy transfer by mass:
$$X_{\text{mass}} = m\psi$$

The exergy of an isolated system during a process always decreases or, in the limiting case of a reversible process, remains constant. This is known as the *decrease of exergy principle* and is expressed as

$$\Delta X_{\text{isolated}} = (X_2 - X_1)_{\text{isolated}} \le 0$$

Exergy balance for *any system* undergoing *any process* can be expressed as

General:
$$\underbrace{X_{\text{in}} - X_{\text{out}}}_{\substack{\text{Net exergy transfer} \\ \text{by heat, work, and mass}}} - \underbrace{X_{\text{destroyed}}}_{\substack{\text{Exergy} \\ \text{destruction}}} = \underbrace{\Delta X_{\text{system}}}_{\substack{\text{Change} \\ \text{in exergy}}}$$

General, rate form:
$$\underbrace{\dot{X}_{\text{in}} - \dot{X}_{\text{out}}}_{\substack{\text{Rate of net exergy transfer} \\ \text{by heat, work, and mass}}} - \underbrace{\dot{X}_{\text{destroyed}}}_{\substack{\text{Rate of exergy} \\ \text{destruction}}} = \underbrace{dX_{\text{system}}/dt}_{\substack{\text{Rate of change} \\ \text{in exergy}}}$$

General, unit-mass basis:
$$(x_{\text{in}} - x_{\text{out}}) - x_{\text{destroyed}} = \Delta x_{\text{system}}$$

where

$$\dot{X}_{\text{heat}} = (1 - T_0/T)\dot{Q}$$
$$\dot{X}_{\text{work}} = \dot{W}_{\text{useful}}$$
$$\dot{X}_{\text{mass}} = \dot{m}\psi$$

For a *reversible process,* the exergy destruction term $X_{\text{destroyed}}$ drops out. Taking the positive direction of heat transfer to be to the system and the positive direction of work transfer to be from the system, the general exergy balance relations can be expressed more explicitly as

$$\sum \left(1 - \frac{T_0}{T_k}\right) Q_k - [W - P_0(V_2 - V_1)]$$
$$+ \sum_{\text{in}} m\psi - \sum_{\text{out}} m\psi - X_{\text{destroyed}} = X_2 - X_1$$

$$\sum \left(1 - \frac{T_0}{T_k}\right) \dot{Q}_k - \left(\dot{W} - P_0 \frac{dV_{\text{CV}}}{dt}\right)$$
$$+ \sum_{\text{in}} \dot{m}\psi - \sum_{\text{out}} \dot{m}\psi - \dot{X}_{\text{destroyed}} = \frac{dX_{\text{CV}}}{dt}$$

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PROBLEMS*

Exergy, Irreversibility, Reversible Work, and Second-Law Efficiency

8–1C How does reversible work differ from useful work?

8–2C Under what conditions does the reversible work equal irreversibility for a process?

8–3C What final state will maximize the work output of a device?

8–4C Is the exergy of a system different in different environments?

8–5C How does useful work differ from actual work? For what kind of systems are these two identical?

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*Problems designated by a "C" are concept questions, and students are encouraged to answer them all. Problems designated by an "E" are in English units, and the SI users can ignore them. Problems with a CD-EES icon ⊙ are solved using EES, and complete solutions together with parametric studies are included on the enclosed DVD. Problems with a computer-EES icon ⊠ are comprehensive in nature, and are intended to be solved with a computer, preferably using the EES software that accompanies this text.

8–6C Consider a process that involves no irreversibilities. Will the actual useful work for that process be equal to the reversible work?

8–7C Consider two geothermal wells whose energy contents are estimated to be the same. Will the exergies of these wells necessarily be the same? Explain.

8–8C Consider two systems that are at the same pressure as the environment. The first system is at the same temperature as the environment, whereas the second system is at a lower temperature than the environment. How would you compare the exergies of these two systems?

8–9C Consider an environment of zero absolute pressure (such as outer space). How will the actual work and the useful work compare in that environment?

8–10C What is the second-law efficiency? How does it differ from the first-law efficiency?

8–11C Does a power plant that has a higher thermal efficiency necessarily have a higher second-law efficiency than one with a lower thermal efficiency? Explain.

8–12C Does a refrigerator that has a higher COP necessarily have a higher second-law efficiency than one with a lower COP? Explain.

8–13C Can a process for which the reversible work is zero be reversible? Can it be irreversible? Explain.

8–14C Consider a process during which no entropy is generated ($S_{gen} = 0$). Does the exergy destruction for this process have to be zero?

8–15 The electric power needs of a community are to be met by windmills with 10-m-diameter rotors. The windmills are to be located where the wind is blowing steadily at an average velocity of 8 m/s. Determine the minimum number of windmills that need to be installed if the required power output is 600 kW.

8–16 One method of meeting the extra electric power demand at peak periods is to pump some water from a large body of water (such as a lake) to a water reservoir at a higher elevation at times of low demand and to generate electricity at times of high demand by letting this water run down and rotate a turbine (i.e., convert the electric energy to potential energy and then back to electric energy). For an energy storage capacity of 5×10^6 kWh, determine the minimum amount of water that needs to be stored at an average elevation (relative to the ground level) of 75 m. *Answer:* 2.45×10^{10} kg
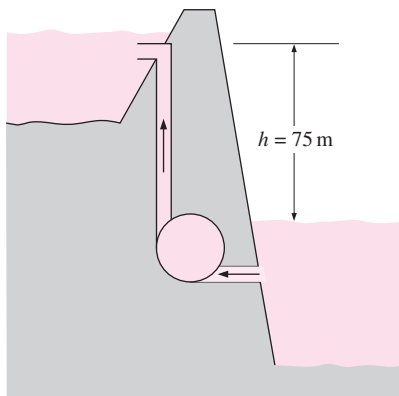
8–17 Consider a thermal energy reservoir at 1500 K that can supply heat at a rate of 150,000 kJ/h. Determine the exergy of this supplied energy, assuming an environmental temperature of 25°C.

8–18 A heat engine receives heat from a source at 1500 K at a rate of 700 kJ/s, and it rejects the waste heat to a medium at 320 K. The measured power output of the heat engine is 320 kW, and the environment temperature is 25°C. Determine (*a*) the reversible power, (*b*) the rate of irreversibility, and (*c*) the second-law efficiency of this heat engine. *Answers:* (*a*) 550.7 kW, (*b*) 230.7 kW, (*c*) 58.1 percent

8–19 Reconsider Prob. 8–18. Using EES (or other) software, study the effect of reducing the temperature at which the waste heat is rejected on the reversible power, the rate of irreversibility, and the second-law efficiency as the rejection temperature is varied from 500 to 298 K, and plot the results.

8–20E A heat engine that rejects waste heat to a sink at 530 R has a thermal efficiency of 36 percent and a second-law efficiency of 60 percent. Determine the temperature of the source that supplies heat to this engine. *Answer:* 1325 R
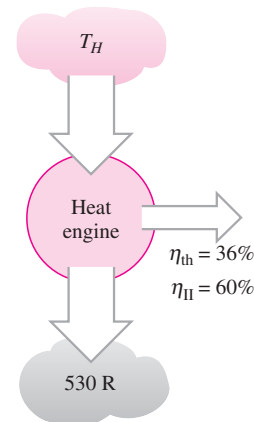

T_H

Heat engine

$\eta_{th} = 36\%$
$\eta_{II} = 60\%$

530 R

FIGURE P8–20E

$h = 75$ m

FIGURE P8–16

8–21 How much of the 100 kJ of thermal energy at 800 K can be converted to useful work? Assume the environment to be at 25°C.

8–22 A heat engine that receives heat from a furnace at 1200°C and rejects waste heat to a river at 20°C has a thermal efficiency of 40 percent. Determine the second-law efficiency of this power plant.

8–23 A house that is losing heat at a rate of 80,000 kJ/h when the outside temperature drops to 15°C is to be heated by electric resistance heaters. If the house is to be maintained at 22°C at all times, determine the reversible work input for this process and the irreversibility. *Answers:* 0.53 kW, 21.69 kW

8–24E A freezer is maintained at 20°F by removing heat from it at a rate of 75 Btu/min. The power input to the freezer is 0.70 hp, and the surrounding air is at 75°F. Determine (*a*) the reversible power, (*b*) the irreversibility, and (*c*) the second-law efficiency of this freezer. *Answers:* (*a*) 0.20 hp, (*b*) 0.50 hp, (*c*) 28.9 percent

8–25 Show that the power produced by a wind turbine is proportional to the cube of the wind velocity and to the square of the blade span diameter.

8–26 A geothermal power plant uses geothermal liquid water at 160°C at a rate of 440 kg/s as the heat source, and produces 14 MW of net power in an environment at 25°C. If 18.5 MW of exergy entering the plant with the geothermal water is destructed within the plant, determine (*a*) the exergy of the geothermal water entering the plant, (*b*) the second-law efficiency, and (*c*) the exergy of the heat rejected from the plant.

Exergy Analysis of Closed Systems

8–27C Is a process during which no entropy is generated ($S_{gen} = 0$) necessarily reversible?

8–28C Can a system have a higher second-law efficiency than the first-law efficiency during a process? Give examples.

8–29 A piston–cylinder device initially contains 2 L of air at 100 kPa and 25°C. Air is now compressed to a final state of 600 kPa and 150°C. The useful work input is 1.2 kJ. Assuming

AIR
$V_1 = 2$ L
$P_1 = 100$ kPa
$T_1 = 25°C$
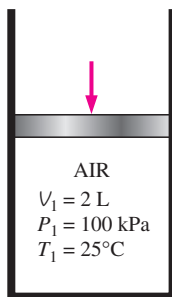
FIGURE P8–29

the surroundings are at 100 kPa and 25°C, determine (*a*) the exergy of the air at the initial and the final states, (*b*) the minimum work that must be supplied to accomplish this compression process, and (*c*) the second-law efficiency of this process. *Answers:* (*a*) 0, 0.171 kJ, (*b*) 0.171 kJ, (*c*) 14.3 percent

8–30 A piston–cylinder device contains 5 kg of refrigerant-134a at 0.7 MPa and 60°C. The refrigerant is now cooled at constant pressure until it exists as a liquid at 24°C. If the surroundings are at 100 kPa and 24°C, determine (*a*) the exergy of the refrigerant at the initial and the final states and (*b*) the exergy destroyed during this process.

8–31 The radiator of a steam heating system has a volume of 20 L and is filled with superheated water vapor at 200 kPa and 200°C. At this moment both the inlet and the exit valves to the radiator are closed. After a while it is observed that the temperature of the steam drops to 80°C as a result of heat transfer to the room air, which is at 21°C. Assuming the surroundings to be at 0°C, determine (*a*) the amount of heat transfer to the room and (*b*) the maximum amount of heat that can be supplied to the room if this heat from the radiator is supplied to a heat engine that is driving a heat pump. Assume the heat engine operates between the radiator and the surroundings. *Answers:* (*a*) 30.3 kJ, (*b*) 116.3 kJ
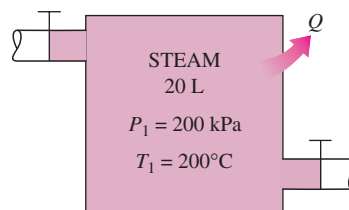

Q

STEAM
20 L

$P_1 = 200$ kPa

$T_1 = 200°C$

FIGURE P8–31

8–32 [EES] Reconsider Prob. 8–31. Using EES (or other) software, investigate the effect of the final steam temperature in the radiator on the amount of actual heat transfer and the maximum amount of heat that can be transferred. Vary the final steam temperature from 80 to 21°C and plot the actual and maximum heat transferred to the room as functions of final steam temperature.

8–33E A well-insulated rigid tank contains 6 lbm of saturated liquid–vapor mixture of water at 35 psia. Initially, three-quarters of the mass is in the liquid phase. An electric resistance heater placed in the tank is turned on and kept on until all the liquid in the tank is vaporized. Assuming the surroundings to be at 75°F and 14.7 psia, determine (*a*) the exergy destruction and (*b*) the second-law efficiency for this process.

8–34 A rigid tank is divided into two equal parts by a partition. One part of the tank contains 1.5 kg of compressed liquid water at 300 kPa and 60°C and the other side is evacuated.

Now the partition is removed, and the water expands to fill the entire tank. If the final pressure in the tank is 15 kPa, determine the exergy destroyed during this process. Assume the surroundings to be at 25°C and 100 kPa.   *Answer: 3.67 kJ*

8–35   Reconsider Prob. 8–34. Using EES (or other) software, study the effect of final pressure in the tank on the exergy destroyed during the process. Plot the exergy destroyed as a function of the final pressure for final pressures between 25 and 15 kPa, and discuss the results.

8–36   An insulated piston–cylinder device contains 2 L of saturated liquid water at a constant pressure of 150 kPa. An electric resistance heater inside the cylinder is turned on, and electrical work is done on the water in the amount of 2200 kJ. Assuming the surroundings to be at 25°C and 100 kPa, determine (*a*) the minimum work with which this process could be accomplished and (*b*) the exergy destroyed during this process.   *Answers: (a) 437.7 kJ, (b) 1705 kJ*
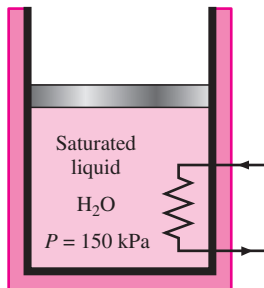
FIGURE P8–36

8–37   Reconsider Prob. 8–36. Using EES (or other) software, investigate the effect of the amount of electrical work supplied to the device on the minimum work and the exergy destroyed as the electrical work is varied from 0 to 2200 kJ, and plot your results.

8–38   An insulated piston–cylinder device contains 0.05 m³ of saturated refrigerant-134a vapor at 0.8 MPa pressure. The refrigerant is now allowed to expand in a reversible manner until the pressure drops to 0.2 MPa. Determine the change in the exergy of the refrigerant during this process and the reversible work. Assume the surroundings to be at 25°C and 100 kPa.

8–39E   Oxygen gas is compressed in a piston–cylinder device from an initial state of 12 ft³/lbm and 75°F to a final state of 1.5 ft³/lbm and 525°F. Determine the reversible work input and the increase in the exergy of the oxygen during this process. Assume the surroundings to be at 14.7 psia and 75°F.   *Answers: 60.7 Btu/lbm, 60.7 Btu/lbm*

8–40   A 1.2-m³ insulated rigid tank contains 2.13 kg of carbon dioxide at 100 kPa. Now paddle-wheel work is done on the system until the pressure in the tank rises to 120 kPa. Determine (*a*) the actual paddle-wheel work done during this process and (*b*) the minimum paddle-wheel work with which this process (between the same end states) could be accomplished. Take $T_0 = 298$ K.   *Answers: (a) 87.0 kJ, (b) 7.74 kJ*
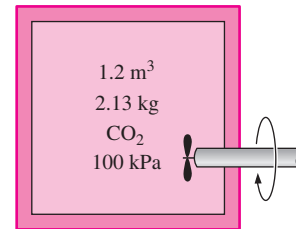

FIGURE P8–40

8–41   An insulated piston–cylinder device initially contains 30 L of air at 120 kPa and 27°C. Air is now heated for 5 min by a 50-W resistance heater placed inside the cylinder. The pressure of air is maintained constant during this process, and the surroundings are at 27°C and 100 kPa. Determine the exergy destroyed during this process.   *Answer: 9.9 kJ*

8–42   A mass of 8 kg of helium undergoes a process from an initial state of 3 m³/kg and 15°C to a final state of 0.5 m³/kg and 80°C. Assuming the surroundings to be at 25°C and 100 kPa, determine the increase in the useful work potential of the helium during this process.

8–43   An insulated rigid tank is divided into two equal parts by a partition. Initially, one part contains 3 kg of argon gas at 300 kPa and 70°C, and the other side is evacuated. The partition is now removed, and the gas fills the entire tank. Assuming the surroundings to be at 25°C, determine the exergy destroyed during this process.   *Answer: 129 kJ*

8–44E   A 70-lbm copper block initially at 250°F is dropped into an insulated tank that contains 1.5 ft³ of water at 75°F. Determine (*a*) the final equilibrium temperature and (*b*) the work potential wasted during this process. Assume the surroundings to be at 75°F.

8–45   An iron block of unknown mass at 85°C is dropped into an insulated tank that contains 100 L of water at 20°C. At the same time, a paddle wheel driven by a 200-W motor is
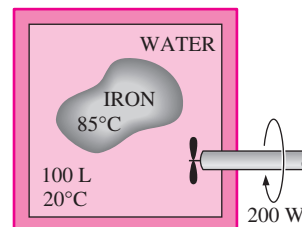

FIGURE P8–45

activated to stir the water. It is observed that thermal equilibrium is established after 20 min with a final temperature of 24°C. Assuming the surroundings to be at 20°C, determine (a) the mass of the iron block and (b) the exergy destroyed during this process. *Answers:* (a) 52.0 kg, (b) 375 kJ

8–46 A 50-kg iron block and a 20-kg copper block, both initially at 80°C, are dropped into a large lake at 15°C. Thermal equilibrium is established after a while as a result of heat transfer between the blocks and the lake water. Assuming the surroundings to be at 20°C, determine the amount of work that could have been produced if the entire process were executed in a reversible manner.

8–47E A 12-ft³ rigid tank contains refrigerant-134a at 40 psia and 55 percent quality. Heat is transferred now to the refrigerant from a source at 120°F until the pressure rises to 60 psia. Assuming the surroundings to be at 75°F, determine (a) the amount of heat transfer between the source and the refrigerant and (b) the exergy destroyed during this process.

8–48 Chickens with an average mass of 2.2 kg and average specific heat of 3.54 kJ/kg · °C are to be cooled by chilled water that enters a continuous-flow-type immersion chiller at 0.5°C and leaves at 2.5°C. Chickens are dropped into the chiller at a uniform temperature of 15°C at a rate of 500 chickens per hour and are cooled to an average temperature of 3°C before they are taken out. The chiller gains heat from the surroundings at a rate of 200 kJ/h. Determine (a) the rate of heat removal from the chicken, in kW, and (b) the rate of exergy destruction during this chilling process. Take $T_0 = 25°C$.

8–49 An ordinary egg can be approximated as a 5.5-cm-diameter sphere. The egg is initially at a uniform temperature of 8°C and is dropped into boiling water at 97°C. Taking the properties of egg to be $\rho = 1020$ kg/m³ and $c_p = 3.32$ kJ/kg · °C, determine how much heat is transferred to the egg by the time the average temperature of the egg rises to 70°C and the amount of exergy destruction associated with this heat transfer process. Take $T_0 = 25°C$.
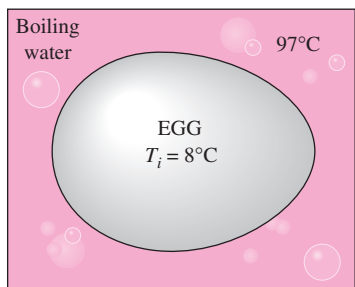
FIGURE P8–49

8–50 Stainless steel ball bearings ($\rho = 8085$ kg/m³ and $c_p = 0.480$ kJ/kg · °C) having a diameter of 1.2 cm are to be quenched in water at a rate of 1400 per minute. The balls leave the oven at a uniform temperature of 900°C and are

exposed to air at 30°C for a while before they are dropped into the water. If the temperature of the balls drops to 850°C prior to quenching, determine (a) the rate of heat transfer from the balls to the air and (b) the rate of exergy destruction due to heat loss from the balls to the air.

8–51 Carbon steel balls ($\rho = 7833$ kg/m³ and $c_p = 0.465$ kJ/kg · °C) 8 mm in diameter are annealed by heating them first to 900°C in a furnace and then allowing them to cool slowly to 100°C in ambient air at 35°C. If 1200 balls are to be annealed per hour, determine (a) the rate of heat transfer from the balls to the air and (b) the rate of exergy destruction due to heat loss from the balls to the air. *Answers:* (a) 260 W, (b) 146 W
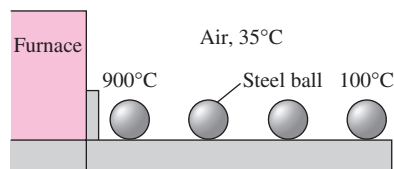

FIGURE P8–51

8–52 A 0.04-m³ tank initially contains air at ambient conditions of 100 kPa and 22°C. Now, a 15-liter tank containing liquid water at 85°C is placed into the tank without causing any air to escape. After some heat transfer from the water to the air and the surroundings, both the air and water are measured to be at 44°C. Determine (a) the amount of heat lost to the surroundings and (b) the exergy destruction during this process.
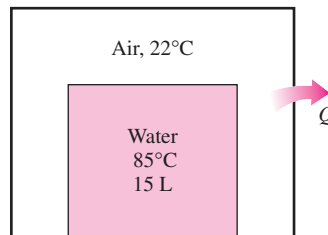

FIGURE P8–52

8–53 A piston–cylinder device initially contains 1.4 kg of refrigerant-134a at 140 kPa and 20°C. Heat is now transferred to the refrigerant, and the piston, which is resting on a set of stops, starts moving when the pressure inside reaches 180 kPa. Heat transfer continues until the temperature reaches 120°C. Assuming the surroundings to be at 25°C and 100 kPa, determine (a) the work done, (b) the heat transfer, (c) the exergy destroyed, and (d) the second-law efficiency of this process. *Answers:* (a) 2.57 kJ, (b) 120 kJ, (c) 13.5 kJ, (d) 0.078

Exergy Analysis of Control Volumes

8–54 Steam is throttled from 8 MPa and 450°C to 6 MPa. Determine the wasted work potential during this throttling
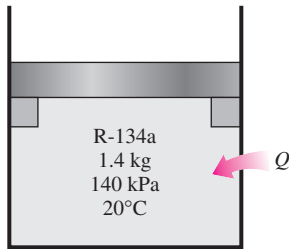
FIGURE P8–53

process. Assume the surroundings to be at 25°C. *Answer:* 36.6 kJ/kg

8–55 Air is compressed steadily by an 8-kW compressor from 100 kPa and 17°C to 600 kPa and 167°C at a rate of 2.1 kg/min. Neglecting the changes in kinetic and potential energies, determine (*a*) the increase in the exergy of the air and (*b*) the rate of exergy destroyed during this process. Assume the surroundings to be at 17°C.
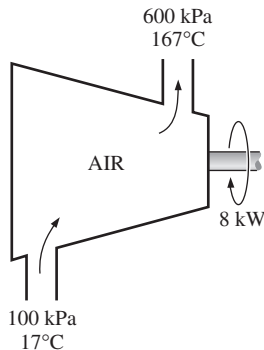

FIGURE P8–55

8–56 Reconsider Prob. 8–55. Using EES (or other) software, solve the problem and in addition determine the actual heat transfer, if any, and its direction, the minimum power input (the reversible power), and the compressor second-law efficiency. Then interpret the results when the outlet temperature is set to, say, 300°C. Explain the values of heat transfer, exergy destroyed, and efficiency when the outlet temperature is set to 209.31°C and mass flow rate to 2.466 kg/min.

8–57 Refrigerant-134a at 1 MPa and 100°C is throttled to a pressure of 0.8 MPa. Determine the reversible work and exergy destroyed during this throttling process. Assume the surroundings to be at 30°C.

8–58 Reconsider Prob. 8–57. Using EES (or other) software, investigate the effect of exit pressure on the reversible work and exergy destruction. Vary the throttle exit pressure from 1 to 0.1 MPa and plot the reversible work and exergy destroyed as functions of the exit pressure. Discuss the results.

8–59 Air enters a nozzle steadily at 300 kPa and 87°C with a velocity of 50 m/s and exits at 95 kPa and 300 m/s. The heat loss from the nozzle to the surrounding medium at 17°C is estimated to be 4 kJ/kg. Determine (*a*) the exit temperature and (*b*) the exergy destroyed during this process. *Answers:* (*a*) 39.5°C, (*b*) 58.4 kJ/kg

8–60 Reconsider Prob. 8–59. Using EES (or other) software, study the effect of varying the nozzle exit velocity from 100 to 300 m/s on both the exit temperature and exergy destroyed, and plot the results.

8–61 Steam enters a diffuser at 10 kPa and 50°C with a velocity of 300 m/s and exits as saturated vapor at 50°C and 70 m/s. The exit area of the diffuser is 3 m². Determine (*a*) the mass flow rate of the steam and (*b*) the wasted work potential during this process. Assume the surroundings to be at 25°C.

8–62E Air is compressed steadily by a compressor from 14.7 psia and 60°F to 100 psia and 480°F at a rate of 22 lbm/min. Assuming the surroundings to be at 60°F, determine the minimum power input to the compressor. Assume air to be an ideal gas with variable specific heats, and neglect the changes in kinetic and potential energies.

8–63 Steam enters an adiabatic turbine at 6 MPa, 600°C, and 80 m/s and leaves at 50 kPa, 100°C, and 140 m/s. If the power output of the turbine is 5 MW, determine (*a*) the reversible power output and (*b*) the second-law efficiency of the turbine. Assume the surroundings to be at 25°C. *Answers:* (*a*) 5.84 MW, (*b*) 85.6 percent
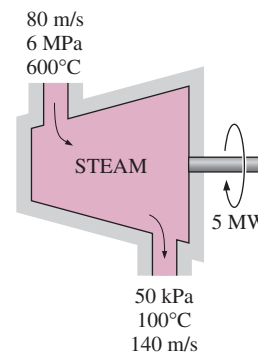

FIGURE P8–63

8–64 Steam is throttled from 9 MPa and 500°C to a pressure of 7 MPa. Determine the decrease in exergy of the steam during this process. Assume the surroundings to be at 25°C. *Answer:* 32.3 kJ/kg

8–65 Combustion gases enter a gas turbine at 900°C, 800 kPa, and 100 m/s and leave at 650°C, 400 kPa, and 220 m/s. Taking $c_p = 1.15$ kJ/kg · °C and $k = 1.3$ for the combustion gases, determine (*a*) the exergy of the combustion gases at the turbine inlet and (*b*) the work output of the turbine under reversible conditions. Assume the surroundings to be at 25°C and 100 kPa. Can this turbine be adiabatic?

8–66E Refrigerant-134a enters an adiabatic compressor as saturated vapor at 30 psia at a rate of 20 ft³/min and exits at 70 psia pressure. If the isentropic efficiency of the compressor is 80 percent, determine (a) the actual power input and (b) the second-law efficiency of the compressor. Assume the surroundings to be at 75°F. *Answers: (a) 2.85 hp, (b) 79.8 percent*

8–67 Refrigerant-134a at 140 kPa and −10°C is compressed by an adiabatic 0.5-kW compressor to an exit state of 700 kPa and 60°C. Neglecting the changes in kinetic and potential energies and assuming the surroundings to be at 27°C, determine (a) the isentropic efficiency and (b) the second-law efficiency of the compressor.

700 kPa
60°C

R-134a

0.5 kW

140 kPa
−10°C
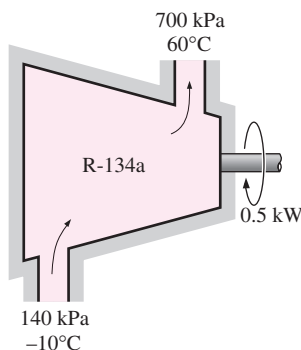
FIGURE P8–67

8–68 Air is compressed by a compressor from 95 kPa and 27°C to 600 kPa and 277°C at a rate of 0.06 kg/s. Neglecting the changes in kinetic and potential energies and assuming the surroundings to be at 25°C, determine the reversible power input for this process. *Answer: 13.7 kW*

8–69 Reconsider Prob. 8–68. Using EES (or other) software, investigate the effect of compressor exit pressure on reversible power. Vary the compressor exit pressure from 200 to 600 kPa while keeping the exit temperature at 277°C. Plot the reversible power input for this process as a function of the compressor exit pressure.

8–70 Argon gas enters an adiabatic compressor at 120 kPa and 30°C with a velocity of 20 m/s and exits at 1.2 MPa, 530°C, and 80 m/s. The inlet area of the compressor is 130 cm². Assuming the surroundings to be at 25°C, determine the reversible power input and exergy destroyed. *Answers: 126 kW, 4.12 kW*

8–71 Steam expands in a turbine steadily at a rate of 15,000 kg/h, entering at 8 MPa and 450°C and leaving at 50 kPa as saturated vapor. Assuming the surroundings to be at 100 kPa and 25°C, determine (a) the power potential of the steam at the inlet conditions and (b) the power output of the turbine if there were no irreversibilities present. *Answers: (a) 5515 kW, (b) 3902 kW*

8–72E Air enters a compressor at ambient conditions of 15 psia and 60°F with a low velocity and exits at 150 psia, 620°F, and 350 ft/s. The compressor is cooled by the ambient

air at 60°F at a rate of 1500 Btu/min. The power input to the compressor is 400 hp. Determine (a) the mass flow rate of air and (b) the portion of the power input that is used just to overcome the irreversibilities.

8–73 Hot combustion gases enter the nozzle of a turbojet engine at 260 kPa, 747°C, and 80 m/s and exit at 70 kPa and 500°C. Assuming the nozzle to be adiabatic and the surroundings to be at 20°C, determine (a) the exit velocity and (b) the decrease in the exergy of the gases. Take $k = 1.3$ and $c_p = 1.15$ kJ/kg · °C for the combustion gases.
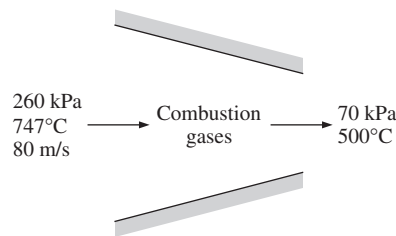

260 kPa
747°C
80 m/s

Combustion
gases

70 kPa
500°C

FIGURE P8–73

8–74 Steam is usually accelerated in the nozzle of a turbine before it strikes the turbine blades. Steam enters an adiabatic nozzle at 7 MPa and 500°C with a velocity of 70 m/s and exits at 5 MPa and 450°C. Assuming the surroundings to be at 25°C, determine (a) the exit velocity of the steam, (b) the isentropic efficiency, and (c) the exergy destroyed within the nozzle.

8–75 Carbon dioxide enters a compressor at 100 kPa and 300 K at a rate of 0.2 kg/s and exits at 600 kPa and 450 K. Determine the power input to the compressor if the process involved no irreversibilities. Assume the surroundings to be at 25°C. *Answer: 25.5 kW*

8–76E A hot-water stream at 160°F enters an adiabatic mixing chamber with a mass flow rate of 4 lbm/s, where it is mixed with a stream of cold water at 70°F. If the mixture leaves the chamber at 110°F, determine (a) the mass flow rate of the cold water and (b) the exergy destroyed during this adiabatic mixing process. Assume all the streams are at a pressure of 50 psia and the surroundings are at 75°F. *Answers: (a) 5.0 lbm/s, (b) 14.6 Btu/s*

8–77 Liquid water at 200 kPa and 20°C is heated in a chamber by mixing it with superheated steam at 200 kPa and

600 kJ/min
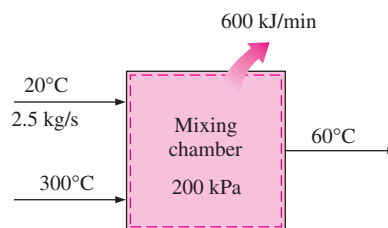
20°C
2.5 kg/s

Mixing
chamber

60°C

300°C

200 kPa

FIGURE P8–77

300°C. Liquid water enters the mixing chamber at a rate of 2.5 kg/s, and the chamber is estimated to lose heat to the surrounding air at 25°C at a rate of 600 kJ/min. If the mixture leaves the mixing chamber at 200 kPa and 60°C, determine (a) the mass flow rate of the superheated steam and (b) the wasted work potential during this mixing process.

8–78 Air enters the evaporator section of a window air conditioner at 100 kPa and 27°C with a volume flow rate of 6 m³/min. Refrigerant-134a at 120 kPa with a quality of 0.3 enters the evaporator at a rate of 2 kg/min and leaves as saturated vapor at the same pressure. Determine the exit temperature of the air and the exergy destruction for this process, assuming (a) the outer surfaces of the air conditioner are insulated and (b) heat is transferred to the evaporator of the air conditioner from the surrounding medium at 32°C at a rate of 30 kJ/min.

8–79 A 0.1-m³ rigid tank initially contains refrigerant-134a at 1.2 MPa and 100 percent quality. The tank is connected by a valve to a supply line that carries refrigerant-134a at 1.6 MPa and 30°C. The valve is now opened, allowing the refrigerant to enter the tank, and it is closed when the tank contains only saturated liquid at 1.4 MPa. The refrigerant exchanges heat with its surroundings at 45°C and 100 kPa during this process. Determine (a) the mass of the refrigerant that entered the tank and (b) the exergy destroyed during this process.

8–80 A 0.6-m³ rigid tank is filled with saturated liquid water at 170°C. A valve at the bottom of the tank is now opened, and one-half of the total mass is withdrawn from the tank in liquid form. Heat is transferred to water from a source of 210°C so that the temperature in the tank remains constant. Determine (a) the amount of heat transfer and (b) the reversible work and exergy destruction for this process. Assume the surroundings to be at 25°C and 100 kPa.
Answers: (a) 2545 kJ, *(b)* 141.2 kJ, 141.2 kJ

8–81E An insulated 150-ft³ rigid tank contains air at 75 psia and 140°F. A valve connected to the tank is opened, and air is allowed to escape until the pressure inside drops to 30 psia. The air temperature during this process is maintained constant by an electric resistance heater placed in the tank. Determine (a) the electrical work done during this process and (b) the exergy destruction. Assume the surroundings to be at 70°F. *Answers: (a)* 1249 Btu, *(b)* 1068 Btu

8–82 A 0.1-m³ rigid tank contains saturated refrigerant-134a at 800 kPa. Initially, 30 percent of the volume is occupied by liquid and the rest by vapor. A valve at the bottom of the tank is opened, and liquid is withdrawn from the tank. Heat is transferred to the refrigerant from a source at 60°C so that the pressure inside the tank remains constant. The valve is closed when no liquid is left in the tank and vapor starts to come out. Assuming the surroundings to be at 25°C, determine (a) the final mass in the tank and (b) the reversible

work associated with this process. *Answers: (a)* 3.90 kg, *(b)* 16.9 kJ

8–83 A vertical piston–cylinder device initially contains 0.1 m³ of helium at 20°C. The mass of the piston is such that it maintains a constant pressure of 300 kPa inside. A valve is now opened, and helium is allowed to escape until the volume inside the cylinder is decreased by one-half. Heat transfer takes place between the helium and its surroundings at 20°C and 95 kPa so that the temperature of helium in the cylinder remains constant. Determine (a) the maximum work potential of the helium at the initial state and (b) the exergy destroyed during this process.
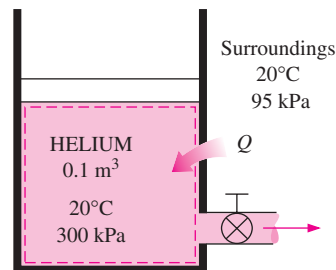

FIGURE P8–83

8–84 A 0.2-m³ rigid tank initially contains saturated refrigerant-134a vapor at 1 MPa. The tank is connected by a valve to a supply line that carries refrigerant-134a at 1.4 MPa and 60°C. The valve is now opened, and the refrigerant is allowed to enter the tank. The valve is closed when one-half of the volume of the tank is filled with liquid and the rest with vapor at 1.2 MPa. The refrigerant exchanges heat during this process with the surroundings at 25°C. Determine (a) the amount of heat transfer and (b) the exergy destruction associated with this process.

8–85 An insulated vertical piston–cylinder device initially contains 15 kg of water, 9 kg of which is in the vapor phase. The mass of the piston is such that it maintains a constant pressure of 200 kPa inside the cylinder. Now steam at 1 MPa and 400°C is allowed to enter the cylinder from a supply line until all the liquid in the cylinder is vaporized. Assuming the surroundings to be at 25°C and 100 kPa, determine (a) the amount of steam that has entered and (b) the exergy destroyed during this process. *Answers: (a)* 23.66 kg, *(b)* 7610 kJ

8–86 Consider a family of four, with each person taking a 6-minute shower every morning. The average flow rate through the shower head is 10 L/min. City water at 15°C is heated to 55°C in an electric water heater and tempered to 42°C by cold water at the T-elbow of the shower before being routed to the shower head. Determine the amount of exergy destroyed by this family per year as a result of taking daily showers. Take $T_0 = 25°C$.

8–87 Ambient air at 100 kPa and 300 K is compressed isentropically in a steady-flow device to 1 MPa. Determine (a) the work input to the compressor, (b) the exergy of the air at the compressor exit, and (c) the exergy of compressed air after it is cooled to 300 K at 1 MPa pressure.

8–88 Cold water ($c_p = 4.18$ kJ/kg · °C) leading to a shower enters a well-insulated, thin-walled, double-pipe, counter-flow heat exchanger at 15°C at a rate of 0.25 kg/s and is heated to 45°C by hot water ($c_p = 4.19$ kJ/kg · °C) that enters at 100°C at a rate of 3 kg/s. Determine (a) the rate of heat transfer and (b) the rate of exergy destruction in the heat exchanger. Take $T_0 = 25$°C.
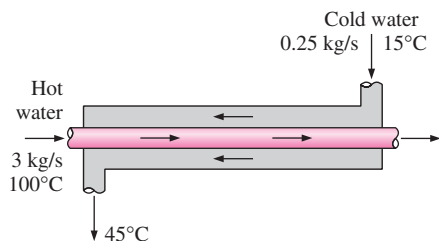
FIGURE P8–88

8–89 Outdoor air ($c_p = 1.005$ kJ/kg · °C) is to be preheated by hot exhaust gases in a cross-flow heat exchanger before it enters the furnace. Air enters the heat exchanger at 95 kPa and 20°C at a rate of 0.8 m³/s. The combustion gases ($c_p = 1.10$ kJ/kg · °C) enter at 180°C at a rate of 1.1 kg/s and leave at 95°C. Determine the rate of heat transfer to the air and the rate of exergy destruction in the heat exchanger.
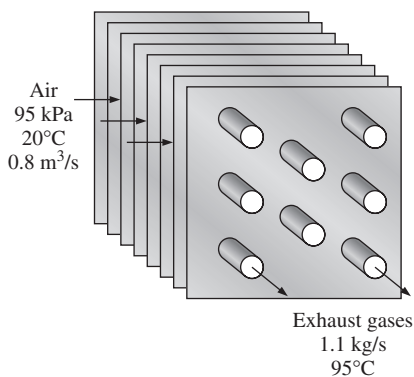

FIGURE P8–89

8–90 A well-insulated shell-and-tube heat exchanger is used to heat water ($c_p = 4.18$ kJ/kg · °C) in the tubes from 20 to 70°C at a rate of 4.5 kg/s. Heat is supplied by hot oil ($c_p = 2.30$ kJ/kg · °C) that enters the shell side at 170°C at a rate of 10 kg/s. Disregarding any heat loss from the heat exchanger,

determine (a) the exit temperature of oil and (b) the rate of exergy destruction in the heat exchanger. Take $T_0 = 25$°C.

8–91E Steam is to be condensed on the shell side of a heat exchanger at 120°F. Cooling water enters the tubes at 60°F at a rate of 115.3 lbm/s and leaves at 73°F. Assuming the heat exchanger to be well-insulated, determine (a) the rate of heat transfer in the heat exchanger and (b) the rate of exergy destruction in the heat exchanger. Take $T_0 = 77$°F.

8–92 Steam enters a turbine at 12 MPa, 550°C, and 60 m/s and leaves at 20 kPa and 130 m/s with a moisture content of 5 percent. The turbine is not adequately insulated and it estimated that heat is lost from the turbine at a rate of 150 kW. The power output of the turbine is 2.5 MW. Assuming the surroundings to be at 25°C, determine (a) the reversible power output of the turbine, (b) the exergy destroyed within the turbine, and (c) the second-law efficiency of the turbine. (d) Also, estimate the possible increase in the power output of the turbine if the turbine were perfectly insulated.
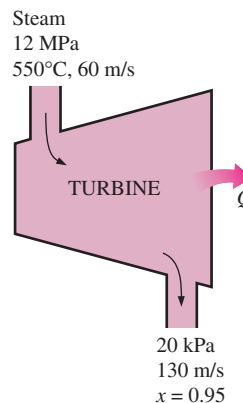

FIGURE P8–92

8–93 Air enters a compressor at ambient conditions of 100 kPa and 20°C at a rate of 4.5 m³/s with a low velocity, and exits at 900 kPa, 60°C, and 80 m/s. The compressor is cooled by cooling water that experiences a temperature rise of 10°C. The isothermal efficiency of the compressor is 70 percent. Determine (a) the actual and reversible power inputs, (b) the second-law efficiency, and (c) the mass flow rate of the cooling water.

8–94 Liquid water at 15°C is heated in a chamber by mixing it with saturated steam. Liquid water enters the chamber at the steam pressure at a rate of 4.6 kg/s and the saturated steam enters at a rate of 0.23 kg/s. The mixture leaves the mixing chamber as a liquid at 45°C. If the surroundings are at 15°C, determine (a) the temperature of saturated steam entering the chamber, (b) the exergy destruction during this mixing process, and (c) the second-law efficiency of the mixing chamber. *Answers:* (a) 114.3°C, (b) 114.7 kW, (c) 0.207
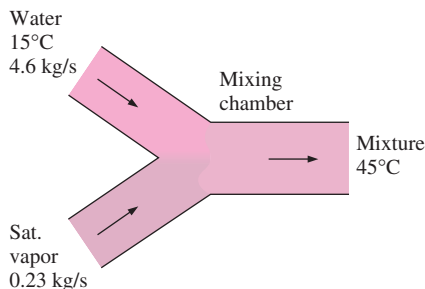
FIGURE P8–94

Review Problems

8–95 Refrigerant-134a is expanded adiabatically in an expansion valve from 1.2 MPa and 40°C to 180 kPa. For environment conditions of 100 kPa and 20°C, determine (*a*) the work potential of R-134a at the inlet, (*b*) the exergy destruction during the process, and (*c*) the second-law efficiency.

8–96 Steam enters an adiabatic nozzle at 3.5 MPa and 300°C with a low velocity and leaves at 1.6 MPa and 250°C at a rate of 0.4 kg/s. If the ambient state is 100 kPa and 18°C, determine (*a*) the exit velocity, (*b*) the rate of exergy destruction, and (*c*) the second-law efficiency.

8–97 A 30-L electrical radiator containing heating oil is placed in a well-sealed 50-m³ room. Both the air in the room and the oil in the radiator are initially at the environment temperature of 10°C. Electricity with a rating of 1.8 kW is now turned on. Heat is also lost from the room at an average rate of 0.35 kW. The heater is turned off after some time when the temperatures of the room air and oil are measured to be 20°C and 50°C, respectively. Taking the density and the specific heat of oil to be 950 kg/m³ and 2.2 kJ/kg · °C, determine (*a*) how long the heater is kept on, (*b*) the exergy destruction, and (*c*) the second-law efficiency for this process. *Answers:* (*a*) 2038 s, (*b*) 3500 kJ, (*c*) 0.046
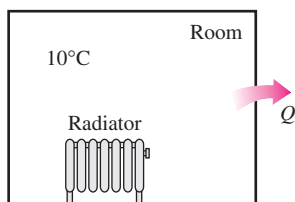

FIGURE P8–97

8–98 Hot exhaust gases leaving an internal combustion engine at 400°C and 150 kPa at a rate of 0.8 kg/s is to be used to produce saturated steam at 200°C in an insulated heat exchanger. Water enters the heat exchanger at the ambient temperature of 20°C, and the exhaust gases leave the heat
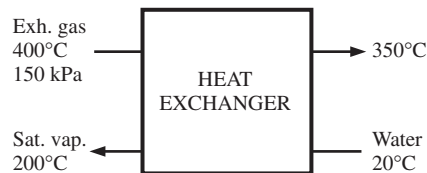

FIGURE P8–98

exchanger at 350°C. Determine (*a*) the rate of steam production, (*b*) the rate of exergy destruction in the heat exchanger, and (*c*) the second-law efficiency of the heat exchanger.

8–99 The inner and outer surfaces of a 5-m × 6-m brick wall of thickness 30 cm are maintained at temperatures of 20°C and 5°C, respectively, and the rate of heat transfer through the wall is 900 W. Determine the rate of exergy destruction associated with this process. Take $T_0 = 0$°C.

FIGURE P8–99
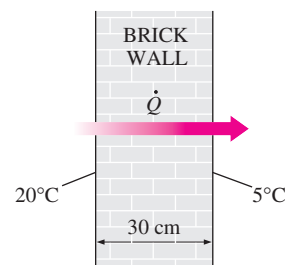
8–100 A 1000-W iron is left on the ironing board with its base exposed to the air at 20°C. If the temperature of the base of the iron is 150°C, determine the rate of exergy destruction for this process due to heat transfer, in steady operation.

8–101 One method of passive solar heating is to stack gallons of liquid water inside the buildings and expose them to the sun. The solar energy stored in the water during the day is released at night to the room air, providing some heating. Consider a house that is maintained at 22°C and whose heating is assisted by a 350-L water storage system. If the water is heated to 45°C during the day, determine the amount of heating this water will provide to the house at night. Assuming an outside temperature of 5°C, determine the exergy destruction associated with this process. *Answers:* 33,548 kJ, 1172 kJ

8–102 The inner and outer surfaces of a 0.5-cm-thick, 2-m × 2-m window glass in winter are 10°C and 3°C, respectively. If the rate of heat loss through the window is 3.2 kJ/s, determine the amount of heat loss, in kJ, through the glass over a period of 5 h. Also, determine the exergy destruction associated with this process. Take $T_0 = 5$°C.

8–103 An aluminum pan has a flat bottom whose diameter is 20 cm. Heat is transferred steadily to boiling water in the pan through its bottom at a rate of 800 W. If the temperatures

of the inner and outer surfaces of the bottom of the pan are 104°C and 105°C, respectively, determine the rate of exergy destruction within the bottom of the pan during this process, in W. Take $T_0 = 25°C$.

8–104 A crater lake has a base area of 20,000 m², and the water it contains is 12 m deep. The ground surrounding the crater is nearly flat and is 140 m below the base of the lake. Determine the maximum amount of electrical work, in kWh, that can be generated by feeding this water to a hydroelectric power plant. *Answer: 95,500 kWh*

8–105E A refrigerator has a second-law efficiency of 45 percent, and heat is removed from the refrigerated space at a rate of 200 Btu/min. If the space is maintained at 35°F while the surrounding air temperature is 75°F, determine the power input to the refrigerator.

8–106 Writing the first- and second-law relations and simplifying, obtain the reversible work relation for a closed system that exchanges heat with the surrounding medium at T_0 in the amount of Q_0 as well as a heat reservoir at T_R in the amount of Q_R. (*Hint:* Eliminate Q_0 between the two equations.)

8–107 Writing the first- and second-law relations and simplifying, obtain the reversible work relation for a steady-flow system that exchanges heat with the surrounding medium at T_0 in the amount of $\dot{Q}_0$ as well as a thermal reservoir at T_R at a rate of $\dot{Q}_R$. (*Hint:* Eliminate $\dot{Q}_0$ between the two equations.)

8–108 Writing the first- and second-law relations and simplifying, obtain the reversible work relation for a uniform-flow system that exchanges heat with the surrounding medium at T_0 in the amount of Q_0 as well as a heat reservoir at T_R in the amount of Q_R. (*Hint:* Eliminate Q_0 between the two equations.)

8–109 A 50-cm-long, 800-W electric resistance heating element whose diameter is 0.5 cm is immersed in 40 kg of water initially at 20°C. Assuming the water container is well-insulated, determine how long it will take for this heater to raise the water temperature to 80°C. Also, determine the minimum work input required and exergy destruction for this process, in kJ. Take $T_0 = 20°C$.
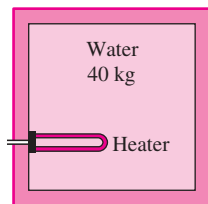

FIGURE P8–109

8–110 A 5-cm-external-diameter, 10-m-long hot water pipe at 80°C is losing heat to the surrounding air at 5°C by natural convection at a rate of 45 W. Determine the rate at which the work potential is wasted during this process as a result of this heat loss.

8–111 Two rigid tanks are connected by a valve. Tank A is insulated and contains 0.2 m³ of steam at 400 kPa and 80 percent quality. Tank B is uninsulated and contains 3 kg of steam at 200 kPa and 250°C. The valve is now opened, and steam flows from tank A to tank B until the pressure in tank A drops to 300 kPa. During this process 900 kJ of heat is transferred from tank B to the surroundings at 0°C. Assuming the steam remaining inside tank A to have undergone a reversible adiabatic process, determine (a) the final temperature in each tank and (b) the work potential wasted during this process.
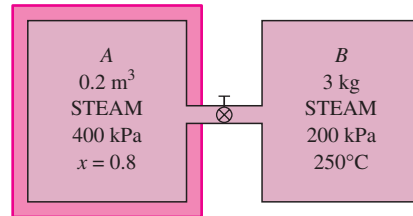

FIGURE P8–111

8–112E A piston–cylinder device initially contains 15 ft³ of helium gas at 25 psia and 70°F. Helium is now compressed in a polytropic process ($PV^n = $ constant) to 70 psia and 300°F. Assuming the surroundings to be at 14.7 psia and 70°F, determine (a) the actual useful work consumed and (b) the minimum useful work input needed for this process. *Answers: (a) 36 Btu, (b) 34.2 Btu*

8–113 A well-insulated 4-m × 4-m × 5-m room initially at 10°C is heated by the radiator of a steam heating system. The radiator has a volume of 15 L and is filled with superheated vapor at 200 kPa and 200°C. At this moment both the inlet and the exit valves to the radiator are closed. A 150-W fan is used to distribute the air in the room. The pressure of the steam is observed to drop to 100 kPa after 30 min as a result of heat transfer to the room. Assuming constant specific heats for air at room temperature, determine (a) the average temperature of room air in 24 min, (b) the entropy change of the steam, (c) the entropy change of the air in the room, and (d) the exergy destruction for this process, in kJ. Assume the air pressure in the room remains constant at 100 kPa at all times, and take $T_0 = 10°C$.

FIGURE P8–113
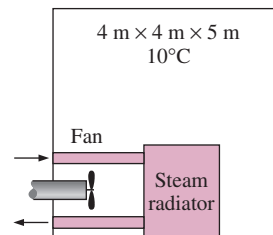
8–114 A passive solar house that is losing heat to the outdoors at 5°C at an average rate of 50,000 kJ/h is maintained at 22°C at all times during a winter night for 10 h. The house is to be heated by 50 glass containers, each containing 20 L of water that is heated to 80°C during the day by absorbing solar energy. A thermostat-controlled 15-kW back-up electric resistance heater turns on whenever necessary to keep the house at 22°C. Determine (a) how long the electric heating system was on that night, (b) the exergy destruction, and (c) the minimum work input required for that night, in kJ.

8–115 Steam at 9 MPa and 500°C enters a two-stage adiabatic turbine at a rate of 15 kg/s. Ten percent of the steam is extracted at the end of the first stage at a pressure of 1.4 MPa for other use. The remainder of the steam is further expanded in the second stage and leaves the turbine at 50 kPa. If the turbine has an isentropic efficiency of 88 percent, determine the wasted power potential during this process as a result of irreversibilities. Assume the surroundings to be at 25°C.

8–116 Steam enters a two-stage adiabatic turbine at 8 MPa and 500°C. It expands in the first stage to a state of 2 MPa and 350°C. Steam is then reheated at constant pressure to a temperature of 500°C before it is routed to the second stage, where it exits at 30 kPa and a quality of 97 percent. The work output of the turbine is 5 MW. Assuming the surroundings to be at 25°C, determine the reversible power output and the rate of exergy destruction within this turbine.
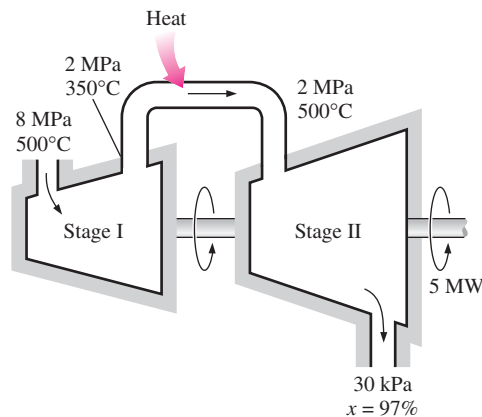*Answers:* 5457 kW, 457 kW

Heat

2 MPa
350°C

2 MPa
500°C

8 MPa
500°C

Stage I   Stage II

5 MW

30 kPa
$x = 97\%$

FIGURE P8–116

8–117 One ton of liquid water at 80°C is brought into a well-insulated and well-sealed 4-m × 5-m × 6-m room initially at 22°C and 100 kPa. Assuming constant specific heats for both the air and water at room temperature, determine (a) the final equilibrium temperature in the room, (b) the exergy destruction, (c) the maximum amount of work that can be produced during this process, in kJ. Take $T_0 = 10°C$.

8–118 Consider a well-insulated horizontal rigid cylinder that is divided into two compartments by a piston that is free to move but does not allow either gas to leak into the other side. Initially, one side of the piston contains 1 m³ of N_2 gas at 500 kPa and 80°C while the other side contains 1 m³ of He gas at 500 kPa and 25°C. Now thermal equilibrium is established in the cylinder as a result of heat transfer through the piston. Using constant specific heats at room temperature, determine (a) the final equilibrium temperature in the cylinder and (b) the wasted work potential during this process. What would your answer be if the piston were not free to move? Take $T_0 = 25°C$.

N_2
1 m³
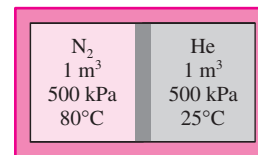500 kPa
80°C

He
1 m³
500 kPa
25°C

FIGURE P8–118

8–119 Repeat Prob. 8–118 by assuming the piston is made of 5 kg of copper initially at the average temperature of the two gases on both sides.

8–120E Argon gas enters an adiabatic turbine at 1500°F and 200 psia at a rate of 40 lbm/min and exhausts at 30 psia. If the power output of the turbine is 95 hp, determine (a) the isentropic efficiency and (b) the second-law efficiency of the turbine. Assume the surroundings to be at 77°F.

8–121 In large steam power plants, the feedwater is frequently heated in closed feedwater heaters, which are basically heat exchangers, by steam extracted from the turbine at some stage. Steam enters the feedwater heater at 1 MPa and 200°C and leaves as saturated liquid at the same pressure. Feedwater enters the heater at 2.5 MPa and 50°C and leaves 10°C below the exit temperature of the
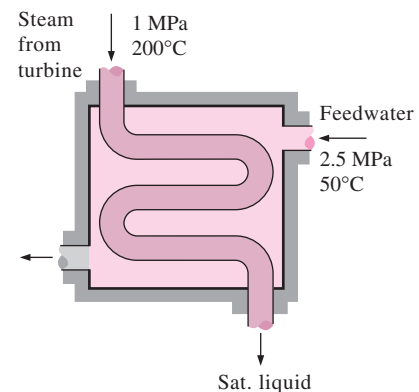
Steam
from
turbine

1 MPa
200°C

Feedwater

2.5 MPa
50°C

Sat. liquid

FIGURE P8–121

steam. Neglecting any heat losses from the outer surfaces of the heater, determine (a) the ratio of the mass flow rates of the extracted steam and the feedwater heater and (b) the reversible work for this process per unit mass of the feedwater. Assume the surroundings to be at 25°C.

Answers: (*a*) 0.247, (*b*) 63.5 kJ/kg

8–122 [EES] Reconsider Prob. 8–121. Using EES (or other) software, investigate the effect of the state of the steam at the inlet of the feedwater heater on the ratio of mass flow rates and the reversible power. Assume the entropy of the extracted steam is constant at the value for 1 MPa, 200°C and decrease the extracted steam pressure from 1 MPa to 100 kPa. Plot both the ratio of the mass flow rates of the extracted steam and the feedwater heater and the reversible work for this process per unit mass of feedwater as functions of the extraction pressure.

8–123 In order to cool 1 ton of water at 20°C in an insulated tank, a person pours 80 kg of ice at −5°C into the water. Determine (a) the final equilibrium temperature in the tank and (b) the exergy destroyed during this process. The melting temperature and the heat of fusion of ice at atmospheric pressure are 0°C and 333.7 kJ/kg, respectively. Take $T_0 = 20$°C.

8–124 Consider a 12-L evacuated rigid bottle that is surrounded by the atmosphere at 100 kPa and 17°C. A valve at the neck of the bottle is now opened and the atmospheric air is allowed to flow into the bottle. The air trapped in the bottle eventually reaches thermal equilibrium with the atmosphere as a result of heat transfer through the wall of the bottle. The valve remains open during the process so that the trapped air also reaches mechanical equilibrium with the atmosphere. Determine the net heat transfer through the wall of the bottle and the exergy destroyed during this filling process.
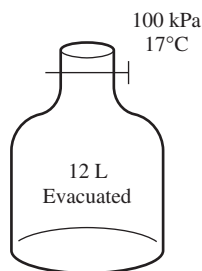

FIGURE P8–124

8–125 Two constant-volume tanks, each filled with 30 kg of air, have temperatures of 900 K and 300 K. A heat engine placed between the two tanks extracts heat from the high-temperature tank, produces work, and rejects heat to the low-temperature tank. Determine the maximum work that can be produced by the heat engine and the final temperatures of the tanks. Assume constant specific heats at room temperature.
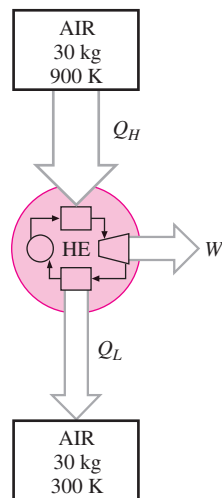

FIGURE P8–125

8–126 Two constant-pressure devices, each filled with 30 kg of air, have temperatures of 900 K and 300 K. A heat engine placed between the two devices extracts heat from the high-temperature device, produces work, and rejects heat to the low-temperature device. Determine the maximum work that can be produced by the heat engine and the final temperatures of the devices. Assume constant specific heats at room temperature.

8–127 A 4-L pressure cooker has an operating pressure of 175 kPa. Initially, one-half of the volume is filled with liquid water and the other half by water vapor. The cooker is now placed on top of a 750-W electrical heating unit that is kept on for 20 min. Assuming the surroundings to be at 25°C and 100 kPa, determine (a) the amount of water that remained in the cooker and (b) the exergy destruction associated with the
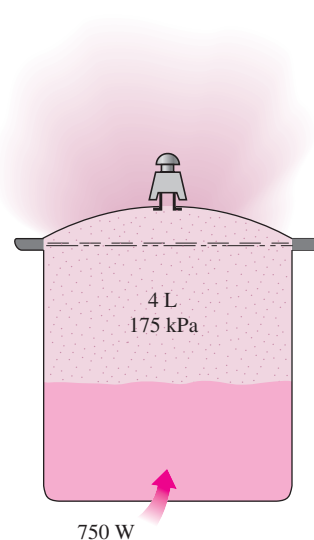

FIGURE P8–127

entire process, including the conversion of electric energy to heat energy.   *Answers:* (*a*) 1.507 kg, (*b*) 689 kJ

8–128   What would your answer to Prob. 8–127 be if heat were supplied to the pressure cooker from a heat source at 180°C instead of the electrical heating unit?

8–129   A constant-volume tank contains 20 kg of nitrogen at 1000 K, and a constant-pressure device contains 10 kg of argon at 300 K. A heat engine placed between the tank and device extracts heat from the high-temperature tank, produces work, and rejects heat to the low-temperature device. Determine the maximum work that can be produced by the heat

FIGURE P8–129

engine and the final temperatures of the nitrogen and argon. Assume constant specific heats at room temperature.

8–130   A constant-volume tank has a temperature of 800 K and a constant-pressure device has a temperature of 290 K. Both the tank and device are filled with 20 kg of air. A heat engine placed between the tank and device receives heat from the high-temperature tank, produces work, and rejects heat to the low-temperature device. Determine the maxi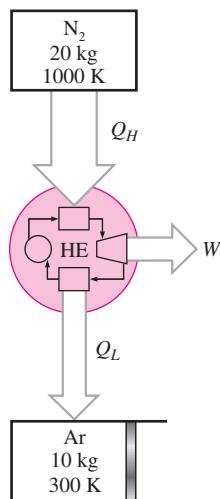mum work that can be produced by the heat engine and the final temperatures of the tank and device. Assume constant specific heats at room temperature.

8–131   Can closed-system exergy be negative? How about flow exergy? Explain using an incompressible substance as an example.

8–132   Obtain a relation for the second-law efficiency of a heat engine that receives heat Q_H from a source at temperature T_H and rejects heat Q_L to a sink at T_L, which is higher than T_0 (the temperature of the surroundings), while producing work in the amount of W.

8–133E   In a production facility, 1.2-in-thick, 2-ft × 2-ft square brass plates (ρ = 532.5 lbm/ft^3 and c_p = 0.091 Btu/lbm · °F) that are initially at a uniform temperature of 75°F are
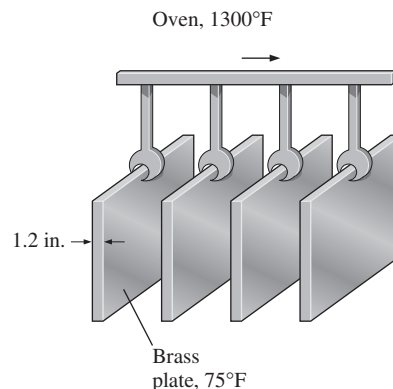

FIGURE P8–133E

heated by passing them through an oven at 1300°F at a rate of 300 per minute. If the plates remain in the oven until their average temperature rises to 1000°F, determine the rate of heat transfer to the plates in the furnace and the rate of exergy destruction associated with this heat transfer process.

8–134   Long cylindrical steel rods (ρ = 7833 kg/m^3 and c_p = 0.465 kJ/kg · °C) of 10-cm diameter are heat-treated by drawing them at a velocity of 3 m/min through a 6-m-long oven maintained at 900°C. If the rods enter the oven at 30°C and leave at 700°C, determine (*a*) the rate of heat transfer to the rods in the oven and (*b*) the rate of exergy destruction associated with this heat transfer process. Take T_0 = 25°C.

8–135   Steam is to be condensed in the condenser of a steam power plant at a temperature of 60°C with cooling water from a nearby lake that enters the tubes of the condenser at 15°C at a rate of 140 kg/s and leaves at 25°C. Assuming the condenser to be perfectly insulated, determine (*a*) the rate of condensation of the steam and (*b*) the rate of exergy destruction in the condenser.   *Answers:* (*a*) 2.48 kg, (*b*) 694 kW

8–136   A well-insulated heat exchanger is to heat water (c_p = 4.18 kJ/kg · °C) from 25°C to 60°C at a rate of 0.4 kg/s. The heating is to be accomplished by geothermal water (c_p = 4.31 kJ/kg · °C) available at 140°C at a mass flow rate of 0.3 kg/s. The inner tube is thin-walled and has a diameter of 0.6 cm. Determine (*a*) the rate of heat transfer and (*b*) the rate of exergy destruction in the heat exchanger.
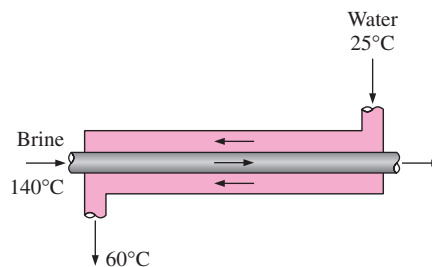

FIGURE P8–136

8–137 An adiabatic heat exchanger is to cool ethylene glycol (c_p = 2.56 kJ/kg · °C) flowing at a rate of 2 kg/s from 80 to 40°C by water (c_p = 4.18 kJ/kg · °C) that enters at 20°C and leaves at 55°C. Determine (*a*) the rate of heat transfer and (*b*) the rate of exergy destruction in the heat exchanger.

8–138 A well-insulated, thin-walled, counter-flow heat exchanger is to be used to cool oil (c_p = 2.20 kJ/kg · °C) from 150 to 40°C at a rate of 2 kg/s by water (c_p = 4.18 kJ/kg · °C) that enters at 22°C at a rate of 1.5 kg/s. The diameter of the tube is 2.5 cm, and its length is 6 m. Determine (*a*) the rate of heat transfer and (*b*) the rate of exergy destruction in the heat exchanger.
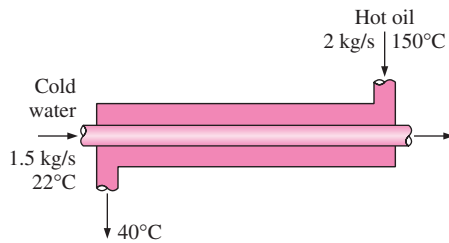

Hot oil
2 kg/s | 150°C

Cold
water

1.5 kg/s
22°C

40°C

FIGURE P8–138

8–139 In a dairy plant, milk at 4°C is pasteurized continuously at 72°C at a rate of 12 L/s for 24 h/day and 365 days/yr. The milk is heated to the pasteurizing temperature by hot water heated in a natural gas-fired boiler having an efficiency of 82 percent. The pasteurized milk is then cooled by cold water at 18°C before it is finally refrigerated back to 4°C. To save energy and money, the plant installs a regenerator that has an effectiveness of 82 percent. If the cost of natural gas is $1.04/therm (1 therm = 105,500 kJ), determine how much energy and money the regenerator will save this company per year and the annual reduction in exergy destruction.

8–140 Combustion gases enter a gas turbine at 750°C and 1.2 MPa at a rate of 3.4 kg/s and leave at 630°C and 500 kPa. It is estimated that heat is lost from the turbine at a rate of
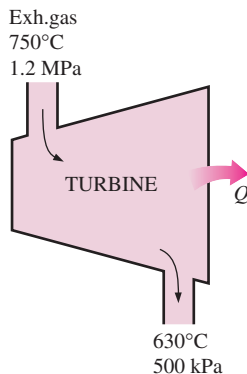

Exh.gas
750°C
1.2 MPa

TURBINE

Q

630°C
500 kPa

FIGURE P8–140

30 kW. Using air properties for the combustion gases and assuming the surroundings to be at 25°C and 100 kPa, determine (*a*) the actual and reversible power outputs of the turbine, (*b*) the exergy destroyed within the turbine, and (*c*) the second-law efficiency of the turbine.

8–141 Refrigerant-134a enters an adiabatic compressor at 160 kPa superheated by 3°C, and leaves at 1.0 MPa. If the compressor has a second-law efficiency of 80 percent, determine (*a*) the actual work input, (*b*) the isentropic efficiency, and (*c*) the exergy destruction. Take the environment temperature to be 25°C. *Answers:* (*a*) 49.8 kJ/kg, (*b*) 0.78, (*c*) 9.95 kJ/kg

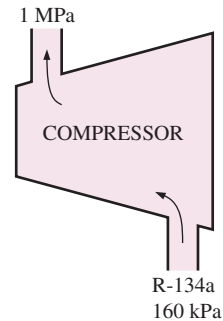1 MPa

COMPRESSOR

R-134a
160 kPa

FIGURE P8–141

8–142 Water enters a pump at 100 kPa and 30°C at a rate of 1.35 kg/s, and leaves at 4 MPa. If the pump has an isentropic efficiency of 70 percent, determine (*a*) the actual power input, (*b*) the rate of frictional heating, (*c*) the exergy destruction, and (*d*) the second-law efficiency for an environment temperature of 20°C.

8–143 Argon gas expands from 3.5 MPa and 100°C to 500 kPa in an adiabatic expansion valve. For environment conditions of 100 kPa and 25°C, determine (*a*) the exergy of argon at the inlet, (*b*) the exergy destruction during the process, and (*c*) the second-law efficiency.

Argon
3.5 MPa
100°C
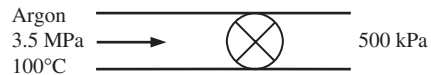
500 kPa

FIGURE P8–143

8–144 Nitrogen gas enters a diffuser at 100 kPa and 150°C with a velocity of 180 m/s, and leaves at 110 kPa and 25 m/s. It is estimated that 4.5 kJ/kg of heat is lost from the diffuser to the surroundings at 100 kPa and 27°C. The exit area of the diffuser is 0.06 m^2. Accounting for the variation of the specific heats with temperature, determine (*a*) the exit temperature, (*b*) the rate of exergy destruction, and (*c*) the second-law efficiency of the diffuser. *Answers:* (*a*) 161°C, (*b*) 5.11 kW, (*c*) 0.892

8–145   Heat is lost through a plane wall steadily at a rate of 800 W. If the inner and outer surface temperatures of the wall are 20°C and 5°C, respectively, and the environment temperature is 0°C, the rate of exergy destruction within the wall is

(*a*) 40 W   (*b*) 17,500 W   (*c*) 765 W
(*d*) 32,800 W   (*e*) 0 W

8–146   Liquid water enters an adiabatic piping system at 15°C at a rate of 5 kg/s. It is observed that the water temperature rises by 0.5°C in the pipe due to friction. If the environment temperature is also 15°C, the rate of exergy destruction in the pipe is

(*a*) 8.36 kW   (*b*) 10.4 kW   (*c*) 197 kW
(*d*) 265 kW   (*e*) 2410 kW

8–147   A heat engine receives heat from a source at 1500 K at a rate of 600 kJ/s and rejects the waste heat to a sink at 300 K. If the power output of the engine is 400 kW, the second-law efficiency of this heat engine is

(*a*) 42%   (*b*) 53%   (*c*) 83%
(*d*) 67%   (*e*) 80%

8–148   A water reservoir contains 100 tons of water at an average elevation of 60 m. The maximum amount of electric power that can be generated from this water is

(*a*) 8 kWh   (*b*) 16 kWh   (*c*) 1630 kWh
(*d*) 16,300 kWh   (*e*) 58,800 kWh

8–149   A house is maintained at 25°C in winter by electric resistance heaters. If the outdoor temperature is 2°C, the second-law efficiency of the resistance heaters is

(*a*) 0%   (*b*) 7.7%   (*c*) 8.7%
(*d*) 13%   (*e*) 100%

8–150   A 12-kg solid whose specific heat is 2.8 kJ/kg · °C is at a uniform temperature of −10°C. For an environment temperature of 20°C, the exergy content of this solid is

(*a*) Less than zero   (*b*) 0 kJ   (*c*) 4.6 kJ
(*d*) 55 kJ   (*e*) 1008 kJ

8–151   Keeping the limitations imposed by the second law of thermodynamics in mind, choose the *wrong* statement below:

(*a*) A heat engine cannot have a thermal efficiency of 100%.

(*b*) For all reversible processes, the second-law efficiency is 100%.

(*c*) The second-law efficiency of a heat engine cannot be greater than its thermal efficiency.

(*d*) The second-law efficiency of a process is 100% if no entropy is generated during that process.

(*e*) The coefficient of performance of a refrigerator can be greater than 1.

8–152   A furnace can supply heat steadily at a 1600 K at a rate of 800 kJ/s. The maximum amount of power that can be produced by using the heat supplied by this furnace in an environment at 300 K is

(*a*) 150 kW   (*b*) 210 kW   (*c*) 325 kW
(*d*) 650 kW   (*e*) 984 kW

8–153   Air is throttled from 50°C and 800 kPa to a pressure of 200 kPa at a rate of 0.5 kg/s in an environment at 25°C. The change in kinetic energy is negligible, and no heat transfer occurs during the process. The power potential wasted during this process is

(*a*) 0   (*b*) 0.20 kW   (*c*) 47 kW
(*d*) 59 kW   (*e*) 119 kW

8–154   Steam enters a turbine steadily at 4 MPa and 400°C and exits at 0.2 MPa and 150°C in an environment at 25°C. The decrease in the exergy of the steam as it flows through the turbine is

(*a*) 58 kJ/kg   (*b*) 445 kJ/kg   (*c*) 458 kJ/kg
(*d*) 518 kJ/kg   (*e*) 597 kJ/kg

Design and Essay Problems

8–155   Obtain the following information about a power plant that is closest to your town: the net power output; the type and amount of fuel used; the power consumed by the pumps, fans, and other auxiliary equipment; stack gas losses; temperatures at several locations; and the rate of heat rejection at the condenser. Using these and other relevant data, determine the rate of irreversibility in that power plant.

8–156   Human beings are probably the most capable creatures, and they have a high level of physical, intellectual, emotional, and spiritual potentials or exergies. Unfortunately people make little use of their exergies, letting most of their exergies go to waste. Draw four exergy versus time charts, and plot your physical, intellectual, emotional, and spiritual exergies on each of these charts for a 24-h period using your best judgment based on your experience. On these four charts, plot your respective exergies that you have utilized during the last 24 h. Compare the two plots on each chart and determine if you are living a "full" life or if you are wasting your life away. Can you think of any ways to reduce the mismatch between your exergies and your utilization of them?

8–157   Consider natural gas, electric resistance, and heat pump heating systems. For a specified heating load, which one of these systems will do the job with the least irreversibility? Explain.

8–158   The domestic hot-water systems involve a high level of irreversibility and thus they have low second-law efficiencies. The water in these systems is heated from about 15°C to about 60°C, and most of the hot water is mixed with cold water to reduce its temperature to 45°C or even lower before it is used for any useful purpose such as taking a shower or washing clothes at a warm setting. The water is discarded at about the same temperature at which it was used and replaced by fresh cold water at 15°C. Redesign a typical residential hot-water system such that the irreversibility is greatly reduced. Draw a sketch of your proposed design.