Chapter 9

GAS POWER CYCLES

Two important areas of application for thermodynamics are power generation and refrigeration. Both are usually accomplished by systems that operate on a thermodynamic cycle. Thermodynamic cycles can be divided into two general categories: *power cycles,* which are discussed in this chapter and Chap. 10, and *refrigeration cycles,* which are discussed in Chap. 11.

The devices or systems used to produce a net power output are often called *engines,* and the thermodynamic cycles they operate on are called *power cycles.* The devices or systems used to produce a refrigeration effect are called *refrigerators, air conditioners,* or *heat pumps,* and the cycles they operate on are called *refrigeration cycles.*

Thermodynamic cycles can also be categorized as *gas cycles* and *vapor cycles,* depending on the *phase* of the working fluid. In gas cycles, the working fluid remains in the gaseous phase throughout the entire cycle, whereas in vapor cycles the working fluid exists in the vapor phase during one part of the cycle and in the liquid phase during another part.

Thermodynamic cycles can be categorized yet another way: *closed* and *open cycles.* In closed cycles, the working fluid is returned to the initial state at the end of the cycle and is recirculated. In open cycles, the working fluid is renewed at the end of each cycle instead of being recirculated. In automobile engines, the combustion gases are exhausted and replaced by fresh air–fuel mixture at the end of each cycle. The engine operates on a mechanical cycle, but the working fluid does not go through a complete thermodynamic cycle.

Heat engines are categorized as *internal combustion* and *external combustion engines,* depending on how the heat is supplied to the working fluid. In external combustion engines (such as steam power plants), heat is supplied to the working fluid from an external source such as a furnace, a geothermal well, a nuclear reactor, or even the sun. In internal combustion engines (such as automobile engines), this is done by burning the fuel within the system boundaries. In this chapter, various gas power cycles are analyzed under some simplifying assumptions.

Objectives

The objectives of Chapter 9 are to:

- Evaluate the performance of gas power cycles for which the working fluid remains a gas throughout the entire cycle.
- Develop simplifying assumptions applicable to gas power cycles.
- Review the operation of reciprocating engines.
- Analyze both closed and open gas power cycles.
- Solve problems based on the Otto, Diesel, Stirling, and Ericsson cycles.
- Solve problems based on the Brayton cycle; the Brayton cycle with regeneration; and the Brayton cycle with intercooling, reheating, and regeneration.
- Analyze jet-propulsion cycles.
- Identify simplifying assumptions for second-law analysis of gas power cycles.
- Perform second-law analysis of gas power cycles.

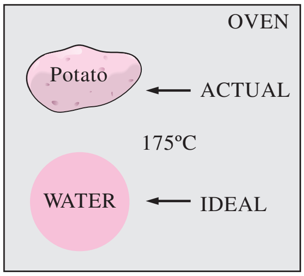


FIGURE 9–1

Modeling is a powerful engineering tool that provides great insight and simplicity at the expense of some loss in accuracy.

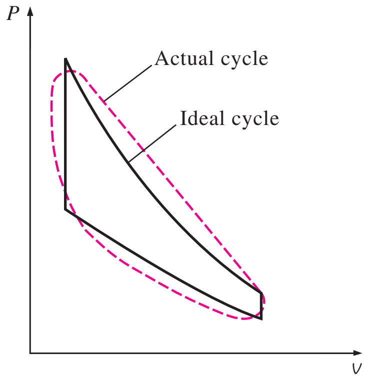


FIGURE 9–2

The analysis of many complex processes can be reduced to a manageable level by utilizing some idealizations.

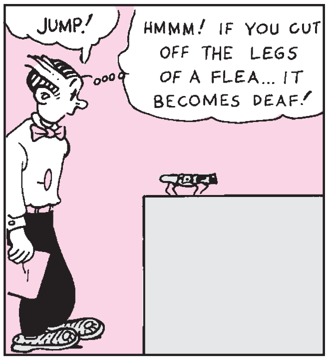


FIGURE 9–3

Care should be exercised in the interpretation of the results from ideal cycles.

9–1 ▪ BASIC CONSIDERATIONS IN THE ANALYSIS OF POWER CYCLES

Most power-producing devices operate on cycles, and the study of power cycles is an exciting and important part of thermodynamics. The cycles encountered in actual devices are difficult to analyze because of the presence of complicating effects, such as friction, and the absence of sufficient time for establishment of the equilibrium conditions during the cycle. To make an analytical study of a cycle feasible, we have to keep the complexities at a manageable level and utilize some idealizations (Fig. 9–1). When the actual cycle is stripped of all the internal irreversibilities and complexities, we end up with a cycle that resembles the actual cycle closely but is made up totally of internally reversible processes. Such a cycle is called an **ideal cycle** (Fig. 9–2).

A simple idealized model enables engineers to study the effects of the major parameters that dominate the cycle without getting bogged down in the details. The cycles discussed in this chapter are somewhat idealized, but they still retain the general characteristics of the actual cycles they represent. The conclusions reached from the analysis of ideal cycles are also applicable to actual cycles. The thermal efficiency of the Otto cycle, the ideal cycle for spark-ignition automobile engines, for example, increases with the compression ratio. This is also the case for actual automobile engines. The numerical values obtained from the analysis of an ideal cycle, however, are not necessarily representative of the actual cycles, and care should be exercised in their interpretation (Fig. 9–3). The simplified analysis presented in this chapter for various power cycles of practical interest may also serve as the starting point for a more in-depth study.

Heat engines are designed for the purpose of converting thermal energy to work, and their performance is expressed in terms of the **thermal efficiency** η_{th}, which is the ratio of the net work produced by the engine to the total heat input:

$$\eta_{th} = \frac{W_{net}}{Q_{in}} \quad \text{or} \quad \eta_{th} = \frac{w_{net}}{q_{in}} \tag{9–1}$$

Recall that heat engines that operate on a totally reversible cycle, such as the Carnot cycle, have the highest thermal efficiency of all heat engines operating between the same temperature levels. That is, nobody can develop a cycle more efficient than the *Carnot cycle.* Then the following question arises naturally: If the Carnot cycle is the best possible cycle, why do we not use it as the model cycle for all the heat engines instead of bothering with several so-called *ideal* cycles? The answer to this question is hardware-related. Most cycles encountered in practice differ significantly from the Carnot cycle, which makes it unsuitable as a realistic model. Each ideal cycle discussed in this chapter is related to a specific work-producing device and is an *idealized* version of the actual cycle.

The ideal cycles are *internally reversible,* but, unlike the Carnot cycle, they are not necessarily externally reversible. That is, they may involve irreversibilities external to the system such as heat transfer through a finite temperature difference. Therefore, the thermal efficiency of an ideal cycle, in general, is less than that of a totally reversible cycle operating between the

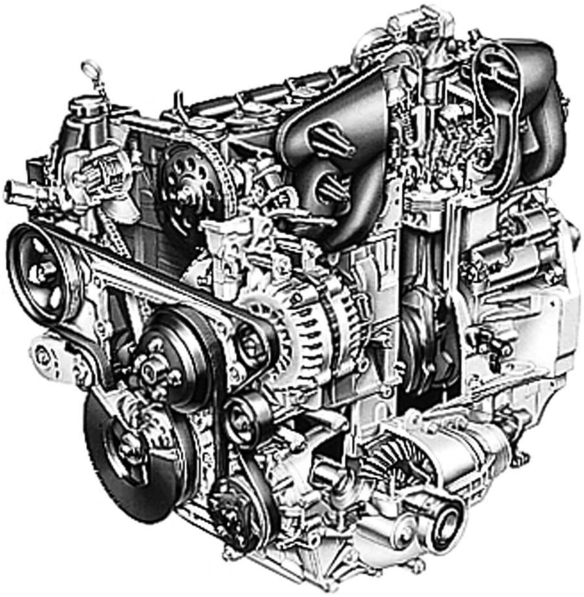


FIGURE 9–4

An automotive engine with the combustion chamber exposed.

Courtesy of General Motors

same temperature limits. However, it is still considerably higher than the thermal efficiency of an actual cycle because of the idealizations utilized (Fig. 9–4).

The idealizations and simplifications commonly employed in the analysis of power cycles can be summarized as follows:

1. The cycle does not involve any *friction.* Therefore, the working fluid does not experience any pressure drop as it flows in pipes or devices such as heat exchangers.
2. All expansion and compression processes take place in a *quasi-equilibrium* manner.
3. The pipes connecting the various components of a system are well insulated, and *heat transfer* through them is negligible.

Neglecting the changes in *kinetic* and *potential energies* of the working fluid is another commonly utilized simplification in the analysis of power cycles. This is a reasonable assumption since in devices that involve shaft work, such as turbines, compressors, and pumps, the kinetic and potential energy terms are usually very small relative to the other terms in the energy equation. Fluid velocities encountered in devices such as condensers, boilers, and mixing chambers are typically low, and the fluid streams experience little change in their velocities, again making kinetic energy changes negligible. The only devices where the changes in kinetic energy are significant are the nozzles and diffusers, which are specifically designed to create large changes in velocity.

In the preceding chapters, *property diagrams* such as the P-v and T-s diagrams have served as valuable aids in the analysis of thermodynamic processes. On both the P-v and T-s diagrams, the area enclosed by the process curves of a cycle represents the net work produced during the cycle (Fig. 9–5), which is also equivalent to the net heat transfer for that cycle.

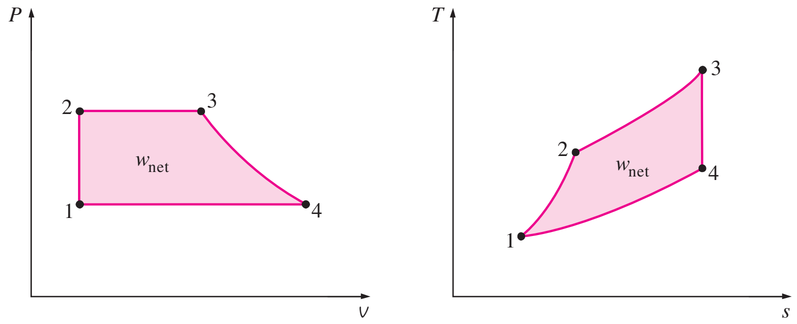


FIGURE 9–5

On both *P*-v and *T*-*s* diagrams, the area enclosed by the process curve represents the net work of the cycle.

The *T*-*s* diagram is particularly useful as a visual aid in the analysis of ideal power cycles. An ideal power cycle does not involve any internal irreversibilities, and so the only effect that can change the entropy of the working fluid during a process is heat transfer.

On a *T*-*s* diagram, a *heat-addition* process proceeds in the direction of increasing entropy, a *heat-rejection* process proceeds in the direction of decreasing entropy, and an *isentropic* (internally reversible, adiabatic) process proceeds at constant entropy. The area under the process curve on a *T*-*s* diagram represents the heat transfer for that process. The area under the heat addition process on a *T*-*s* diagram is a geometric measure of the total heat supplied during the cycle q_{in}, and the area under the heat rejection process is a measure of the total heat rejected q_{out}. The difference between these two (the area enclosed by the cyclic curve) is the net heat transfer, which is also the net work produced during the cycle. Therefore, on a *T*-*s* diagram, the ratio of the area enclosed by the cyclic curve to the area under the heat-addition process curve represents the thermal efficiency of the cycle. *Any modification that increases the ratio of these two areas will also increase the thermal efficiency of the cycle.*

Although the working fluid in an ideal power cycle operates on a closed loop, the type of individual processes that comprises the cycle depends on the individual devices used to execute the cycle. In the Rankine cycle, which is the ideal cycle for steam power plants, the working fluid flows through a series of steady-flow devices such as the turbine and condenser, whereas in the Otto cycle, which is the ideal cycle for the spark-ignition automobile engine, the working fluid is alternately expanded and compressed in a piston–cylinder device. Therefore, equations pertaining to steady-flow systems should be used in the analysis of the Rankine cycle, and equations pertaining to closed systems should be used in the analysis of the Otto cycle.

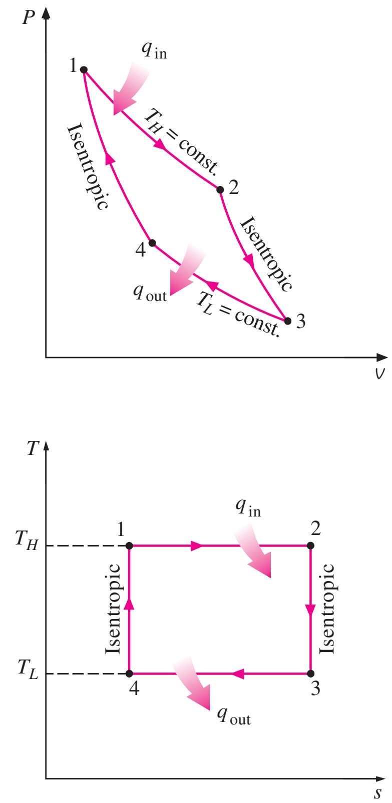


FIGURE 9–6

P-v and *T*-*s* diagrams of a Carnot cycle.

9–2 ▪ THE CARNOT CYCLE AND ITS VALUE IN ENGINEERING

The Carnot cycle is composed of four totally reversible processes: isothermal heat addition, isentropic expansion, isothermal heat rejection, and isentropic compression. The *P*-v and *T*-*s* diagrams of a Carnot cycle are replotted in Fig. 9–6. The Carnot cycle can be executed in a closed system (a piston–cylinder device) or a steady-flow system (utilizing two turbines and two compressors, as shown in Fig. 9–7), and either a gas or a vapor can

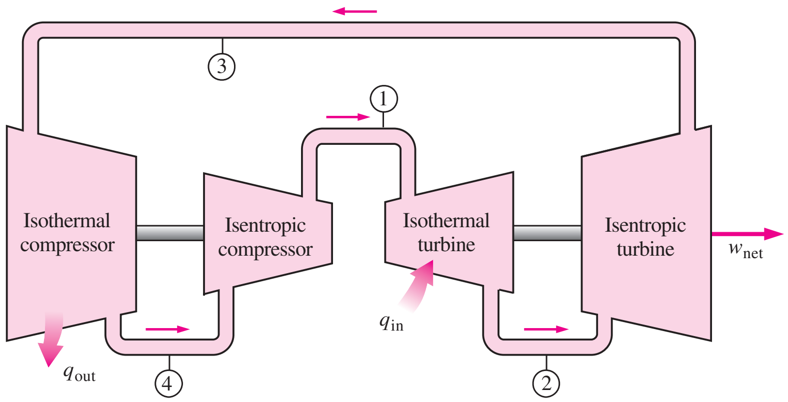


FIGURE 9–7
A steady-flow Carnot engine.

be utilized as the working fluid. The Carnot cycle is the most efficient cycle that can be executed between a heat source at temperature T_H and a sink at temperature T_L, and its thermal efficiency is expressed as

$$\eta_{th,Carnot} = 1 - \frac{T_L}{T_H} \quad \textbf{(9–2)}$$

Reversible isothermal heat transfer is very difficult to achieve in reality because it would require very large heat exchangers and it would take a very long time (a power cycle in a typical engine is completed in a fraction of a second). Therefore, it is not practical to build an engine that would operate on a cycle that closely approximates the Carnot cycle.

The real value of the Carnot cycle comes from its being a standard against which the actual or the ideal cycles can be compared. The thermal efficiency of the Carnot cycle is a function of the sink and source temperatures only, and the thermal efficiency relation for the Carnot cycle (Eq. 9–2) conveys an important message that is equally applicable to both ideal and actual cycles: *Thermal efficiency increases with an increase in the average temperature at which heat is supplied to the system or with a decrease in the average temperature at which heat is rejected from the system.*

The source and sink temperatures that can be used in practice are not without limits, however. The highest temperature in the cycle is limited by the maximum temperature that the components of the heat engine, such as the piston or the turbine blades, can withstand. The lowest temperature is limited by the temperature of the cooling medium utilized in the cycle such as a lake, a river, or the atmospheric air.

EXAMPLE 9–1 Derivation of the Efficiency of the Carnot Cycle

Show that the thermal efficiency of a Carnot cycle operating between the temperature limits of T_H and T_L is solely a function of these two temperatures and is given by Eq. 9–2.

Solution It is to be shown that the efficiency of a Carnot cycle depends on the source and sink temperatures alone.

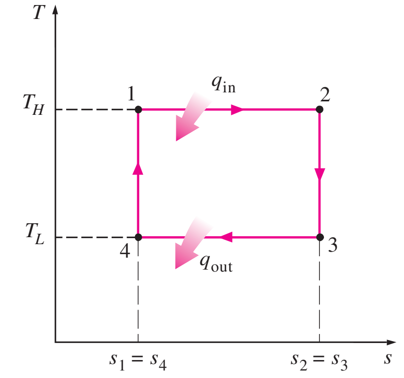


FIGURE 9–8

T-s diagram for Example 9–1.

Analysis The *T-s* diagram of a Carnot cycle is redrawn in Fig. 9–8. All four processes that comprise the Carnot cycle are reversible, and thus the area under each process curve represents the heat transfer for that process. Heat is transferred to the system during process 1-2 and rejected during process 3-4. Therefore, the amount of heat input and heat output for the cycle can be expressed as

$$q_{in} = T_H(s_2 - s_1) \quad \text{and} \quad q_{out} = T_L(s_3 - s_4) = T_L(s_2 - s_1)$$

since processes 2-3 and 4-1 are isentropic, and thus $s_2 = s_3$ and $s_4 = s_1$. Substituting these into Eq. 9–1, we see that the thermal efficiency of a Carnot cycle is

$$\eta_{th} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{T_L(s_2 - s_1)}{T_H(s_2 - s_1)} = 1 - \frac{T_L}{T_H}$$

Discussion Notice that the thermal efficiency of a Carnot cycle is independent of the type of the working fluid used (an ideal gas, steam, etc.) or whether the cycle is executed in a closed or steady-flow system.

9–3 ▪ AIR-STANDARD ASSUMPTIONS

In gas power cycles, the working fluid remains a gas throughout the entire cycle. Spark-ignition engines, diesel engines, and conventional gas turbines are familiar examples of devices that operate on gas cycles. In all these engines, energy is provided by burning a fuel within the system boundaries. That is, they are *internal combustion engines.* Because of this combustion process, the composition of the working fluid changes from air and fuel to combustion products during the course of the cycle. However, considering that air is predominantly nitrogen that undergoes hardly any chemical reactions in the combustion chamber, the working fluid closely resembles air at all times.

Even though internal combustion engines operate on a mechanical cycle (the piston returns to its starting position at the end of each revolution), the working fluid does not undergo a complete thermodynamic cycle. It is thrown out of the engine at some point in the cycle (as exhaust gases) instead of being returned to the initial state. Working on an open cycle is the characteristic of all internal combustion engines.

The actual gas power cycles are rather complex. To reduce the analysis to a manageable level, we utilize the following approximations, commonly known as the **air-standard assumptions:**

1. The working fluid is air, which continuously circulates in a closed loop and always behaves as an ideal gas.
2. All the processes that make up the cycle are internally reversible.
3. The combustion process is replaced by a heat-addition process from an external source (Fig. 9–9).
4. The exhaust process is replaced by a heat-rejection process that restores the working fluid to its initial state.

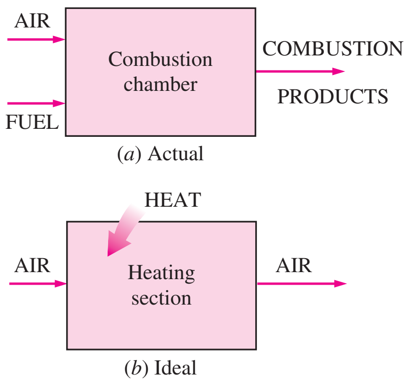


FIGURE 9–9

The combustion process is replaced by a heat-addition process in ideal cycles.

Another assumption that is often utilized to simplify the analysis even more is that air has constant specific heats whose values are determined at *room temperature* (25°C, or 77°F). When this assumption is utilized, the air-standard assumptions are called the **cold-air-standard assumptions.** A cycle for which the air-standard assumptions are applicable is frequently referred to as an **air-standard cycle.**

The air-standard assumptions previously stated provide considerable simplification in the analysis without significantly deviating from the actual cycles. This simplified model enables us to study qualitatively the influence of major parameters on the performance of the actual engines.

9–4 ▪ AN OVERVIEW OF RECIPROCATING ENGINES

Despite its simplicity, the reciprocating engine (basically a piston–cylinder device) is one of the rare inventions that has proved to be very versatile and to have a wide range of applications. It is the powerhouse of the vast majority of automobiles, trucks, light aircraft, ships, and electric power generators, as well as many other devices.

The basic components of a reciprocating engine are shown in Fig. 9–10. The piston reciprocates in the cylinder between two fixed positions called the **top dead center** (TDC)—the position of the piston when it forms the smallest volume in the cylinder—and the **bottom dead center** (BDC)—the position of the piston when it forms the largest volume in the cylinder. The distance between the TDC and the BDC is the largest distance that the piston can travel in one direction, and it is called the **stroke** of the engine. The diameter of the piston is called the **bore.** The air or air–fuel mixture is drawn into the cylinder through the **intake valve,** and the combustion products are expelled from the cylinder through the **exhaust valve.**

The minimum volume formed in the cylinder when the piston is at TDC is called the **clearance volume** (Fig. 9–11). The volume displaced by the piston as it moves between TDC and BDC is called the **displacement volume.** The ratio of the maximum volume formed in the cylinder to the minimum (clearance) volume is called the **compression ratio** r of the engine:

$$r = \frac{V_{max}}{V_{min}} = \frac{V_{BDC}}{V_{TDC}} \tag{9–3}$$

Notice that the compression ratio is a *volume ratio* and should not be confused with the pressure ratio.

Another term frequently used in conjunction with reciprocating engines is the **mean effective pressure** (MEP). It is a fictitious pressure that, if it acted on the piston during the entire power stroke, would produce the same amount of net work as that produced during the actual cycle (Fig. 9–12). That is,

$$W_{net} = \text{MEP} \times \text{Piston area} \times \text{Stroke} = \text{MEP} \times \text{Displacement volume}$$

or

$$\text{MEP} = \frac{W_{net}}{V_{max} - V_{min}} = \frac{w_{net}}{v_{max} - v_{min}} \quad \text{(kPa)} \tag{9–4}$$

The mean effective pressure can be used as a parameter to compare the performances of reciprocating engines of equal size. The engine with a larger value of MEP delivers more net work per cycle and thus performs better.

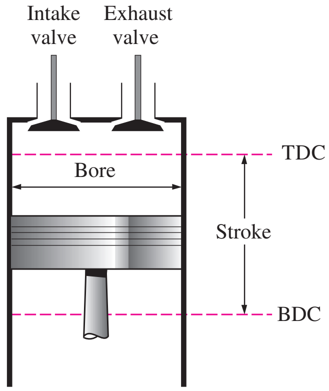
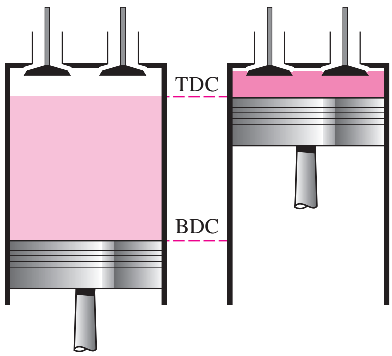


FIGURE 9–10
Nomenclature for reciprocating engines.



(*a*) Displacement volume (*b*) Clearance volume

FIGURE 9–11
Displacement and clearance volumes of a reciprocating engine.

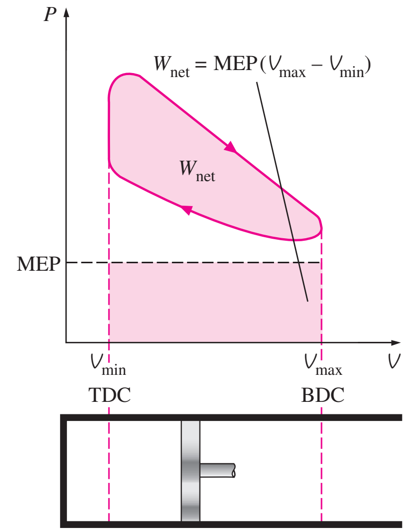


FIGURE 9–12

The net work output of a cycle is equivalent to the product of the mean effective pressure and the displacement volume.

Reciprocating engines are classified as **spark-ignition (SI) engines** or **compression-ignition (CI) engines,** depending on how the combustion process in the cylinder is initiated. In SI engines, the combustion of the air–fuel mixture is initiated by a spark plug. In CI engines, the air–fuel mixture is self-ignited as a result of compressing the mixture above its self-ignition temperature. In the next two sections, we discuss the *Otto* and *Diesel cycles,* which are the ideal cycles for the SI and CI reciprocating engines, respectively.

9–5 ▪ OTTO CYCLE: THE IDEAL CYCLE FOR SPARK-IGNITION ENGINES

The Otto cycle is the ideal cycle for spark-ignition reciprocating engines. It is named after Nikolaus A. Otto, who built a successful four-stroke engine in 1876 in Germany using the cycle proposed by Frenchman Beau de Rochas in 1862. In most spark-ignition engines, the piston executes four complete strokes (two mechanical cycles) within the cylinder, and the crankshaft completes two revolutions for each thermodynamic cycle. These engines are called **four-stroke** internal combustion engines. A schematic of each stroke as well as a *P-v* diagram for an actual four-stroke spark-ignition engine is given in Fig. 9–13(*a*).

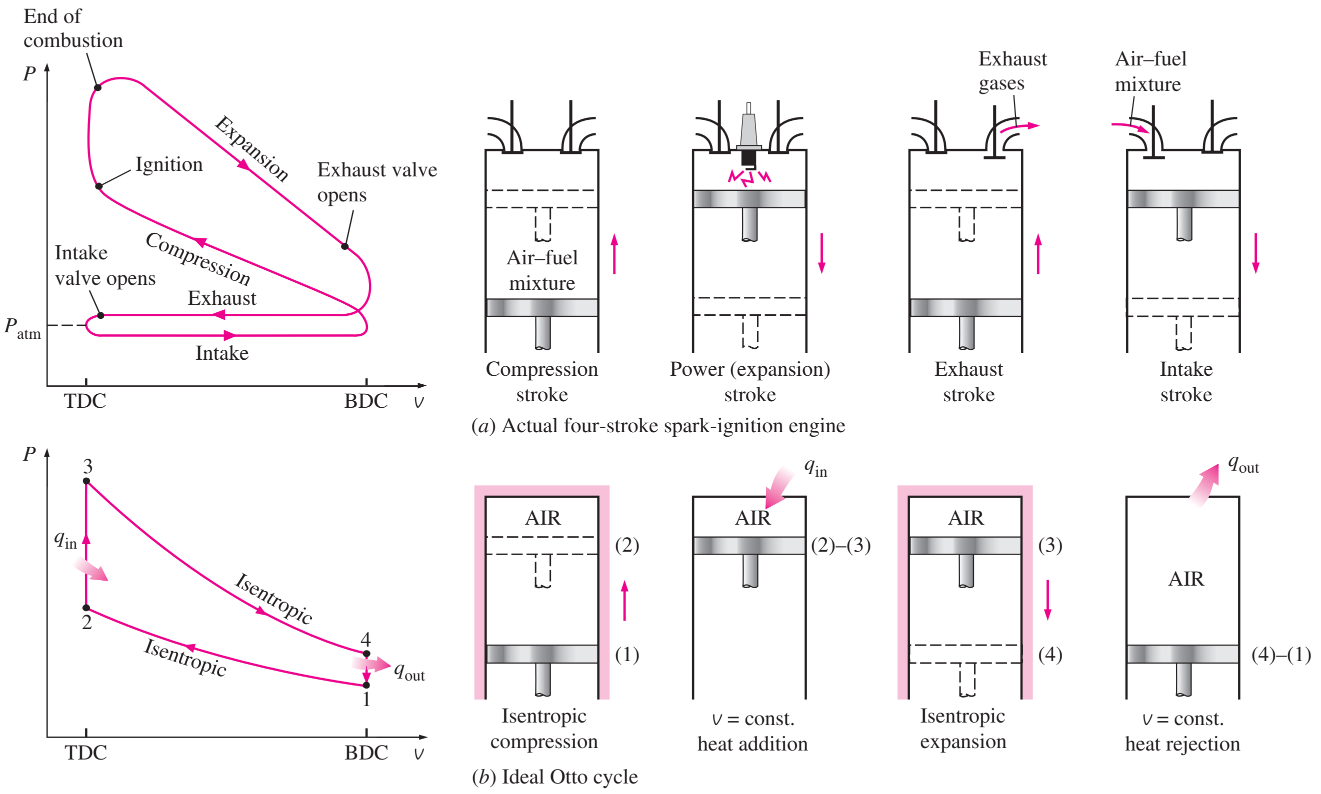


FIGURE 9–13

Actual and ideal cycles in spark-ignition engines and their *P-v* diagrams.

Initially, both the intake and the exhaust valves are closed, and the piston is at its lowest position (BDC). During the *compression stroke,* the piston moves upward, compressing the air–fuel mixture. Shortly before the piston reaches its highest position (TDC), the spark plug fires and the mixture ignites, increasing the pressure and temperature of the system. The high-pressure gases force the piston down, which in turn forces the crankshaft to rotate, producing a useful work output during the *expansion* or *power stroke.* At the end of this stroke, the piston is at its lowest position (the completion of the first mechanical cycle), and the cylinder is filled with combustion products. Now the piston moves upward one more time, purging the exhaust gases through the exhaust valve (the *exhaust stroke*), and down a second time, drawing in fresh air–fuel mixture through the intake valve (the *intake stroke*). Notice that the pressure in the cylinder is slightly above the atmospheric value during the exhaust stroke and slightly below during the intake stroke.

In **two-stroke engines,** all four functions described above are executed in just two strokes: the power stroke and the compression stroke. In these engines, the crankcase is sealed, and the outward motion of the piston is used to slightly pressurize the air–fuel mixture in the crankcase, as shown in Fig. 9–14. Also, the intake and exhaust valves are replaced by openings in the lower portion of the cylinder wall. During the latter part of the power stroke, the piston uncovers first the exhaust port, allowing the exhaust gases to be partially expelled, and then the intake port, allowing the fresh air–fuel mixture to rush in and drive most of the remaining exhaust gases out of the cylinder. This mixture is then compressed as the piston moves upward during the compression stroke and is subsequently ignited by a spark plug.

The two-stroke engines are generally less efficient than their four-stroke counterparts because of the incomplete expulsion of the exhaust gases and the partial expulsion of the fresh air–fuel mixture with the exhaust gases. However, they are relatively simple and inexpensive, and they have high power-to-weight and power-to-volume ratios, which make them suitable for applications requiring small size and weight such as for motorcycles, chain saws, and lawn mowers (Fig. 9–15).

Advances in several technologies—such as direct fuel injection, stratified charge combustion, and electronic controls—brought about a renewed interest in two-stroke engines that can offer high performance and fuel economy while satisfying the stringent emission requirements. For a given weight and displacement, a well-designed two-stroke engine can provide significantly more power than its four-stroke counterpart because two-stroke engines produce power on every engine revolution instead of every other one. In the new two-stroke engines, the highly atomized fuel spray that is injected into the combustion chamber toward the end of the compression stroke burns much more completely. The fuel is sprayed after the exhaust valve is closed, which prevents unburned fuel from being ejected into the atmosphere. With stratified combustion, the flame that is initiated by igniting a small amount of the rich fuel–air mixture near the spark plug propagates through the combustion chamber filled with a much leaner mixture, and this results in much cleaner combustion. Also, the advances in electronics have made it possible to ensure the optimum operation under varying engine load and speed conditions.

INTERACTIVE TUTORIAL

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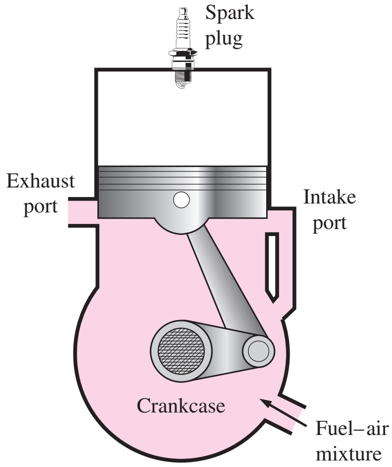

FIGURE 9–14

Schematic of a two-stroke reciprocating engine.

FIGURE 9–15

Two-stroke engines are commonly used in motorcycles and lawn mowers.

Major car companies have research programs underway on two-stroke engines which are expected to make a comeback in the future.

The thermodynamic analysis of the actual four-stroke or two-stroke cycles described is not a simple task. However, the analysis can be simplified significantly if the air-standard assumptions are utilized. The resulting cycle, which closely resembles the actual operating conditions, is the ideal **Otto cycle.** It consists of four internally reversible processes:

1-2 Isentropic compression
2-3 Constant-volume heat addition
3-4 Isentropic expansion
4-1 Constant-volume heat rejection

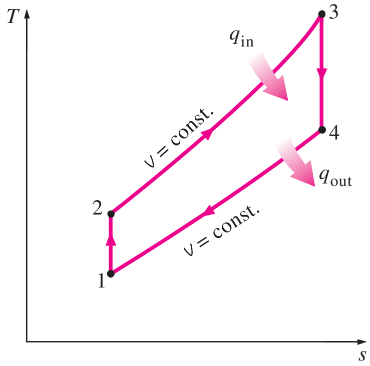

FIGURE 9–16
T-s diagram of the ideal Otto cycle.

The execution of the Otto cycle in a piston–cylinder device together with a P-v diagram is illustrated in Fig. 9–13*b*. The T-s diagram of the Otto cycle is given in Fig. 9–16.

The Otto cycle is executed in a closed system, and disregarding the changes in kinetic and potential energies, the energy balance for any of the processes is expressed, on a unit-mass basis, as

$$(q_{in} - q_{out}) + (w_{in} - w_{out}) = \Delta u \qquad (\text{kJ/kg}) \tag{9–5}$$

No work is involved during the two heat transfer processes since both take place at constant volume. Therefore, heat transfer to and from the working fluid can be expressed as

$$q_{in} = u_3 - u_2 = c_v(T_3 - T_2) \tag{9–6a}$$

and

$$q_{out} = u_4 - u_1 = c_v(T_4 - T_1) \tag{9–6b}$$

Then the thermal efficiency of the ideal Otto cycle under the cold air standard assumptions becomes

$$\eta_{th,Otto} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{T_4 - T_1}{T_3 - T_2} = 1 - \frac{T_1(T_4/T_1 - 1)}{T_2(T_3/T_2 - 1)}$$

Processes 1-2 and 3-4 are isentropic, and $v_2 = v_3$ and $v_4 = v_1$. Thus,

$$\frac{T_1}{T_2} = \left(\frac{v_2}{v_1}\right)^{k-1} = \left(\frac{v_3}{v_4}\right)^{k-1} = \frac{T_4}{T_3} \tag{9–7}$$

Substituting these equations into the thermal efficiency relation and simplifying give

$$\eta_{th,Otto} = 1 - \frac{1}{r^{k-1}} \tag{9–8}$$

where

$$r = \frac{V_{max}}{V_{min}} = \frac{V_1}{V_2} = \frac{v_1}{v_2} \tag{9–9}$$

is the **compression ratio** and k is the specific heat ratio c_p/c_v.

Equation 9–8 shows that under the cold-air-standard assumptions, the thermal efficiency of an ideal Otto cycle depends on the compression ratio of the engine and the specific heat ratio of the working fluid. The thermal efficiency of the ideal Otto cycle increases with both the compression ratio and the specific heat ratio. This is also true for actual spark-ignition internal combustion engines. A plot of thermal efficiency versus the compression ratio is given in Fig. 9–17 for $k = 1.4$, which is the specific heat ratio value of air at room temperature. For a given compression ratio, the thermal efficiency of an actual spark-ignition engine is less than that of an ideal Otto cycle because of the irreversibilities, such as friction, and other factors such as incomplete combustion.

We can observe from Fig. 9–17 that the thermal efficiency curve is rather steep at low compression ratios but flattens out starting with a compression ratio value of about 8. Therefore, the increase in thermal efficiency with the compression ratio is not as pronounced at high compression ratios. Also, when high compression ratios are used, the temperature of the air–fuel mixture rises above the autoignition temperature of the fuel (the temperature at which the fuel ignites without the help of a spark) during the combustion process, causing an early and rapid burn of the fuel at some point or points ahead of the flame front, followed by almost instantaneous inflammation of the end gas. This premature ignition of the fuel, called **autoignition,** produces an audible noise, which is called **engine knock.** Autoignition in spark-ignition engines cannot be tolerated because it hurts performance and can cause engine damage. The requirement that autoignition not be allowed places an upper limit on the compression ratios that can be used in spark-ignition internal combustion engines.

Improvement of the thermal efficiency of gasoline engines by utilizing higher compression ratios (up to about 12) without facing the autoignition problem has been made possible by using gasoline blends that have good antiknock characteristics, such as gasoline mixed with tetraethyl lead. Tetraethyl lead had been added to gasoline since the 1920s because it is an inexpensive method of raising the *octane rating,* which is a measure of the engine knock resistance of a fuel. Leaded gasoline, however, has a very undesirable side effect: it forms compounds during the combustion process that are hazardous to health and pollute the environment. In an effort to combat air pollution, the government adopted a policy in the mid-1970s that resulted in the eventual phase-out of leaded gasoline. Unable to use lead, the refiners developed other techniques to improve the antiknock characteristics of gasoline. Most cars made since 1975 have been designed to use unleaded gasoline, and the compression ratios had to be lowered to avoid engine knock. The ready availability of high octane fuels made it possible to raise the compression ratios again in recent years. Also, owing to the improvements in other areas (reduction in overall automobile weight, improved aerodynamic design, etc.), today's cars have better fuel economy and consequently get more miles per gallon of fuel. This is an example of how engineering decisions involve compromises, and efficiency is only one of the considerations in final design.

The second parameter affecting the thermal efficiency of an ideal Otto cycle is the specific heat ratio k. For a given compression ratio, an ideal Otto cycle using a monatomic gas (such as argon or helium, $k = 1.667$) as the working fluid will have the highest thermal efficiency. The specific heat ratio k, and thus the thermal efficiency of the ideal Otto cycle, decreases as the molecules of the working fluid get larger (Fig. 9–18). At room temperature it is 1.4 for air, 1.3 for carbon dioxide, and 1.2 for ethane. The working

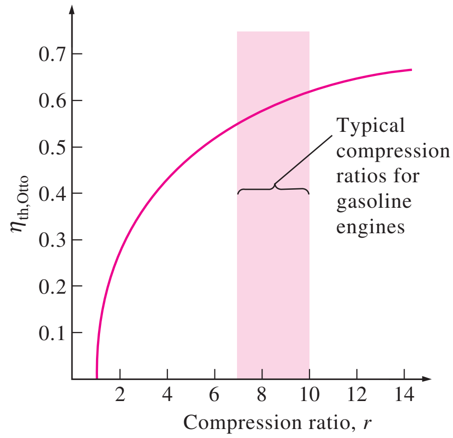

FIGURE 9–17

Thermal efficiency of the ideal Otto cycle as a function of compression ratio ($k = 1.4$).

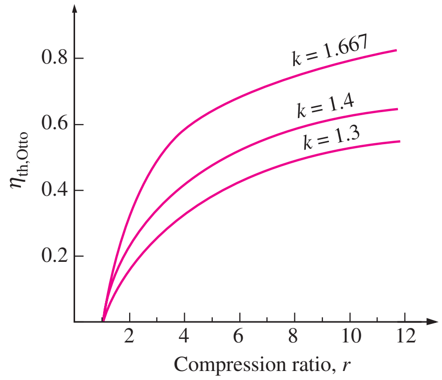

FIGURE 9–18

The thermal efficiency of the Otto cycle increases with the specific heat ratio k of the working fluid.

fluid in actual engines contains larger molecules such as carbon dioxide, and the specific heat ratio decreases with temperature, which is one of the reasons that the actual cycles have lower thermal efficiencies than the ideal Otto cycle. The thermal efficiencies of actual spark-ignition engines range from about 25 to 30 percent.

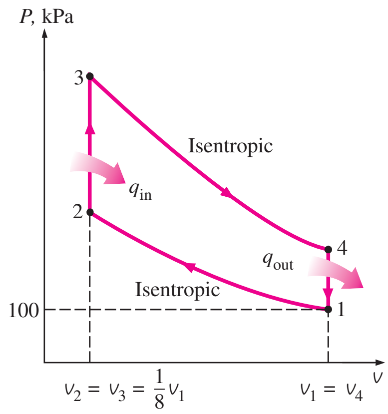

FIGURE 9–19

P-*v* diagram for the Otto cycle discussed in Example 9–2.

EXAMPLE 9–2 The Ideal Otto Cycle

An ideal Otto cycle has a compression ratio of 8. At the beginning of the compression process, air is at 100 kPa and 17°C, and 800 kJ/kg of heat is transferred to air during the constant-volume heat-addition process. Accounting for the variation of specific heats of air with temperature, determine (*a*) the maximum temperature and pressure that occur during the cycle, (*b*) the net work output, (*c*) the thermal efficiency, and (*d*) the mean effective pressure for the cycle.

Solution An ideal Otto cycle is considered. The maximum temperature and pressure, the net work output, the thermal efficiency, and the mean effective pressure are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** The variation of specific heats with temperature is to be accounted for.

Analysis The *P*-*v* diagram of the ideal Otto cycle described is shown in Fig. 9–19. We note that the air contained in the cylinder forms a closed system.

(*a*) The maximum temperature and pressure in an Otto cycle occur at the end of the constant-volume heat-addition process (state 3). But first we need to determine the temperature and pressure of air at the end of the isentropic compression process (state 2), using data from Table A–17:

$$T_1 = 290\text{ K} \;\rightarrow\; u_1 = 206.91\text{ kJ/kg}$$
$$v_{r1} = 676.1$$

Process 1-2 (isentropic compression of an ideal gas):

$$\frac{v_{r2}}{v_{r1}} = \frac{v_2}{v_1} = \frac{1}{r} \;\rightarrow\; v_{r2} = \frac{v_{r1}}{r} = \frac{676.1}{8} = 84.51 \;\rightarrow\; T_2 = 652.4\text{ K}$$
$$u_2 = 475.11\text{ kJ/kg}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \;\rightarrow\; P_2 = P_1\left(\frac{T_2}{T_1}\right)\left(\frac{v_1}{v_2}\right)$$
$$= (100\text{ kPa})\left(\frac{652.4\text{ K}}{290\text{ K}}\right)(8) = 1799.7\text{ kPa}$$

Process 2-3 (constant-volume heat addition):

$$q_{in} = u_3 - u_2$$
$$800\text{ kJ/kg} = u_3 - 475.11\text{ kJ/kg}$$
$$u_3 = 1275.11\text{ kJ/kg} \;\rightarrow\; T_3 = \mathbf{1575.1\text{ K}}$$
$$v_{r3} = 6.108$$

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \rightarrow P_3 = P_2\left(\frac{T_3}{T_2}\right)\left(\frac{v_2}{v_3}\right)$$

$$= (1.7997 \text{ MPa})\left(\frac{1575.1 \text{ K}}{652.4 \text{ K}}\right)(1) = \mathbf{4.345 \text{ MPa}}$$

(*b*) The net work output for the cycle is determined either by finding the boundary ($P\,dV$) work involved in each process by integration and adding them or by finding the net heat transfer that is equivalent to the net work done during the cycle. We take the latter approach. However, first we need to find the internal energy of the air at state 4:

Process 3-4 (isentropic expansion of an ideal gas):

$$\frac{v_{r4}}{v_{r3}} = \frac{v_4}{v_3} = r \rightarrow v_{r4} = rv_{r3} = (8)(6.108) = 48.864 \rightarrow T_4 = 795.6 \text{ K}$$

$$u_4 = 588.74 \text{ kJ/kg}$$

Process 4-1 (constant-volume heat rejection):

$$-q_{\text{out}} = u_1 - u_4 \rightarrow q_{\text{out}} = u_4 - u_1$$

$$q_{\text{out}} = 588.74 - 206.91 = 381.83 \text{ kJ/kg}$$

Thus,

$$w_{\text{net}} = q_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 800 - 381.83 = \mathbf{418.17 \text{ kJ/kg}}$$

(*c*) The thermal efficiency of the cycle is determined from its definition:

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{418.17 \text{ kJ/kg}}{800 \text{ kJ/kg}} = \mathbf{0.523 \text{ or } 52.3\%}$$

Under the cold-air-standard assumptions (constant specific heat values at room temperature), the thermal efficiency would be (Eq. 9–8)

$$\eta_{\text{th,Otto}} = 1 - \frac{1}{r^{k-1}} = 1 - r^{1-k} = 1 - (8)^{1-1.4} = 0.565 \text{ or } 56.5\%$$

which is considerably different from the value obtained above. Therefore, care should be exercised in utilizing the cold-air-standard assumptions.

(*d*) The mean effective pressure is determined from its definition, Eq. 9–4:

$$\text{MEP} = \frac{w_{\text{net}}}{v_1 - v_2} = \frac{w_{\text{net}}}{v_1 - v_1/r} = \frac{w_{\text{net}}}{v_1(1 - 1/r)}$$

where

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(290 \text{ K})}{100 \text{ kPa}} = 0.832 \text{ m}^3/\text{kg}$$

Thus,

$$\text{MEP} = \frac{418.17 \text{ kJ/kg}}{(0.832 \text{ m}^3/\text{kg})(1 - \frac{1}{8})}\left(\frac{1 \text{ kPa}\cdot\text{m}^3}{1 \text{ kJ}}\right) = \mathbf{574 \text{ kPa}}$$

Discussion Note that a constant pressure of 574 kPa during the power stroke would produce the same net work output as the entire cycle.

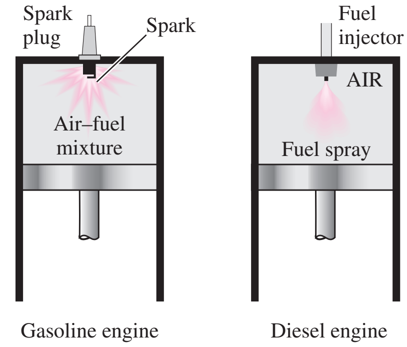

FIGURE 9–20

In diesel engines, the spark plug is replaced by a fuel injector, and only air is compressed during the compression process.

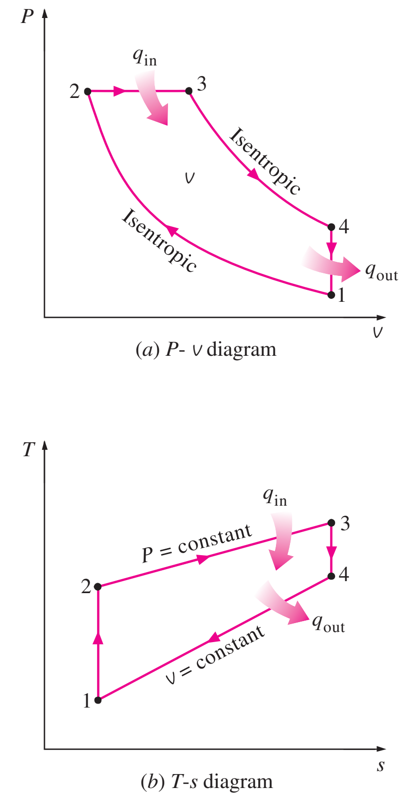

FIGURE 9–21

T-s and *P-v* diagrams for the ideal Diesel cycle.

9–6 ▪ DIESEL CYCLE: THE IDEAL CYCLE FOR COMPRESSION-IGNITION ENGINES

The Diesel cycle is the ideal cycle for CI reciprocating engines. The CI engine, first proposed by Rudolph Diesel in the 1890s, is very similar to the SI engine discussed in the last section, differing mainly in the method of initiating combustion. In spark-ignition engines (also known as *gasoline engines*), the air–fuel mixture is compressed to a temperature that is below the autoignition temperature of the fuel, and the combustion process is initiated by firing a spark plug. In CI engines (also known as *diesel engines*), the air is compressed to a temperature that is above the autoignition temperature of the fuel, and combustion starts on contact as the fuel is injected into this hot air. Therefore, the spark plug and carburetor are replaced by a fuel injector in diesel engines (Fig. 9–20).

In gasoline engines, a mixture of air and fuel is compressed during the compression stroke, and the compression ratios are limited by the onset of autoignition or engine knock. In diesel engines, only air is compressed during the compression stroke, eliminating the possibility of autoignition. Therefore, diesel engines can be designed to operate at much higher compression ratios, typically between 12 and 24. Not having to deal with the problem of autoignition has another benefit: many of the stringent requirements placed on the gasoline can now be removed, and fuels that are less refined (thus less expensive) can be used in diesel engines.

The fuel injection process in diesel engines starts when the piston approaches TDC and continues during the first part of the power stroke. Therefore, the combustion process in these engines takes place over a longer interval. Because of this longer duration, the combustion process in the ideal Diesel cycle is approximated as a constant-pressure heat-addition process. In fact, this is the only process where the Otto and the Diesel cycles differ. The remaining three processes are the same for both ideal cycles. That is, process 1-2 is isentropic compression, 3-4 is isentropic expansion, and 4-1 is constant-volume heat rejection. The similarity between the two cycles is also apparent from the *P-v* and *T-s* diagrams of the Diesel cycle, shown in Fig. 9–21.

Noting that the Diesel cycle is executed in a piston–cylinder device, which forms a closed system, the amount of heat transferred to the working fluid at constant pressure and rejected from it at constant volume can be expressed as

$$q_{in} - w_{b,out} = u_3 - u_2 \rightarrow q_{in} = P_2(v_3 - v_2) + (u_3 - u_2)$$
$$= h_3 - h_2 = c_p(T_3 - T_2) \quad \textbf{(9–10a)}$$

and

$$-q_{out} = u_1 - u_4 \rightarrow q_{out} = u_4 - u_1 = c_v(T_4 - T_1) \quad \textbf{(9–10b)}$$

Then the thermal efficiency of the ideal Diesel cycle under the cold-air-standard assumptions becomes

$$\eta_{th,Diesel} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{T_4 - T_1}{k(T_3 - T_2)} = 1 - \frac{T_1(T_4/T_1 - 1)}{kT_2(T_3/T_2 - 1)}$$

We now define a new quantity, the **cutoff ratio** r_c, as the ratio of the cylinder volumes after and before the combustion process:

$$r_c = \frac{V_3}{V_2} = \frac{v_3}{v_2} \tag{9–11}$$

Utilizing this definition and the isentropic ideal-gas relations for processes 1-2 and 3-4, we see that the thermal efficiency relation reduces to

$$\eta_{\text{th,Diesel}} = 1 - \frac{1}{r^{k-1}}\left[\frac{r_c^k - 1}{k(r_c - 1)}\right] \tag{9–12}$$

where r is the compression ratio defined by Eq. 9–9. Looking at Eq. 9–12 carefully, one would notice that under the cold-air-standard assumptions, the efficiency of a Diesel cycle differs from the efficiency of an Otto cycle by the quantity in the brackets. This quantity is always greater than 1. Therefore,

$$\eta_{\text{th,Otto}} > \eta_{\text{th,Diesel}} \tag{9–13}$$

when both cycles operate on the same compression ratio. Also, as the cutoff ratio decreases, the efficiency of the Diesel cycle increases (Fig. 9–22). For the limiting case of $r_c = 1$, the quantity in the brackets becomes unity (can you prove it?), and the efficiencies of the Otto and Diesel cycles become identical. Remember, though, that diesel engines operate at much higher compression ratios and thus are usually more efficient than the spark-ignition (gasoline) engines. The diesel engines also burn the fuel more completely since they usually operate at lower revolutions per minute and the air–fuel mass ratio is much higher than spark-ignition engines. Thermal efficiencies of large diesel engines range from about 35 to 40 percent.

The higher efficiency and lower fuel costs of diesel engines make them attractive in applications requiring relatively large amounts of power, such as in locomotive engines, emergency power generation units, large ships, and heavy trucks. As an example of how large a diesel engine can be, a 12-cylinder diesel engine built in 1964 by the Fiat Corporation of Italy had a normal power output of 25,200 hp (18.8 MW) at 122 rpm, a cylinder bore of 90 cm, and a stroke of 91 cm.

Approximating the combustion process in internal combustion engines as a constant-volume or a constant-pressure heat-addition process is overly simplistic and not quite realistic. Probably a better (but slightly more complex) approach would be to model the combustion process in both gasoline and diesel engines as a combination of two heat-transfer processes, one at constant volume and the other at constant pressure. The ideal cycle based on this concept is called the **dual cycle,** and a P-v diagram for it is given in Fig. 9–23. The relative amounts of heat transferred during each process can be adjusted to approximate the actual cycle more closely. Note that both the Otto and the Diesel cycles can be obtained as special cases of the dual cycle.

INTERACTIVE TUTORIAL

SEE TUTORIAL CH. 9, SEC. 3 ON THE DVD.

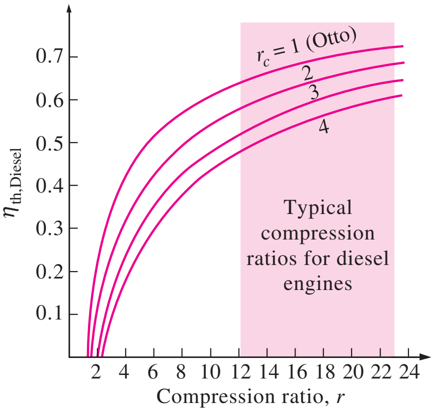

FIGURE 9–22

Thermal efficiency of the ideal Diesel cycle as a function of compression and cutoff ratios ($k = 1.4$).

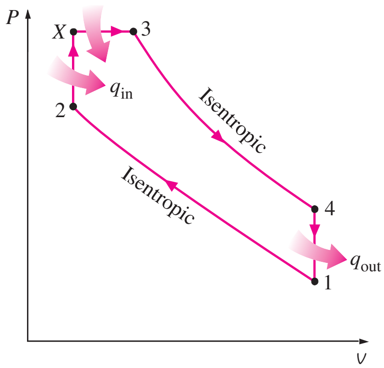

FIGURE 9–23

P-v diagram of an ideal dual cycle.

EXAMPLE 9–3 The Ideal Diesel Cycle

An ideal Diesel cycle with air as the working fluid has a compression ratio of 18 and a cutoff ratio of 2. At the beginning of the compression process, the working fluid is at 14.7 psia, 80°F, and 117 in^3. Utilizing the cold-air-standard assumptions, determine (*a*) the temperature and pressure of air at

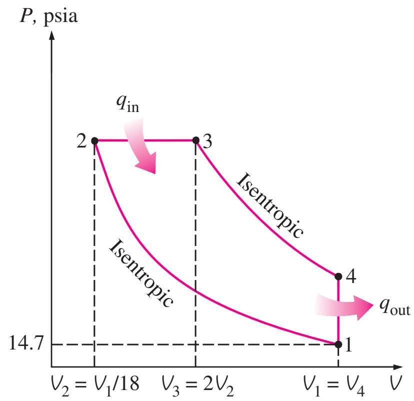

FIGURE 9–24

P-V diagram for the ideal Diesel cycle discussed in Example 9–3.

the end of each process, (*b*) the net work output and the thermal efficiency, and (*c*) the mean effective pressure.

Solution An ideal Diesel cycle is considered. The temperature and pressure at the end of each process, the net work output, the thermal efficiency, and the mean effective pressure are to be determined.

Assumptions **1** The cold-air-standard assumptions are applicable and thus air can be assumed to have constant specific heats at room temperature. **2** Kinetic and potential energy changes are negligible.

Properties The gas constant of air is $R = 0.3704$ psia · ft^3/lbm · R and its other properties at room temperature are $c_p = 0.240$ Btu/lbm · R, $c_v = 0.171$ Btu/lbm · R, and $k = 1.4$ (Table A–2E*a*).

Analysis The *P-V* diagram of the ideal Diesel cycle described is shown in Fig. 9–24. We note that the air contained in the cylinder forms a closed system.

(*a*) The temperature and pressure values at the end of each process can be determined by utilizing the ideal-gas isentropic relations for processes 1-2 and 3-4. But first we determine the volumes at the end of each process from the definitions of the compression ratio and the cutoff ratio:

$$V_2 = \frac{V_1}{r} = \frac{117 \text{ in}^3}{18} = 6.5 \text{ in}^3$$

$$V_3 = r_c V_2 = (2)(6.5 \text{ in}^3) = 13 \text{ in}^3$$

$$V_4 = V_1 = 117 \text{ in}^3$$

Process 1-2 (isentropic compression of an ideal gas, constant specific heats):

$$T_2 = T_1\left(\frac{V_1}{V_2}\right)^{k-1} = (540 \text{ R})(18)^{1.4-1} = \mathbf{1716\ R}$$

$$P_2 = P_1\left(\frac{V_1}{V_2}\right)^{k} = (14.7 \text{ psia})(18)^{1.4} = \mathbf{841\ psia}$$

Process 2-3 (constant-pressure heat addition to an ideal gas):

$$P_3 = P_2 = \mathbf{841\ psia}$$

$$\frac{P_2V_2}{T_2} = \frac{P_3V_3}{T_3} \longrightarrow T_3 = T_2\left(\frac{V_3}{V_2}\right) = (1716 \text{ R})(2) = \mathbf{3432\ R}$$

Process 3-4 (isentropic expansion of an ideal gas, constant specific heats):

$$T_4 = T_3\left(\frac{V_3}{V_4}\right)^{k-1} = (3432 \text{ R})\left(\frac{13 \text{ in}^3}{117 \text{ in}^3}\right)^{1.4-1} = \mathbf{1425\ R}$$

$$P_4 = P_3\left(\frac{V_3}{V_4}\right)^{k} = (841 \text{ psia})\left(\frac{13 \text{ in}^3}{117 \text{ in}^3}\right)^{1.4} = \mathbf{38.8\ psia}$$

(*b*) The net work for a cycle is equivalent to the net heat transfer. But first we find the mass of air:

$$m = \frac{P_1V_1}{RT_1} = \frac{(14.7 \text{ psia})(117 \text{ in}^3)}{(0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(540 \text{ R})}\left(\frac{1 \text{ ft}^3}{1728 \text{ in}^3}\right) = 0.00498 \text{ lbm}$$

Process 2-3 is a constant-pressure heat-addition process, for which the boundary work and Δu terms can be combined into Δh. Thus,

$$\begin{aligned} Q_{\text{in}} &= m(h_3 - h_2) = mc_p(T_3 - T_2) \\ &= (0.00498 \text{ lbm})(0.240 \text{ Btu/lbm}\cdot\text{R})[(3432 - 1716) \text{ R}] \\ &= 2.051 \text{ Btu} \end{aligned}$$

Process 4-1 is a constant-volume heat-rejection process (it involves no work interactions), and the amount of heat rejected is

$$\begin{aligned} Q_{\text{out}} &= m(u_4 - u_1) = mc_v(T_4 - T_1) \\ &= (0.00498 \text{ lbm})(0.171 \text{ Btu/lbm}\cdot\text{R})[(1425 - 540) \text{ R}] \\ &= 0.754 \text{ Btu} \end{aligned}$$

Thus,

$$W_{\text{net}} = Q_{\text{in}} - Q_{\text{out}} = 2.051 - 0.754 = \mathbf{1.297\ Btu}$$

Then the thermal efficiency becomes

$$\eta_{\text{th}} = \frac{W_{\text{net}}}{Q_{\text{in}}} = \frac{1.297 \text{ Btu}}{2.051 \text{ Btu}} = \mathbf{0.632 \text{ or } 63.2\%}$$

The thermal efficiency of this Diesel cycle under the cold-air-standard assumptions could also be determined from Eq. 9–12.

(*c*) The mean effective pressure is determined from its definition, Eq. 9–4:

$$\begin{aligned} \text{MEP} &= \frac{W_{\text{net}}}{V_{\max} - V_{\min}} = \frac{W_{\text{net}}}{V_1 - V_2} = \frac{1.297 \text{ Btu}}{(117 - 6.5) \text{ in}^3}\left(\frac{778.17 \text{ lbf}\cdot\text{ft}}{1 \text{ Btu}}\right)\left(\frac{12 \text{ in.}}{1 \text{ ft}}\right) \\ &= \mathbf{110\ psia} \end{aligned}$$

Discussion Note that a constant pressure of 110 psia during the power stroke would produce the same net work output as the entire Diesel cycle.

9–7 ▪ STIRLING AND ERICSSON CYCLES

The ideal Otto and Diesel cycles discussed in the preceding sections are composed entirely of internally reversible processes and thus are internally reversible cycles. These cycles are not totally reversible, however, since they involve heat transfer through a finite temperature difference during the non-isothermal heat-addition and heat-rejection processes, which are irreversible. Therefore, the thermal efficiency of an Otto or Diesel engine will be less than that of a Carnot engine operating between the same temperature limits.

Consider a heat engine operating between a heat source at T_H and a heat sink at T_L. For the heat-engine cycle to be totally reversible, the temperature difference between the working fluid and the heat source (or sink) should never exceed a differential amount dT during any heat-transfer process. That is, both the heat-addition and heat-rejection processes during the cycle must take place isothermally, one at a temperature of T_H and the other at a temperature of T_L. This is precisely what happens in a Carnot cycle.

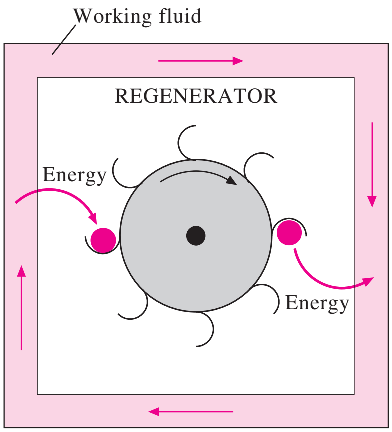

FIGURE 9–25

A regenerator is a device that borrows energy from the working fluid during one part of the cycle and pays it back (without interest) during another part.

There are two other cycles that involve an isothermal heat-addition process at T_H and an isothermal heat-rejection process at T_L: the *Stirling cycle* and the *Ericsson cycle.* They differ from the Carnot cycle in that the two isentropic processes are replaced by two constant-volume regeneration processes in the Stirling cycle and by two constant-pressure regeneration processes in the Ericsson cycle. Both cycles utilize **regeneration,** a process during which heat is transferred to a thermal energy storage device (called a *regenerator*) during one part of the cycle and is transferred back to the working fluid during another part of the cycle (Fig. 9–25).

Figure 9–26(*b*) shows the *T-s* and *P-v* diagrams of the **Stirling cycle,** which is made up of four totally reversible processes:

1-2 T = *constant* expansion (heat addition from the external source)

2-3 v = *constant* regeneration (internal heat transfer from the working fluid to the regenerator)

3-4 T = *constant* compression (heat rejection to the external sink)

4-1 v = *constant* regeneration (internal heat transfer from the regenerator back to the working fluid)

The execution of the Stirling cycle requires rather innovative hardware. The actual Stirling engines, including the original one patented by Robert Stirling, are heavy and complicated. To spare the reader the complexities, the execution of the Stirling cycle in a closed system is explained with the help of the hypothetical engine shown in Fig. 9–27.

This system consists of a cylinder with two pistons on each side and a regenerator in the middle. The regenerator can be a wire or a ceramic mesh

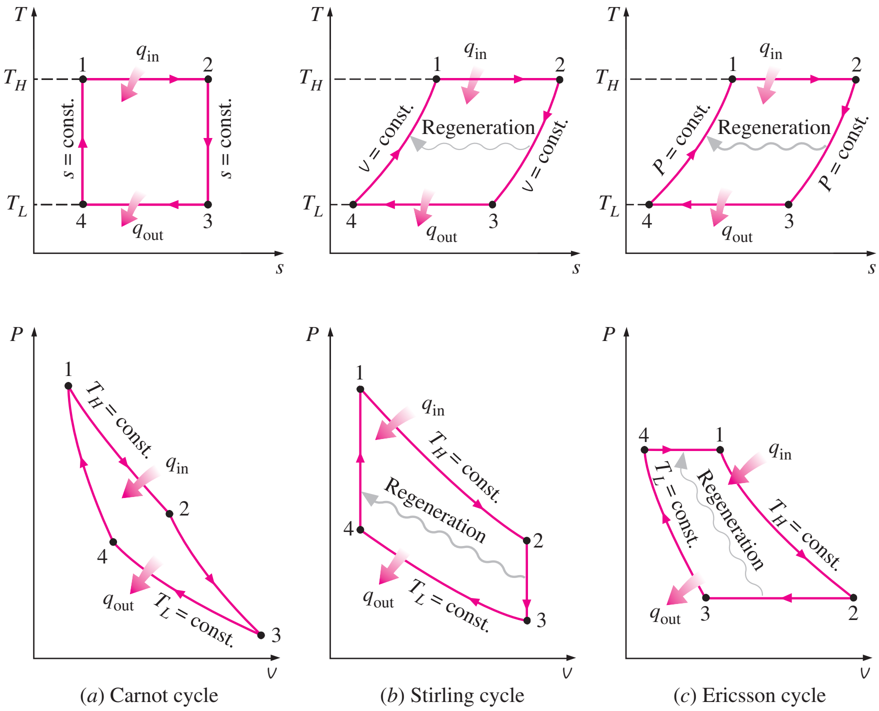

FIGURE 9–26

T-s and *P-v* diagrams of Carnot, Stirling, and Ericsson cycles.

or any kind of porous plug with a high thermal mass (mass times specific heat). It is used for the temporary storage of thermal energy. The mass of the working fluid contained within the regenerator at any instant is considered negligible.

Initially, the left chamber houses the entire working fluid (a gas), which is at a high temperature and pressure. During process 1-2, heat is transferred to the gas at T_H from a source at T_H. As the gas expands isothermally, the left piston moves outward, doing work, and the gas pressure drops. During process 2-3, both pistons are moved to the right at the same rate (to keep the volume constant) until the entire gas is forced into the right chamber. As the gas passes through the regenerator, heat is transferred to the regenerator and the gas temperature drops from T_H to T_L. For this heat transfer process to be reversible, the temperature difference between the gas and the regenerator should not exceed a differential amount dT at any point. Thus, the temperature of the regenerator will be T_H at the left end and T_L at the right end of the regenerator when state 3 is reached. During process 3-4, the right piston is moved inward, compressing the gas. Heat is transferred from the gas to a sink at temperature T_L so that the gas temperature remains constant at T_L while the pressure rises. Finally, during process 4-1, both pistons are moved to the left at the same rate (to keep the volume constant), forcing the entire gas into the left chamber. The gas temperature rises from T_L to T_H as it passes through the regenerator and picks up the thermal energy stored there during process 2-3. This completes the cycle.

Notice that the second constant-volume process takes place at a smaller volume than the first one, and the net heat transfer to the regenerator during a cycle is zero. That is, the amount of energy stored in the regenerator during process 2-3 is equal to the amount picked up by the gas during process 4-1.

The *T*-*s* and *P*-*v* diagrams of the **Ericsson cycle** are shown in Fig. 9–26*c*. The Ericsson cycle is very much like the Stirling cycle, except that the two constant-volume processes are replaced by two constant-pressure processes.

A steady-flow system operating on an Ericsson cycle is shown in Fig. 9–28. Here the isothermal expansion and compression processes are executed in a compressor and a turbine, respectively, and a counter-flow heat exchanger serves as a regenerator. Hot and cold fluid streams enter the heat exchanger from opposite ends, and heat transfer takes place between the two streams. In the ideal case, the temperature difference between the two fluid streams does not exceed a differential amount at any point, and the cold fluid stream leaves the heat exchanger at the inlet temperature of the hot stream.

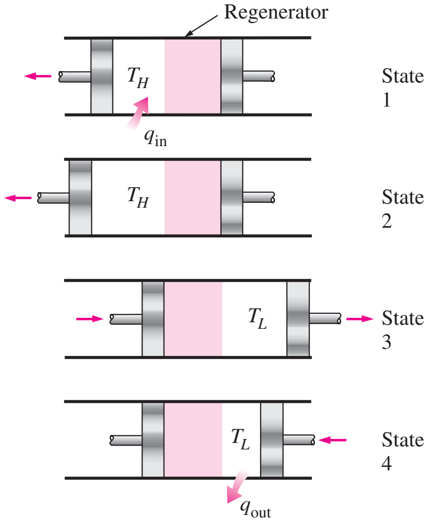

FIGURE 9–27

The execution of the Stirling cycle.

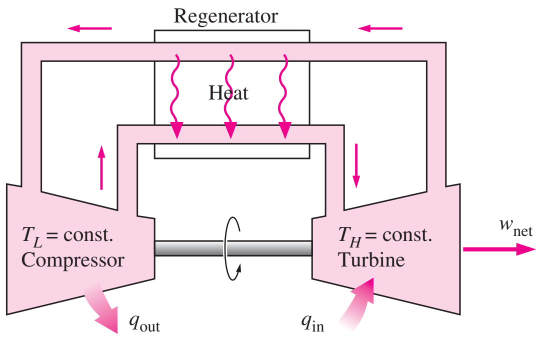

FIGURE 9–28

A steady-flow Ericsson engine.

Both the Stirling and Ericsson cycles are totally reversible, as is the Carnot cycle, and thus according to the Carnot principle, all three cycles must have the same thermal efficiency when operating between the same temperature limits:

$$\eta_{\text{th,Stirling}} = \eta_{\text{th,Ericsson}} = \eta_{\text{th,Carnot}} = 1 - \frac{T_L}{T_H} \tag{9–14}$$

This is proved for the Carnot cycle in Example 9–1 and can be proved in a similar manner for both the Stirling and Ericsson cycles.

EXAMPLE 9–4 Thermal Efficiency of the Ericsson Cycle

Using an ideal gas as the working fluid, show that the thermal efficiency of an Ericsson cycle is identical to the efficiency of a Carnot cycle operating between the same temperature limits.

Solution It is to be shown that the thermal efficiencies of Carnot and Ericsson cycles are identical.

Analysis Heat is transferred to the working fluid isothermally from an external source at temperature T_H during process 1-2, and it is rejected again isothermally to an external sink at temperature T_L during process 3-4. For a reversible isothermal process, heat transfer is related to the entropy change by

$$q = T\,\Delta s$$

The entropy change of an ideal gas during an isothermal process is

$$\Delta s = c_p \ln \overset{\nearrow 0}{\frac{T_e}{T_i}} - R \ln \frac{P_e}{P_i} = -R \ln \frac{P_e}{P_i}$$

The heat input and heat output can be expressed as

$$q_{\text{in}} = T_H(s_2 - s_1) = T_H\left(-R \ln \frac{P_2}{P_1}\right) = RT_H \ln \frac{P_1}{P_2}$$

and

$$q_{\text{out}} = T_L(s_4 - s_3) = -T_L\left(-R \ln \frac{P_4}{P_3}\right) = RT_L \ln \frac{P_4}{P_3}$$

Then the thermal efficiency of the Ericsson cycle becomes

$$\eta_{\text{th,Ericsson}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{RT_L \ln(P_4/P_3)}{RT_H \ln(P_1/P_2)} = 1 - \frac{T_L}{T_H}$$

since $P_1 = P_4$ and $P_3 = P_2$. Notice that this result is independent of whether the cycle is executed in a closed or steady-flow system.

Stirling and Ericsson cycles are difficult to achieve in practice because they involve heat transfer through a differential temperature difference in all components including the regenerator. This would require providing infinitely large surface areas for heat transfer or allowing an infinitely long time for the process. Neither is practical. In reality, all heat transfer processes take place through a finite temperature difference, the regenerator does not have an efficiency of 100 percent, and the pressure losses in the regenerator are considerable. Because of these limitations, both Stirling and Ericsson cycles have long been of only theoretical interest. However, there is renewed interest in engines that operate on these cycles because of their potential for higher efficiency and better emission control. The Ford Motor Company, General Motors Corporation, and the Phillips Research Laboratories of the Netherlands have successfully developed Stirling engines suitable for trucks, buses, and even automobiles. More research and development are needed before these engines can compete with the gasoline or diesel engines.

Both the Stirling and the Ericsson engines are *external combustion* engines. That is, the fuel in these engines is burned outside the cylinder, as opposed to gasoline or diesel engines, where the fuel is burned inside the cylinder.

External combustion offers several advantages. First, a variety of fuels can be used as a source of thermal energy. Second, there is more time for combustion, and thus the combustion process is more complete, which means less air pollution and more energy extraction from the fuel. Third, these engines operate on closed cycles, and thus a working fluid that has the most desirable characteristics (stable, chemically inert, high thermal conductivity) can be utilized as the working fluid. Hydrogen and helium are two gases commonly employed in these engines.

Despite the physical limitations and impracticalities associated with them, both the Stirling and Ericsson cycles give a strong message to design engineers: *Regeneration can increase efficiency.* It is no coincidence that modern gas-turbine and steam power plants make extensive use of regeneration. In fact, the Brayton cycle with intercooling, reheating, and regeneration, which is utilized in large gas-turbine power plants and discussed later in this chapter, closely resembles the Ericsson cycle.

9–8 ▪ BRAYTON CYCLE: THE IDEAL CYCLE FOR GAS-TURBINE ENGINES

INTERACTIVE TUTORIAL

SEE TUTORIAL CH. 9, SEC. 4 ON THE DVD.

The Brayton cycle was first proposed by George Brayton for use in the reciprocating oil-burning engine that he developed around 1870. Today, it is used for gas turbines only where both the compression and expansion processes take place in rotating machinery. Gas turbines usually operate on an *open cycle,* as shown in Fig. 9–29. Fresh air at ambient conditions is drawn into the compressor, where its temperature and pressure are raised. The high-pressure air proceeds into the combustion chamber, where the fuel is burned at constant pressure. The resulting high-temperature gases then enter the turbine, where they expand to the atmospheric pressure while producing power. The exhaust gases leaving the turbine are thrown out (not recirculated), causing the cycle to be classified as an open cycle.

The open gas-turbine cycle described above can be modeled as a *closed cycle,* as shown in Fig. 9–30, by utilizing the air-standard assumptions. Here the compression and expansion processes remain the same, but the combustion process is replaced by a constant-pressure heat-addition process from an external source, and the exhaust process is replaced by a constant-pressure heat-rejection process to the ambient air. The ideal cycle that the working fluid undergoes in this closed loop is the **Brayton cycle,** which is made up of four internally reversible processes:

1-2 Isentropic compression (in a compressor)

2-3 Constant-pressure heat addition

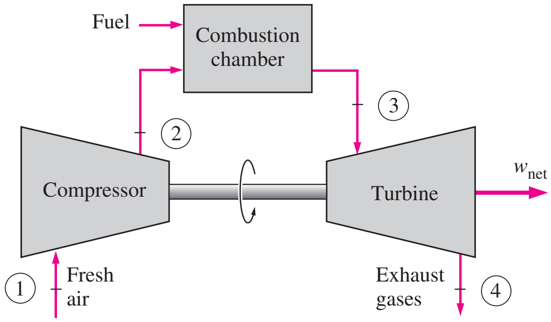

FIGURE 9–29
An open-cycle gas-turbine engine.

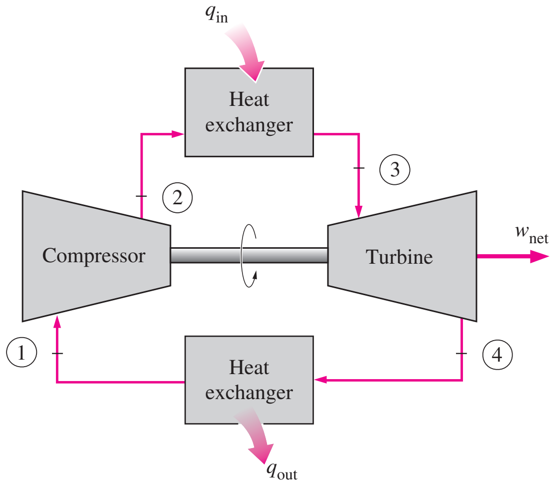

FIGURE 9–30
A closed-cycle gas-turbine engine.

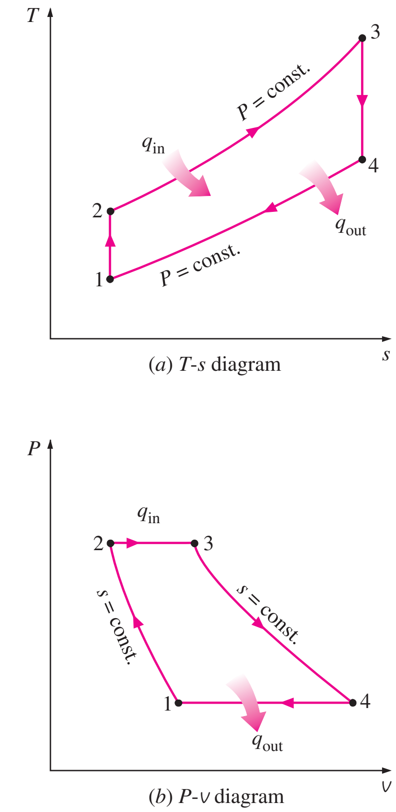

FIGURE 9–31
T-s and P-v diagrams for the ideal Brayton cycle.

3-4 Isentropic expansion (in a turbine)
4-1 Constant-pressure heat rejection

The T-s and P-v diagrams of an ideal Brayton cycle are shown in Fig. 9–31. Notice that all four processes of the Brayton cycle are executed in steady-flow devices; thus, they should be analyzed as steady-flow processes. When the changes in kinetic and potential energies are neglected, the energy balance for a steady-flow process can be expressed, on a unit–mass basis, as

$$(q_{in} - q_{out}) + (w_{in} - w_{out}) = h_{exit} - h_{inlet} \tag{9–15}$$

Therefore, heat transfers to and from the working fluid are

$$q_{in} = h_3 - h_2 = c_p(T_3 - T_2) \tag{9–16a}$$

and

$$q_{out} = h_4 - h_1 = c_p(T_4 - T_1) \tag{9–16b}$$

Then the thermal efficiency of the ideal Brayton cycle under the cold-air-standard assumptions becomes

$$\eta_{th,Brayton} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{c_p(T_4 - T_1)}{c_p(T_3 - T_2)} = 1 - \frac{T_1(T_4/T_1 - 1)}{T_2(T_3/T_2 - 1)}$$

Processes 1-2 and 3-4 are isentropic, and $P_2 = P_3$ and $P_4 = P_1$. Thus,

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{(k-1)/k} = \left(\frac{P_3}{P_4}\right)^{(k-1)/k} = \frac{T_3}{T_4}$$

Substituting these equations into the thermal efficiency relation and simplifying give

$$\eta_{th,Brayton} = 1 - \frac{1}{r_p^{(k-1)/k}} \tag{9–17}$$

where

$$r_p = \frac{P_2}{P_1} \tag{9–18}$$

is the **pressure ratio** and k is the specific heat ratio. Equation 9–17 shows that under the cold-air-standard assumptions, the thermal efficiency of an ideal Brayton cycle depends on the pressure ratio of the gas turbine and the specific heat ratio of the working fluid. The thermal efficiency increases with both of these parameters, which is also the case for actual gas turbines. A plot of thermal efficiency versus the pressure ratio is given in Fig. 9–32 for $k = 1.4$, which is the specific-heat-ratio value of air at room temperature.

The highest temperature in the cycle occurs at the end of the combustion process (state 3), and it is limited by the maximum temperature that the turbine blades can withstand. This also limits the pressure ratios that can be used in the cycle. For a fixed turbine inlet temperature T_3, the net work output per cycle increases with the pressure ratio, reaches a maximum, and then starts to decrease, as shown in Fig. 9–33. Therefore, there should be a compromise between the pressure ratio (thus the thermal efficiency) and the net work output. With less work output per cycle, a larger mass flow rate (thus a larger system) is needed to maintain the same power output, which may not be economical. In most common designs, the pressure ratio of gas turbines ranges from about 11 to 16.

The air in gas turbines performs two important functions: It supplies the necessary oxidant for the combustion of the fuel, and it serves as a coolant to keep the temperature of various components within safe limits. The second function is accomplished by drawing in more air than is needed for the complete combustion of the fuel. In gas turbines, an air–fuel mass ratio of 50 or above is not uncommon. Therefore, in a cycle analysis, treating the combustion gases as air does not cause any appreciable error. Also, the mass flow rate through the turbine is greater than that through the compressor, the difference being equal to the mass flow rate of the fuel. Thus, assuming a constant mass flow rate throughout the cycle yields conservative results for open-loop gas-turbine engines.

The two major application areas of gas-turbine engines are *aircraft propulsion* and *electric power generation.* When it is used for aircraft propulsion, the gas turbine produces just enough power to drive the compressor and a small generator to power the auxiliary equipment. The high-velocity exhaust gases are responsible for producing the necessary thrust to propel the aircraft. Gas turbines are also used as stationary power plants to generate electricity as stand-alone units or in conjunction with steam power plants on the high-temperature side. In these plants, the exhaust gases of the gas turbine serve as the heat source for the steam. The gas-turbine cycle can also be executed as a closed cycle for use in nuclear power plants. This time the working fluid is not limited to air, and a gas with more desirable characteristics (such as helium) can be used.

The majority of the Western world's naval fleets already use gas-turbine engines for propulsion and electric power generation. The General Electric LM2500 gas turbines used to power ships have a simple-cycle thermal efficiency of 37 percent. The General Electric WR-21 gas turbines equipped with intercooling and regeneration have a thermal efficiency of 43 percent and

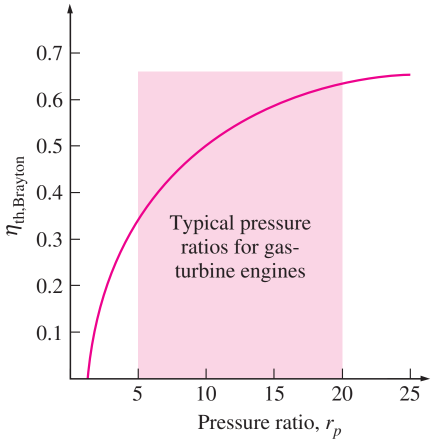

FIGURE 9–32

Thermal efficiency of the ideal Brayton cycle as a function of the pressure ratio.

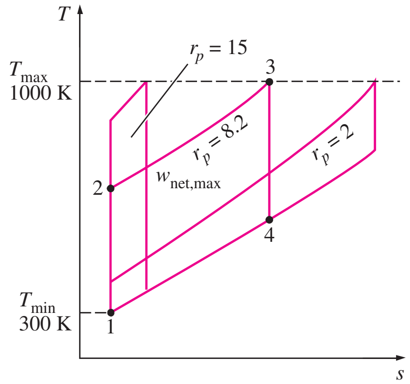

FIGURE 9–33

For fixed values of T_{min} and T_{max}, the net work of the Brayton cycle first increases with the pressure ratio, then reaches a maximum at $r_p = (T_{max}/T_{min})^{k/[2(k-1)]}$, and finally decreases.

produce 21.6 MW (29,040 hp). The regeneration also reduces the exhaust temperature from 600°C (1100°F) to 350°C (650°F). Air is compressed to 3 atm before it enters the intercooler. Compared to steam-turbine and diesel-propulsion systems, the gas turbine offers greater power for a given size and weight, high reliability, long life, and more convenient operation. The engine start-up time has been reduced from 4 h required for a typical steam-propulsion system to less than 2 min for a gas turbine. Many modern marine propulsion systems use gas turbines together with diesel engines because of the high fuel consumption of simple-cycle gas-turbine engines. In combined diesel and gas-turbine systems, diesel is used to provide for efficient low-power and cruise operation, and gas turbine is used when high speeds are needed.

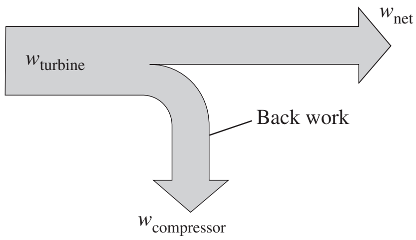

FIGURE 9–34

The fraction of the turbine work used to drive the compressor is called the back work ratio.

In gas-turbine power plants, the ratio of the compressor work to the turbine work, called the **back work ratio,** is very high (Fig. 9–34). Usually more than one-half of the turbine work output is used to drive the compressor. The situation is even worse when the isentropic efficiencies of the compressor and the turbine are low. This is quite in contrast to steam power plants, where the back work ratio is only a few percent. This is not surprising, however, since a liquid is compressed in steam power plants instead of a gas, and the steady-flow work is proportional to the specific volume of the working fluid.

A power plant with a high back work ratio requires a larger turbine to provide the additional power requirements of the compressor. Therefore, the turbines used in gas-turbine power plants are larger than those used in steam power plants of the same net power output.

Development of Gas Turbines

The gas turbine has experienced phenomenal progress and growth since its first successful development in the 1930s. The early gas turbines built in the 1940s and even 1950s had simple-cycle efficiencies of about 17 percent because of the low compressor and turbine efficiencies and low turbine inlet temperatures due to metallurgical limitations of those times. Therefore, gas turbines found only limited use despite their versatility and their ability to burn a variety of fuels. The efforts to improve the cycle efficiency concentrated in three areas:

1. **Increasing the turbine inlet (or firing) temperatures** This has been the primary approach taken to improve gas-turbine efficiency. The turbine inlet temperatures have increased steadily from about 540°C (1000°F) in the 1940s to 1425°C (2600°F) and even higher today. These increases were made possible by the development of new materials and the innovative cooling techniques for the critical components such as coating the turbine blades with ceramic layers and cooling the blades with the discharge air from the compressor. Maintaining high turbine inlet temperatures with an air-cooling technique requires the combustion temperature to be higher to compensate for the cooling effect of the cooling air. However, higher combustion temperatures increase the amount of nitrogen oxides (NO_x), which are responsible for the formation of ozone at ground level and smog. Using steam as the coolant allowed an increase in the turbine inlet temperatures by 200°F without an increase in the combustion temperature. Steam is also a much more effective heat transfer medium than air.

2. **Increasing the efficiencies of turbomachinery components** The performance of early turbines suffered greatly from the inefficiencies of turbines and compressors. However, the advent of computers and advanced techniques for computer-aided design made it possible to design these components aerodynamically with minimal losses. The increased efficiencies of the turbines and compressors resulted in a significant increase in the cycle efficiency.

3. **Adding modifications to the basic cycle** The simple-cycle efficiencies of early gas turbines were practically doubled by incorporating intercooling, regeneration (or recuperation), and reheating, discussed in the next two sections. These improvements, of course, come at the expense of increased initial and operation costs, and they cannot be justified unless the decrease in fuel costs offsets the increase in other costs. The relatively low fuel prices, the general desire in the industry to minimize installation costs, and the tremendous increase in the simple-cycle efficiency to about 40 percent left little desire for opting for these modifications.

The first gas turbine for an electric utility was installed in 1949 in Oklahoma as part of a combined-cycle power plant. It was built by General Electric and produced 3.5 MW of power. Gas turbines installed until the mid-1970s suffered from low efficiency and poor reliability. In the past, the base-load electric power generation was dominated by large coal and nuclear power plants. However, there has been an historic shift toward natural gas–fired gas turbines because of their higher efficiencies, lower capital costs, shorter installation times, and better emission characteristics, and the abundance of natural gas supplies, and more and more electric utilities are using gas turbines for base-load power production as well as for peaking. The construction costs for gas-turbine power plants are roughly half that of comparable conventional fossil-fuel steam power plants, which were the primary base-load power plants until the early 1980s. More than half of all power plants to be installed in the foreseeable future are forecast to be gas-turbine or combined gas–steam turbine types.

A gas turbine manufactured by General Electric in the early 1990s had a pressure ratio of 13.5 and generated 135.7 MW of net power at a thermal efficiency of 33 percent in simple-cycle operation. A more recent gas turbine manufactured by General Electric uses a turbine inlet temperature of 1425°C (2600°F) and produces up to 282 MW while achieving a thermal efficiency of 39.5 percent in the simple-cycle mode. A 1.3-ton small-scale gas turbine labeled OP-16, built by the Dutch firm Opra Optimal Radial Turbine, can run on gas or liquid fuel and can replace a 16-ton diesel engine. It has a pressure ratio of 6.5 and produces up to 2 MW of power. Its efficiency is 26 percent in the simple-cycle operation, which rises to 37 percent when equipped with a regenerator.

EXAMPLE 9–5 The Simple Ideal Brayton Cycle

A gas-turbine power plant operating on an ideal Brayton cycle has a pressure ratio of 8. The gas temperature is 300 K at the compressor inlet and 1300 K at the turbine inlet. Utilizing the air-standard assumptions, determine (*a*) the

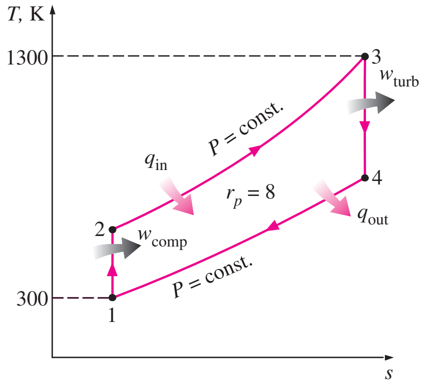

FIGURE 9–35

T-s diagram for the Brayton cycle discussed in Example 9–5.

gas temperature at the exits of the compressor and the turbine, (*b*) the back work ratio, and (*c*) the thermal efficiency.

Solution A power plant operating on the ideal Brayton cycle is considered. The compressor and turbine exit temperatures, back work ratio, and the thermal efficiency are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** The variation of specific heats with temperature is to be considered.

Analysis The *T-s* diagram of the ideal Brayton cycle described is shown in Fig. 9–35. We note that the components involved in the Brayton cycle are steady-flow devices.

(*a*) The air temperatures at the compressor and turbine exits are determined from isentropic relations:

Process 1-2 (isentropic compression of an ideal gas):

$$T_1 = 300 \text{ K} \rightarrow h_1 = 300.19 \text{ kJ/kg}$$
$$P_{r1} = 1.386$$

$$P_{r2} = \frac{P_2}{P_1}P_{r1} = (8)(1.386) = 11.09 \rightarrow T_2 = \mathbf{540\ K} \qquad \text{(at compressor exit)}$$
$$h_2 = 544.35 \text{ kJ/kg}$$

Process 3-4 (isentropic expansion of an ideal gas):

$$T_3 = 1300 \text{ K} \rightarrow h_3 = 1395.97 \text{ kJ/kg}$$
$$P_{r3} = 330.9$$

$$P_{r4} = \frac{P_4}{P_3}P_{r3} = \left(\frac{1}{8}\right)(330.9) = 41.36 \rightarrow T_4 = \mathbf{770\ K} \qquad \text{(at turbine exit)}$$
$$h_4 = 789.37 \text{ kJ/kg}$$

(*b*) To find the back work ratio, we need to find the work input to the compressor and the work output of the turbine:

$$w_{comp,in} = h_2 - h_1 = 544.35 - 300.19 = 244.16 \text{ kJ/kg}$$
$$w_{turb,out} = h_3 - h_4 = 1395.97 - 789.37 = 606.60 \text{ kJ/kg}$$

Thus,

$$r_{bw} = \frac{w_{comp,in}}{w_{turb,out}} = \frac{244.16 \text{ kJ/kg}}{606.60 \text{ kJ/kg}} = \mathbf{0.403}$$

That is, 40.3 percent of the turbine work output is used just to drive the compressor.

(*c*) The thermal efficiency of the cycle is the ratio of the net power output to the total heat input:

$$q_{in} = h_3 - h_2 = 1395.97 - 544.35 = 851.62 \text{ kJ/kg}$$
$$w_{net} = w_{out} - w_{in} = 606.60 - 244.16 = 362.4 \text{ kJ/kg}$$

Thus,

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{362.4 \text{ kJ/kg}}{851.62 \text{ kJ/kg}} = \mathbf{0.426 \text{ or } 42.6\%}$$

The thermal efficiency could also be determined from

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}}$$

where

$$q_{\text{out}} = h_4 - h_1 = 789.37 - 300.19 = 489.2 \text{ kJ/kg}$$

Discussion Under the cold-air-standard assumptions (constant specific heat values at room temperature), the thermal efficiency would be, from Eq. 9–17,

$$\eta_{\text{th,Brayton}} = 1 - \frac{1}{r_p^{(k-1)/k}} = 1 - \frac{1}{8^{(1.4-1)/1.4}} = 0.448$$

which is sufficiently close to the value obtained by accounting for the variation of specific heats with temperature.

Deviation of Actual Gas-Turbine Cycles from Idealized Ones

The actual gas-turbine cycle differs from the ideal Brayton cycle on several accounts. For one thing, some pressure drop during the heat-addition and heat-rejection processes is inevitable. More importantly, the actual work input to the compressor is more, and the actual work output from the turbine is less because of irreversibilities. The deviation of actual compressor and turbine behavior from the idealized isentropic behavior can be accurately accounted for by utilizing the isentropic efficiencies of the turbine and compressor as

$$\eta_C = \frac{w_s}{w_a} \cong \frac{h_{2s} - h_1}{h_{2a} - h_1} \tag{9–19}$$

and

$$\eta_T = \frac{w_a}{w_s} \cong \frac{h_3 - h_{4a}}{h_3 - h_{4s}} \tag{9–20}$$

where states $2a$ and $4a$ are the actual exit states of the compressor and the turbine, respectively, and $2s$ and $4s$ are the corresponding states for the isentropic case, as illustrated in Fig. 9–36. The effect of the turbine and compressor efficiencies on the thermal efficiency of the gas-turbine engines is illustrated below with an example.

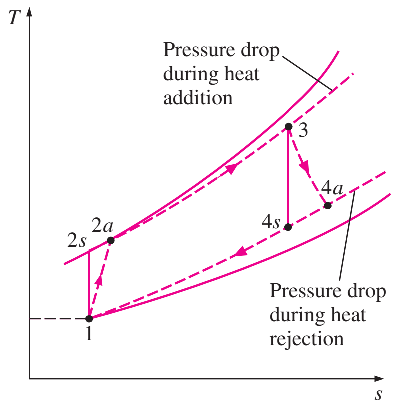

FIGURE 9–36
The deviation of an actual gas-turbine cycle from the ideal Brayton cycle as a result of irreversibilities.

EXAMPLE 9–6 An Actual Gas-Turbine Cycle

Assuming a compressor efficiency of 80 percent and a turbine efficiency of 85 percent, determine (*a*) the back work ratio, (*b*) the thermal efficiency, and (*c*) the turbine exit temperature of the gas-turbine cycle discussed in Example 9–5.

Solution The Brayton cycle discussed in Example 9–5 is reconsidered. For specified turbine and compressor efficiencies, the back work ratio, the thermal efficiency, and the turbine exit temperature are to be determined.

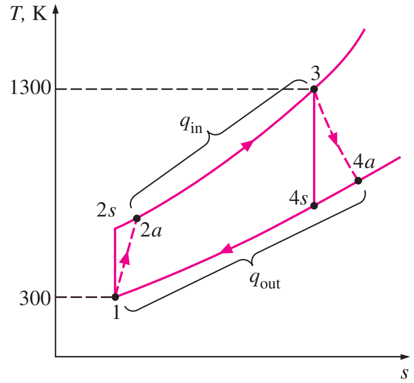

FIGURE 9–37

T-s diagram of the gas-turbine cycle discussed in Example 9–6.

Analysis (*a*) The T-s diagram of the cycle is shown in Fig. 9–37. The actual compressor work and turbine work are determined by using the definitions of compressor and turbine efficiencies, Eqs. 9–19 and 9–20:

Compressor:
$$w_{\text{comp,in}} = \frac{w_s}{\eta_C} = \frac{244.16 \text{ kJ/kg}}{0.80} = 305.20 \text{ kJ/kg}$$

Turbine:
$$w_{\text{turb,out}} = \eta_T w_s = (0.85)(606.60 \text{ kJ/kg}) = 515.61 \text{ kJ/kg}$$

Thus,

$$r_{\text{bw}} = \frac{w_{\text{comp,in}}}{w_{\text{turb,out}}} = \frac{305.20 \text{ kJ/kg}}{515.61 \text{ kJ/kg}} = \mathbf{0.592}$$

That is, the compressor is now consuming 59.2 percent of the work produced by the turbine (up from 40.3 percent). This increase is due to the irreversibilities that occur within the compressor and the turbine.

(*b*) In this case, air leaves the compressor at a higher temperature and enthalpy, which are determined to be

$$\begin{aligned} w_{\text{comp,in}} = h_{2a} - h_1 \rightarrow h_{2a} &= h_1 + w_{\text{comp,in}} \\ &= 300.19 + 305.20 \\ &= 605.39 \text{ kJ/kg} \quad (\text{and } T_{2a} = 598 \text{ K}) \end{aligned}$$

Thus,

$$q_{\text{in}} = h_3 - h_{2a} = 1395.97 - 605.39 = 790.58 \text{ kJ/kg}$$
$$w_{\text{net}} = w_{\text{out}} - w_{\text{in}} = 515.61 - 305.20 = 210.41 \text{ kJ/kg}$$

and

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{210.41 \text{ kJ/kg}}{790.58 \text{ kJ/kg}} = \mathbf{0.266 \text{ or } 26.6\%}$$

That is, the irreversibilities occurring within the turbine and compressor caused the thermal efficiency of the gas turbine cycle to drop from 42.6 to 26.6 percent. This example shows how sensitive the performance of a gas-turbine power plant is to the efficiencies of the compressor and the turbine. In fact, gas-turbine efficiencies did not reach competitive values until significant improvements were made in the design of gas turbines and compressors.

(*c*) The air temperature at the turbine exit is determined from an energy balance on the turbine:

$$\begin{aligned} w_{\text{turb,out}} = h_3 - h_{4a} \rightarrow h_{4a} &= h_3 - w_{\text{turb,out}} \\ &= 1395.97 - 515.61 \\ &= 880.36 \text{ kJ/kg} \end{aligned}$$

Then, from Table A–17,

$$T_{4a} = \mathbf{853 \text{ K}}$$

Discussion The temperature at turbine exit is considerably higher than that at the compressor exit (T_{2a} = 598 K), which suggests the use of regeneration to reduce fuel cost.

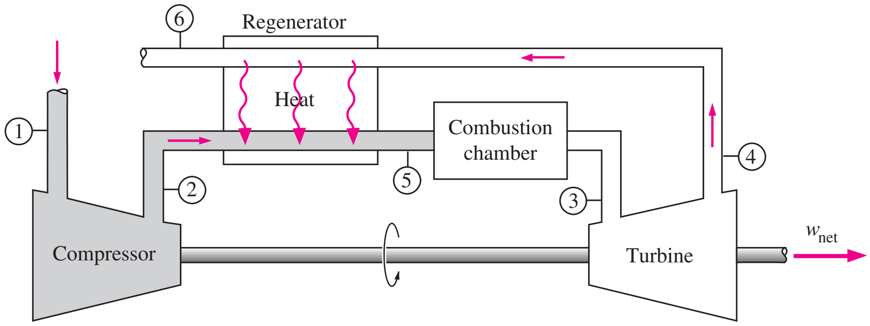

FIGURE 9–38

A gas-turbine engine with regenerator.

9–9 ▪ THE BRAYTON CYCLE WITH REGENERATION

In gas-turbine engines, the temperature of the exhaust gas leaving the turbine is often considerably higher than the temperature of the air leaving the compressor. Therefore, the high-pressure air leaving the compressor can be heated by transferring heat to it from the hot exhaust gases in a counter-flow heat exchanger, which is also known as a *regenerator* or a *recuperator.* A sketch of the gas-turbine engine utilizing a regenerator and the *T-s* diagram of the new cycle are shown in Figs. 9–38 and 9–39, respectively.

The thermal efficiency of the Brayton cycle increases as a result of regeneration since the portion of energy of the exhaust gases that is normally rejected to the surroundings is now used to preheat the air entering the combustion chamber. This, in turn, decreases the heat input (thus fuel) requirements for the same net work output. Note, however, that the use of a regenerator is recommended only when the turbine exhaust temperature is higher than the compressor exit temperature. Otherwise, heat will flow in the reverse direction (*to* the exhaust gases), decreasing the efficiency. This situation is encountered in gas-turbine engines operating at very high pressure ratios.

The highest temperature occurring within the regenerator is T_4, the temperature of the exhaust gases leaving the turbine and entering the regenerator. Under no conditions can the air be preheated in the regenerator to a temperature above this value. Air normally leaves the regenerator at a lower temperature, T_5. In the limiting (ideal) case, the air exits the regenerator at the inlet temperature of the exhaust gases T_4. Assuming the regenerator to be well insulated and any changes in kinetic and potential energies to be negligible, the actual and maximum heat transfers from the exhaust gases to the air can be expressed as

$$q_{\text{regen,act}} = h_5 - h_2 \tag{9–21}$$

and

$$q_{\text{regen,max}} = h_{5'} - h_2 = h_4 - h_2 \tag{9–22}$$

The extent to which a regenerator approaches an ideal regenerator is called the **effectiveness** $\boldsymbol{\epsilon}$ and is defined as

$$\epsilon = \frac{q_{\text{regen,act}}}{q_{\text{regen,max}}} = \frac{h_5 - h_2}{h_4 - h_2} \tag{9–23}$$

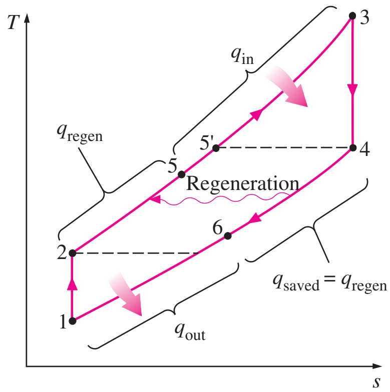

FIGURE 9–39

T-s diagram of a Brayton cycle with regeneration.

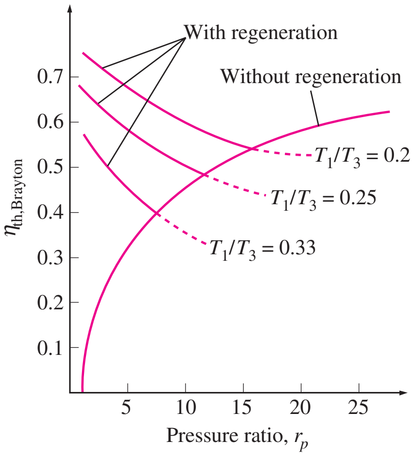

FIGURE 9–40

Thermal efficiency of the ideal Brayton cycle with and without regeneration.

When the cold-air-standard assumptions are utilized, it reduces to

$$\epsilon \cong \frac{T_5 - T_2}{T_4 - T_2} \tag{9–24}$$

A regenerator with a higher effectiveness obviously saves a greater amount of fuel since it preheats the air to a higher temperature prior to combustion. However, achieving a higher effectiveness requires the use of a larger regenerator, which carries a higher price tag and causes a larger pressure drop. Therefore, the use of a regenerator with a very high effectiveness cannot be justified economically unless the savings from the fuel costs exceed the additional expenses involved. The effectiveness of most regenerators used in practice is below 0.85.

Under the cold-air-standard assumptions, the thermal efficiency of an ideal Brayton cycle with regeneration is

$$\eta_{\text{th,regen}} = 1 - \left(\frac{T_1}{T_3}\right)(r_p)^{(k-1)/k} \tag{9–25}$$

Therefore, the thermal efficiency of an ideal Brayton cycle with regeneration depends on the ratio of the minimum to maximum temperatures as well as the pressure ratio. The thermal efficiency is plotted in Fig. 9–40 for various pressure ratios and minimum-to-maximum temperature ratios. This figure shows that regeneration is most effective at lower pressure ratios and low minimum-to-maximum temperature ratios.

EXAMPLE 9–7 Actual Gas-Turbine Cycle with Regeneration

Determine the thermal efficiency of the gas-turbine described in Example 9–6 if a regenerator having an effectiveness of 80 percent is installed.

Solution The gas-turbine discussed in Example 9–6 is equipped with a regenerator. For a specified effectiveness, the thermal efficiency is to be determined.

Analysis The *T-s* diagram of the cycle is shown in Fig. 9–41. We first determine the enthalpy of the air at the exit of the regenerator, using the definition of effectiveness:

$$\epsilon = \frac{h_5 - h_{2a}}{h_{4a} - h_{2a}}$$

$$0.80 = \frac{(h_5 - 605.39)\text{ kJ/kg}}{(880.36 - 605.39)\text{ kJ/kg}} \rightarrow h_5 = 825.37\text{ kJ/kg}$$

Thus,

$$q_{\text{in}} = h_3 - h_5 = (1395.97 - 825.37)\text{ kJ/kg} = 570.60\text{ kJ/kg}$$

This represents a savings of 220.0 kJ/kg from the heat input requirements. The addition of a regenerator (assumed to be frictionless) does not affect the net work output. Thus,

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{210.41\text{ kJ/kg}}{570.60\text{ kJ/kg}} = \mathbf{0.369 \text{ or } 36.9\%}$$

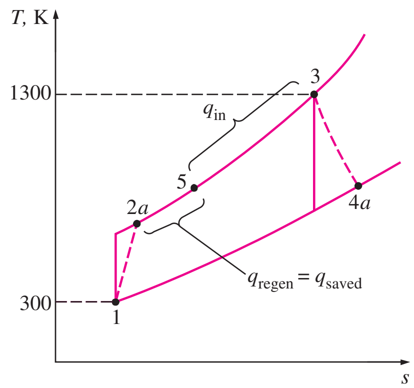

FIGURE 9–41

T-s diagram of the regenerative Brayton cycle described in Example 9–7.

Discussion Note that the thermal efficiency of the gas turbine has gone up from 26.6 to 36.9 percent as a result of installing a regenerator that helps to recuperate some of the thermal energy of the exhaust gases.

9–10 ▪ THE BRAYTON CYCLE WITH INTERCOOLING, REHEATING, AND REGENERATION

INTERACTIVE TUTORIAL

SEE TUTORIAL CH. 9, SEC. 5 ON THE DVD.

The net work of a gas-turbine cycle is the difference between the turbine work output and the compressor work input, and it can be increased by either decreasing the compressor work or increasing the turbine work, or both. It was shown in Chap. 7 that the work required to compress a gas between two specified pressures can be decreased by carrying out the compression process in stages and cooling the gas in between (Fig. 9–42)—that is, using *multistage compression with intercooling*. As the number of stages is increased, the compression process becomes nearly isothermal at the compressor inlet temperature, and the compression work decreases.

Likewise, the work output of a turbine operating between two pressure levels can be increased by expanding the gas in stages and reheating it in between—that is, utilizing *multistage expansion with reheating*. This is accomplished without raising the maximum temperature in the cycle. As the number of stages is increased, the expansion process becomes nearly isothermal. The foregoing argument is based on a simple principle: *The steady-flow compression or expansion work is proportional to the specific volume of the fluid. Therefore, the specific volume of the working fluid should be as low as possible during a compression process and as high as possible during an expansion process.* This is precisely what intercooling and reheating accomplish.

Combustion in gas turbines typically occurs at four times the amount of air needed for complete combustion to avoid excessive temperatures. Therefore, the exhaust gases are rich in oxygen, and reheating can be accomplished by simply spraying additional fuel into the exhaust gases between two expansion states.

The working fluid leaves the compressor at a lower temperature, and the turbine at a higher temperature, when intercooling and reheating are utilized. This makes regeneration more attractive since a greater potential for regeneration exists. Also, the gases leaving the compressor can be heated to a higher temperature before they enter the combustion chamber because of the higher temperature of the turbine exhaust.

A schematic of the physical arrangement and the *T-s* diagram of an ideal two-stage gas-turbine cycle with intercooling, reheating, and regeneration are shown in Figs. 9–43 and 9–44. The gas enters the first stage of the compressor at state 1, is compressed isentropically to an intermediate pressure P_2, is cooled at constant pressure to state 3 ($T_3 = T_1$), and is compressed in the second stage isentropically to the final pressure P_4. At state 4 the gas enters the regenerator, where it is heated to T_5 at constant pressure. In an ideal regenerator, the gas leaves the regenerator at the temperature of the turbine exhaust, that is, $T_5 = T_9$. The primary heat addition (or combustion) process takes

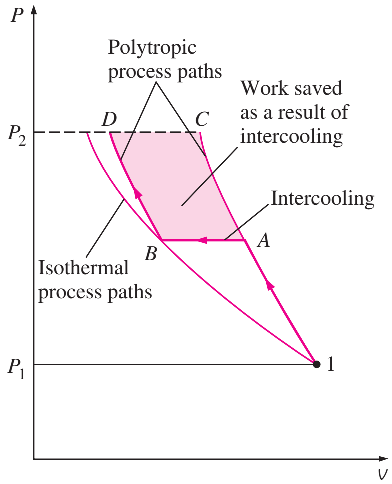

FIGURE 9–42

Comparison of work inputs to a single-stage compressor (1*AC*) and a two-stage compressor with intercooling (1*ABD*).

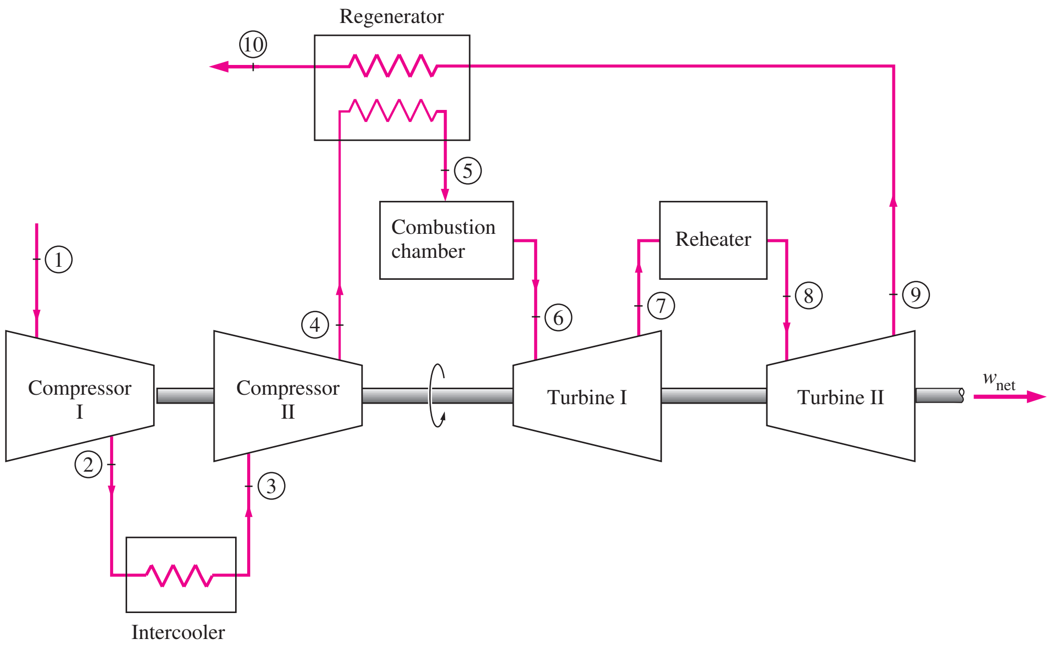

FIGURE 9–43

A gas-turbine engine with two-stage compression with intercooling, two-stage expansion with reheating, and regeneration.

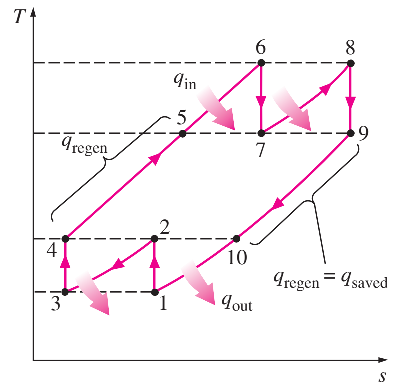

FIGURE 9–44

T-s diagram of an ideal gas-turbine cycle with intercooling, reheating, and regeneration.

place between states 5 and 6. The gas enters the first stage of the turbine at state 6 and expands isentropically to state 7, where it enters the reheater. It is reheated at constant pressure to state 8 ($T_8 = T_6$), where it enters the second stage of the turbine. The gas exits the turbine at state 9 and enters the regenerator, where it is cooled to state 10 at constant pressure. The cycle is completed by cooling the gas to the initial state (or purging the exhaust gases).

It was shown in Chap. 7 that the work input to a two-stage compressor is minimized when equal pressure ratios are maintained across each stage. It can be shown that this procedure also maximizes the turbine work output. Thus, for best performance we have

$$\frac{P_2}{P_1} = \frac{P_4}{P_3} \quad \text{and} \quad \frac{P_6}{P_7} = \frac{P_8}{P_9} \tag{9–26}$$

In the analysis of the actual gas-turbine cycles, the irreversibilities that are present within the compressor, the turbine, and the regenerator as well as the pressure drops in the heat exchangers should be taken into consideration.

The back work ratio of a gas-turbine cycle improves as a result of intercooling and reheating. However, this does not mean that the thermal efficiency also improves. The fact is, intercooling and reheating always decreases the thermal efficiency unless they are accompanied by regeneration. This is because intercooling decreases the average temperature at which heat is added, and reheating increases the average temperature at which heat is rejected. This is also apparent from Fig. 9–44. Therefore, in gas-turbine power plants, intercooling and reheating are always used in conjunction with regeneration.

If the number of compression and expansion stages is increased, the ideal gas-turbine cycle with intercooling, reheating, and regeneration approaches the Ericsson cycle, as illustrated in Fig. 9–45, and the thermal efficiency approaches the theoretical limit (the Carnot efficiency). However, the contribution of each additional stage to the thermal efficiency is less and less, and the use of more than two or three stages cannot be justified economically.

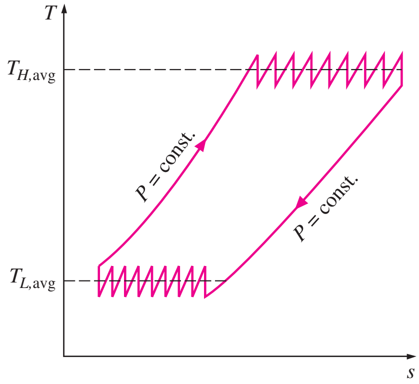

FIGURE 9–45

As the number of compression and expansion stages increases, the gas-turbine cycle with intercooling, reheating, and regeneration approaches the Ericsson cycle.

EXAMPLE 9–8 A Gas Turbine with Reheating and Intercooling

An ideal gas-turbine cycle with two stages of compression and two stages of expansion has an overall pressure ratio of 8. Air enters each stage of the compressor at 300 K and each stage of the turbine at 1300 K. Determine the back work ratio and the thermal efficiency of this gas-turbine cycle, assuming (*a*) no regenerators and (*b*) an ideal regenerator with 100 percent effectiveness. Compare the results with those obtained in Example 9–5.

Solution An ideal gas-turbine cycle with two stages of compression and two stages of expansion is considered. The back work ratio and the thermal efficiency of the cycle are to be determined for the cases of no regeneration and maximum regeneration.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible.

Analysis The *T-s* diagram of the ideal gas-turbine cycle described is shown in Fig. 9–46. We note that the cycle involves two stages of expansion, two stages of compression, and regeneration.

For two-stage compression and expansion, the work input is minimized and the work output is maximized when both stages of the compressor and the turbine have the same pressure ratio. Thus,

$$\frac{P_2}{P_1} = \frac{P_4}{P_3} = \sqrt{8} = 2.83 \quad \text{and} \quad \frac{P_6}{P_7} = \frac{P_8}{P_9} = \sqrt{8} = 2.83$$

Air enters each stage of the compressor at the same temperature, and each stage has the same isentropic efficiency (100 percent in this case). Therefore, the temperature (and enthalpy) of the air at the exit of each compression stage will be the same. A similar argument can be given for the turbine. Thus,

$$\textit{At inlets:} \qquad T_1 = T_3, \quad h_1 = h_3 \quad \text{and} \quad T_6 = T_8, \quad h_6 = h_8$$

$$\textit{At exits:} \qquad T_2 = T_4, \quad h_2 = h_4 \quad \text{and} \quad T_7 = T_9, \quad h_7 = h_9$$

Under these conditions, the work input to each stage of the compressor will be the same, and so will the work output from each stage of the turbine.

(*a*) In the absence of any regeneration, the back work ratio and the thermal efficiency are determined by using data from Table A–17 as follows:

$$T_1 = 300 \text{ K} \rightarrow h_1 = 300.19 \text{ kJ/kg}$$

$$P_{r1} = 1.386$$

$$P_{r2} = \frac{P_2}{P_1} P_{r1} = \sqrt{8}(1.386) = 3.92 \rightarrow T_2 = 403.3 \text{ K}$$

$$h_2 = 404.31 \text{ kJ/kg}$$

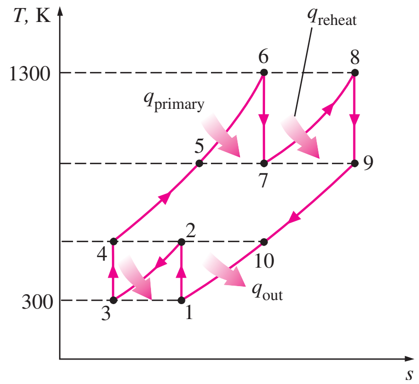

FIGURE 9–46

T-s diagram of the gas-turbine cycle discussed in Example 9–8.

$$T_6 = 1300 \text{ K} \rightarrow h_6 = 1395.97 \text{ kJ/kg}$$
$$P_{r6} = 330.9$$
$$P_{r7} = \frac{P_7}{P_6}P_{r6} = \frac{1}{\sqrt{8}}(330.9) = 117.0 \rightarrow T_7 = 1006.4 \text{ K}$$
$$h_7 = 1053.33 \text{ kJ/kg}$$

Then

$$w_{\text{comp,in}} = 2(w_{\text{comp,in,I}}) = 2(h_2 - h_1) = 2(404.31 - 300.19) = 208.24 \text{ kJ/kg}$$
$$w_{\text{turb,out}} = 2(w_{\text{turb,out,I}}) = 2(h_6 - h_7) = 2(1395.97 - 1053.33) = 685.28 \text{ kJ/kg}$$
$$w_{\text{net}} = w_{\text{turb,out}} - w_{\text{comp,in}} = 685.28 - 208.24 = 477.04 \text{ kJ/kg}$$
$$q_{\text{in}} = q_{\text{primary}} + q_{\text{reheat}} = (h_6 - h_4) + (h_8 - h_7)$$
$$= (1395.97 - 404.31) + (1395.97 - 1053.33) = 1334.30 \text{ kJ/kg}$$

Thus,

$$r_{\text{bw}} = \frac{w_{\text{comp,in}}}{w_{\text{turb,out}}} = \frac{208.24 \text{ kJ/kg}}{685.28 \text{ kJ/kg}} = \mathbf{0.304 \text{ or } 30.4\%}$$

and

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{477.04 \text{ kJ/kg}}{1334.30 \text{ kJ/kg}} = \mathbf{0.358 \text{ or } 35.8\%}$$

A comparison of these results with those obtained in Example 9–5 (single-stage compression and expansion) reveals that multistage compression with intercooling and multistage expansion with reheating improve the back work ratio (it drops from 40.3 to 30.4 percent) but hurt the thermal efficiency (it drops from 42.6 to 35.8 percent). Therefore, intercooling and reheating are not recommended in gas-turbine power plants unless they are accompanied by regeneration.

(*b*) The addition of an ideal regenerator (no pressure drops, 100 percent effectiveness) does not affect the compressor work and the turbine work. Therefore, the net work output and the back work ratio of an ideal gas-turbine cycle are identical whether there is a regenerator or not. A regenerator, however, reduces the heat input requirements by preheating the air leaving the compressor, using the hot exhaust gases. In an ideal regenerator, the compressed air is heated to the turbine exit temperature T_9 before it enters the combustion chamber. Thus, under the air-standard assumptions, $h_5 = h_7 = h_9$.

The heat input and the thermal efficiency in this case are

$$q_{\text{in}} = q_{\text{primary}} + q_{\text{reheat}} = (h_6 - h_5) + (h_8 - h_7)$$
$$= (1395.97 - 1053.33) + (1395.97 - 1053.33) = 685.28 \text{ kJ/kg}$$

and

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{477.04 \text{ kJ/kg}}{685.28 \text{ kJ/kg}} = \mathbf{0.696 \text{ or } 69.6\%}$$

Discussion Note that the thermal efficiency almost doubles as a result of regeneration compared to the no-regeneration case. The overall effect of two-stage compression and expansion with intercooling, reheating, and regeneration on the thermal efficiency is an increase of 63 percent. As the number of compression and expansion stages is increased, the cycle will approach the Ericsson cycle, and the thermal efficiency will approach

$$\eta_{\text{th,Ericsson}} = \eta_{\text{th,Carnot}} = 1 - \frac{T_L}{T_H} = 1 - \frac{300\text{ K}}{1300\text{ K}} = 0.769$$

Adding a second stage increases the thermal efficiency from 42.6 to 69.6 percent, an increase of 27 percentage points. This is a significant increase in efficiency, and usually it is well worth the extra cost associated with the second stage. Adding more stages, however (no matter how many), can increase the efficiency an additional 7.3 percentage points at most, and usually cannot be justified economically.

9–11 ▪ IDEAL JET-PROPULSION CYCLES

Gas-turbine engines are widely used to power aircraft because they are light and compact and have a high power-to-weight ratio. Aircraft gas turbines operate on an open cycle called a **jet-propulsion cycle.** The ideal jet-propulsion cycle differs from the simple ideal Brayton cycle in that the gases are not expanded to the ambient pressure in the turbine. Instead, they are expanded to a pressure such that the power produced by the turbine is just sufficient to drive the compressor and the auxiliary equipment, such as a small generator and hydraulic pumps. That is, the net work output of a jet-propulsion cycle is zero. The gases that exit the turbine at a relatively high pressure are subsequently accelerated in a nozzle to provide the thrust to propel the aircraft (Fig. 9–47). Also, aircraft gas turbines operate at higher pressure ratios (typically between 10 and 25), and the fluid passes through a diffuser first, where it is decelerated and its pressure is increased before it enters the compressor.

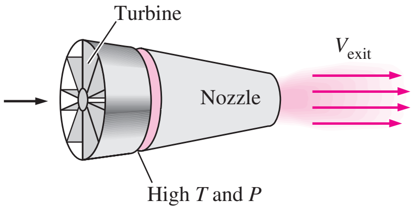

FIGURE 9–47

In jet engines, the high-temperature and high-pressure gases leaving the turbine are accelerated in a nozzle to provide thrust.

Aircraft are propelled by accelerating a fluid in the opposite direction to motion. This is accomplished by either slightly accelerating a large mass of fluid (*propeller-driven engine*) or greatly accelerating a small mass of fluid (*jet* or *turbojet engine*) or both (*turboprop engine*).

A schematic of a turbojet engine and the *T-s* diagram of the ideal turbojet cycle are shown in Fig. 9–48. The pressure of air rises slightly as it is decelerated in the diffuser. Air is compressed by the compressor. It is mixed with fuel in the combustion chamber, where the mixture is burned at constant pressure. The high-pressure and high-temperature combustion gases partially expand in the turbine, producing enough power to drive the compressor and other equipment. Finally, the gases expand in a nozzle to the ambient pressure and leave the engine at a high velocity.

In the ideal case, the turbine work is assumed to equal the compressor work. Also, the processes in the diffuser, the compressor, the turbine, and the nozzle are assumed to be isentropic. In the analysis of actual cycles, however, the irreversibilities associated with these devices should be considered. The effect of the irreversibilities is to reduce the thrust that can be obtained from a turbojet engine.

The **thrust** developed in a turbojet engine is the unbalanced force that is caused by the difference in the momentum of the low-velocity air entering the engine and the high-velocity exhaust gases leaving the engine, and it is

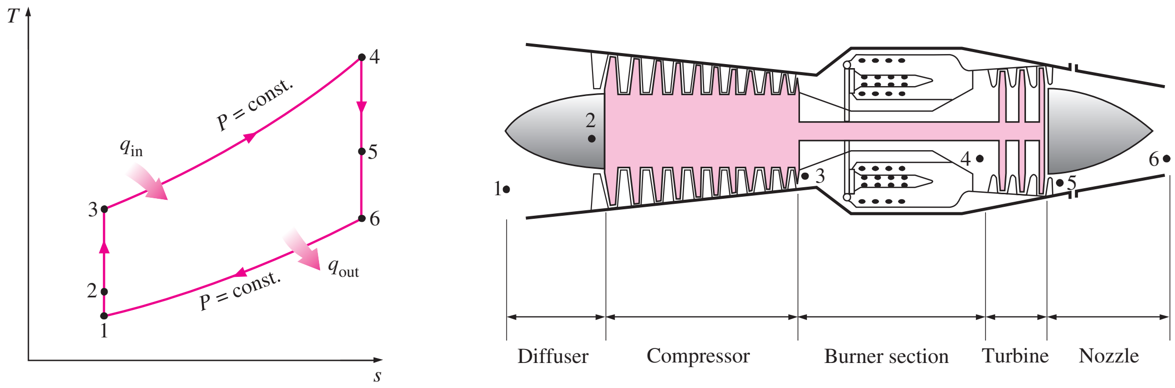

FIGURE 9–48

Basic components of a turbojet engine and the *T-s* diagram for the ideal turbojet cycle.

Source: The Aircraft Gas Turbine Engine and Its Operation. © United Aircraft Corporation (now United Technologies Corp.), 1951, 1974.

determined from Newton's second law. The pressures at the inlet and the exit of a turbojet engine are identical (the ambient pressure); thus, the net thrust developed by the engine is

$$F = (\dot{m}V)_{\text{exit}} - (\dot{m}V)_{\text{inlet}} = \dot{m}(V_{\text{exit}} - V_{\text{inlet}}) \qquad \text{(N)} \tag{9–27}$$

where V_{exit} is the exit velocity of the exhaust gases and V_{inlet} is the inlet velocity of the air, both relative to the aircraft. Thus, for an aircraft cruising in still air, V_{inlet} is the aircraft velocity. In reality, the mass flow rates of the gases at the engine exit and the inlet are different, the difference being equal to the combustion rate of the fuel. However, the air–fuel mass ratio used in jet-propulsion engines is usually very high, making this difference very small. Thus, $\dot{m}$ in Eq. 9–27 is taken as the mass flow rate of air through the engine. For an aircraft cruising at a constant speed, the thrust is used to overcome air drag, and the net force acting on the body of the aircraft is zero. Commercial airplanes save fuel by flying at higher altitudes during long trips since air at higher altitudes is thinner and exerts a smaller drag force on aircraft.

The power developed from the thrust of the engine is called the **propulsive power** $\dot{W}_P$, which is the *propulsive force* (*thrust*) times the *distance* this force acts on the aircraft per unit time, that is, the thrust times the aircraft velocity (Fig. 9–49):

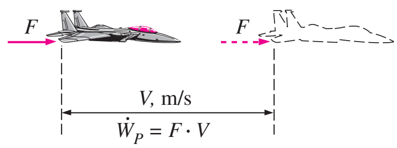

FIGURE 9–49

Propulsive power is the thrust acting on the aircraft through a distance per unit time.

$$\dot{W}_P = FV_{\text{aircraft}} = \dot{m}(V_{\text{exit}} - V_{\text{inlet}})V_{\text{aircraft}} \qquad \text{(kW)} \tag{9–28}$$

The net work developed by a turbojet engine is zero. Thus, we cannot define the efficiency of a turbojet engine in the same way as stationary gas-turbine engines. Instead, we should use the general definition of efficiency, which is the ratio of the desired output to the required input. The desired output in a turbojet engine is the *power produced* to propel the aircraft $\dot{W}_P$, and the required input is the *heating value of the fuel* $\dot{Q}_{\text{in}}$. The ratio of these two quantities is called the **propulsive efficiency** and is given by

$$\eta_P = \frac{\text{Propulsive power}}{\text{Energy input rate}} = \frac{\dot{W}_P}{\dot{Q}_{\text{in}}} \tag{9–29}$$

Propulsive efficiency is a measure of how efficiently the thermal energy released during the combustion process is converted to propulsive energy. The remaining part of the energy released shows up as the kinetic energy of the exhaust gases relative to a fixed point on the ground and as an increase in the enthalpy of the gases leaving the engine.

EXAMPLE 9–9 The Ideal Jet-Propulsion Cycle

A turbojet aircraft flies with a velocity of 850 ft/s at an altitude where the air is at 5 psia and −40°F. The compressor has a pressure ratio of 10, and the temperature of the gases at the turbine inlet is 2000°F. Air enters the compressor at a rate of 100 lbm/s. Utilizing the cold-air-standard assumptions, determine (*a*) the temperature and pressure of the gases at the turbine exit, (*b*) the velocity of the gases at the nozzle exit, and (*c*) the propulsive efficiency of the cycle.

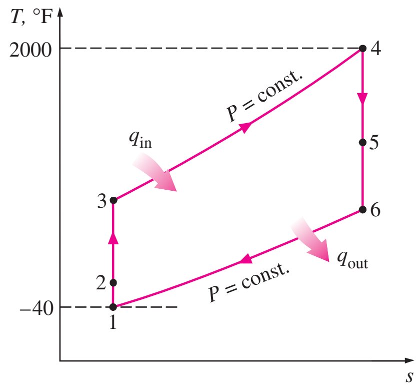

FIGURE 9–50

T-s diagram for the turbojet cycle described in Example 9–9.

Solution The operating conditions of a turbojet aircraft are specified. The temperature and pressure at the turbine exit, the velocity of gases at the nozzle exit, and the propulsive efficiency are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The cold-air-standard assumptions are applicable and thus air can be assumed to have constant specific heats at room temperature (c_p = 0.240 Btu/lbm · °F and k = 1.4). **3** Kinetic and potential energies are negligible, except at the diffuser inlet and the nozzle exit. **4** The turbine work output is equal to the compressor work input.

Analysis The *T-s* diagram of the ideal jet propulsion cycle described is shown in Fig. 9–50. We note that the components involved in the jet-propulsion cycle are steady-flow devices.

(*a*) Before we can determine the temperature and pressure at the turbine exit, we need to find the temperatures and pressures at other states:

Process 1-2 (isentropic compression of an ideal gas in a diffuser): For convenience, we can assume that the aircraft is stationary and the air is moving toward the aircraft at a velocity of V_1 = 850 ft/s. Ideally, the air exits the diffuser with a negligible velocity ($V_2 \cong 0$):

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

$$0 = c_p(T_2 - T_1) - \frac{V_1^2}{2}$$

$$T_2 = T_1 + \frac{V_1^2}{2c_p}$$

$$= 420\text{ R} + \frac{(850\text{ ft/s})^2}{2(0.240\text{ Btu/lbm}\cdot\text{R})}\left(\frac{1\text{ Btu/lbm}}{25{,}037\text{ ft}^2/\text{s}^2}\right)$$

$$= 480\text{ R}$$

$$P_2 = P_1\left(\frac{T_2}{T_1}\right)^{k/(k-1)} = (5\text{ psia})\left(\frac{480\text{ R}}{420\text{ R}}\right)^{1.4/(1.4-1)} = 8.0\text{ psia}$$

Process 2-3 (isentropic compression of an ideal gas in a compressor):

$$P_3 = (r_p)(P_2) = (10)(8.0\text{ psia}) = 80\text{ psia } (= P_4)$$

$$T_3 = T_2\left(\frac{P_3}{P_2}\right)^{(k-1)/k} = (480\text{ R})(10)^{(1.4-1)/1.4} = 927\text{ R}$$

Process 4-5 (isentropic expansion of an ideal gas in a turbine): Neglecting the kinetic energy changes across the compressor and the turbine and assuming the turbine work to be equal to the compressor work, we find the temperature and pressure at the turbine exit to be

$$w_{\text{comp,in}} = w_{\text{turb,out}}$$

$$h_3 - h_2 = h_4 - h_5$$

$$c_p(T_3 - T_2) = c_p(T_4 - T_5)$$

$$T_5 = T_4 - T_3 + T_2 = 2460 - 927 + 480 = \mathbf{2013\ R}$$

$$P_5 = P_4\left(\frac{T_5}{T_4}\right)^{k/(k-1)} = (80\text{ psia})\left(\frac{2013\text{ R}}{2460\text{ R}}\right)^{1.4/(1.4-1)} = \mathbf{39.7\ psia}$$

(*b*) To find the air velocity at the nozzle exit, we need to first determine the nozzle exit temperature and then apply the steady-flow energy equation.

Process 5-6 (isentropic expansion of an ideal gas in a nozzle):

$$T_6 = T_5\left(\frac{P_6}{P_5}\right)^{(k-1)/k} = (2013\text{ R})\left(\frac{5\text{ psia}}{39.7\text{ psia}}\right)^{(1.4-1)/1.4} = 1114\text{ R}$$

$$h_6 + \frac{V_6^2}{2} = h_5 + \frac{V_5^2}{2}^{\nearrow 0}$$

$$0 = c_p(T_6 - T_5) + \frac{V_6^2}{2}$$

$$V_6 = \sqrt{2c_p(T_5 - T_6)}$$

$$= \sqrt{2(0.240\text{ Btu/lbm}\cdot\text{R})[(2013 - 1114)\text{ R}]\left(\frac{25{,}037\text{ ft}^2/\text{s}^2}{1\text{ Btu/lbm}}\right)}$$

$$= \mathbf{3288\ ft/s}$$

(*c*) The propulsive efficiency of a turbojet engine is the ratio of the propulsive power developed $\dot{W}_P$ to the total heat transfer rate to the working fluid:

$$\dot{W}_P = \dot{m}(V_{\text{exit}} - V_{\text{inlet}})V_{\text{aircraft}}$$

$$= (100\text{ lbm/s})[(3288 - 850)\text{ ft/s}](850\text{ ft/s})\left(\frac{1\text{ Btu/lbm}}{25{,}037\text{ ft}^2/\text{s}^2}\right)$$

$$= 8276\text{ Btu/s} \quad (\text{or } 11{,}707\text{ hp})$$

$$\dot{Q}_{\text{in}} = \dot{m}(h_4 - h_3) = \dot{m}c_p(T_4 - T_3)$$

$$= (100\text{ lbm/s})(0.240\text{ Btu/lbm}\cdot\text{R})[(2460 - 927)\text{ R}]$$

$$= 36{,}794\text{ Btu/s}$$

$$\eta_P = \frac{\dot{W}_P}{\dot{Q}_{\text{in}}} = \frac{8276\text{ Btu/s}}{36{,}794\text{ Btu/s}} = \mathbf{22.5\%}$$

That is, 22.5 percent of the energy input is used to propel the aircraft and to overcome the drag force exerted by the atmospheric air.

Discussion For those who are wondering what happened to the rest of the energy, here is a brief account:

$$\dot{KE}_{out} = \dot{m}\frac{V_g^2}{2} = (100 \text{ lbm/s})\left\{\frac{[(3288 - 850)\text{ft/s}]^2}{2}\right\}\left(\frac{1 \text{ Btu/lbm}}{25{,}037 \text{ ft}^2/\text{s}^2}\right)$$

$$= 11{,}867 \text{ Btu/s} \qquad (32.2\%)$$

$$\dot{Q}_{out} = \dot{m}(h_6 - h_1) = \dot{m}c_p(T_6 - T_1)$$

$$= (100 \text{ lbm/s})(0.24 \text{ Btu/lbm}\cdot\text{R})[(1114 - 420)\text{ R}]$$

$$= 16{,}651 \text{ Btu/s} \qquad (45.3\%)$$

Thus, 32.2 percent of the energy shows up as excess kinetic energy (kinetic energy of the gases relative to a fixed point on the ground). Notice that for the highest propulsion efficiency, the velocity of the exhaust gases relative to the ground V_g should be zero. That is, the exhaust gases should leave the nozzle at the velocity of the aircraft. The remaining 45.3 percent of the energy shows up as an increase in enthalpy of the gases leaving the engine. These last two forms of energy eventually become part of the internal energy of the atmospheric air (Fig. 9–51).

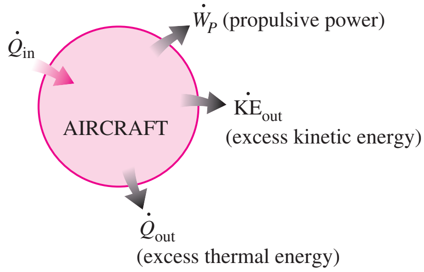

FIGURE 9–51

Energy supplied to an aircraft (from the burning of a fuel) manifests itself in various forms.

Modifications to Turbojet Engines

The first airplanes built were all propeller-driven, with propellers powered by engines essentially identical to automobile engines. The major breakthrough in commercial aviation occurred with the introduction of the turbojet engine in 1952. Both propeller-driven engines and jet-propulsion-driven engines have their own strengths and limitations, and several attempts have been made to combine the desirable characteristics of both in one engine. Two such modifications are the *propjet engine* and the *turbofan engine.*

The most widely used engine in aircraft propulsion is the **turbofan** (or *fanjet*) engine wherein a large fan driven by the turbine forces a considerable amount of air through a duct (cowl) surrounding the engine, as shown in Figs. 9–52 and 9–53. The fan exhaust leaves the duct at a higher velocity, enhancing the total thrust of the engine significantly. A turbofan engine is based on the principle that for the same power, a large volume of slower-moving air produces more thrust than a small volume of fast-moving air. The first commercial turbofan engine was successfully tested in 1955.

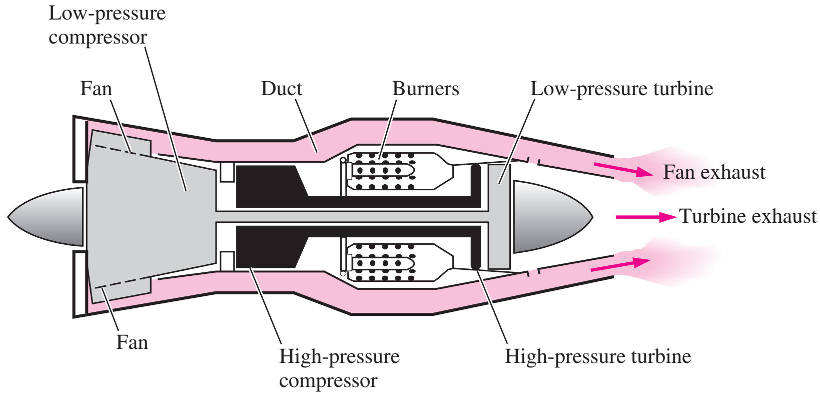

FIGURE 9–52

A turbofan engine.

Source: The Aircraft Gas Turbine and Its Operation. © United Aircraft Corporation (now United Technologies Corp.), 1951, 1974.

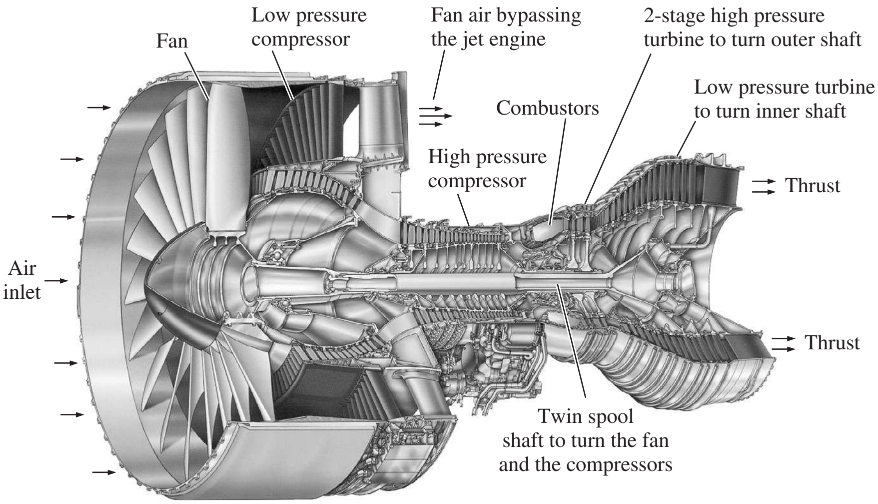

FIGURE 9–53

A modern jet engine used to power Boeing 777 aircraft. This is a Pratt & Whitney PW4084 turbofan capable of producing 84,000 pounds of thrust. It is 4.87 m (192 in.) long, has a 2.84 m (112 in.) diameter fan, and it weighs 6800 kg (15,000 lbm).

Courtesy of Pratt & Whitney Corp.

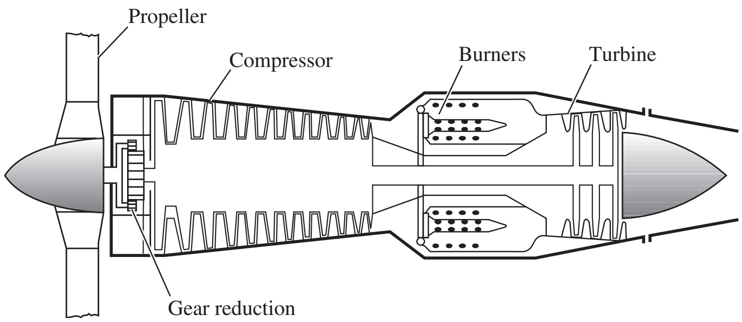

FIGURE 9–54

A turboprop engine.

Source: The Aircraft Gas Turbine Engine and Its Operation. © United Aircraft Corporation (now United Technologies Corp.), 1951, 1974.

The turbofan engine on an airplane can be distinguished from the less-efficient turbojet engine by its fat cowling covering the large fan. All the thrust of a turbojet engine is due to the exhaust gases leaving the engine at about twice the speed of sound. In a turbofan engine, the high-speed exhaust gases are mixed with the lower-speed air, which results in a considerable reduction in noise.

New cooling techniques have resulted in considerable increases in efficiencies by allowing gas temperatures at the burner exit to reach over 1500°C, which is more than 100°C above the melting point of the turbine blade materials. Turbofan engines deserve most of the credit for the success of jumbo jets that weigh almost 400,000 kg and are capable of carrying over 400 passengers for up to a distance of 10,000 km at speeds over 950 km/h with less fuel per passenger mile.

The ratio of the mass flow rate of air bypassing the combustion chamber to that of air flowing through it is called the *bypass ratio*. The first commercial high-bypass-ratio engines had a bypass ratio of 5. Increasing the bypass ratio of a turbofan engine increases thrust. Thus, it makes sense to remove the cowl from the fan. The result is a **propjet** engine, as shown in Fig. 9–54. Turbofan and propjet engines differ primarily in their bypass ratios: 5 or 6 for turbofans and as high as 100 for propjets. As a general rule, propellers

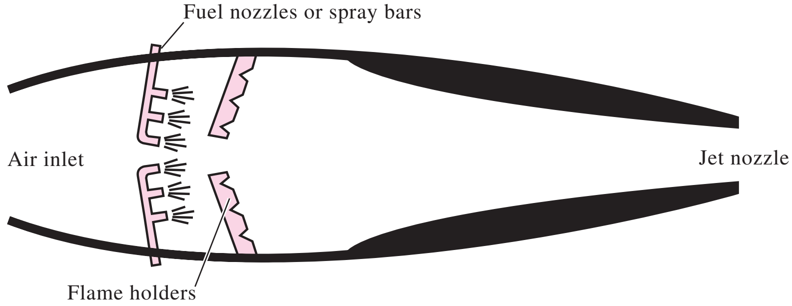

FIGURE 9–55

A ramjet engine.

Source: The Aircraft Gas Turbine Engine and Its Operation. © United Aircraft Corporation (now United Technologies Corp.), 1951, 1974.

are more efficient than jet engines, but they are limited to low-speed and low-altitude operation since their efficiency decreases at high speeds and altitudes. The old propjet engines (*turboprops*) were limited to speeds of about Mach 0.62 and to altitudes of around 9100 m. The new propjet engines (*propfans*) are expected to achieve speeds of about Mach 0.82 and altitudes of about 12,200 m. Commercial airplanes of medium size and range propelled by propfans are expected to fly as high and as fast as the planes propelled by turbofans, and to do so on less fuel.

Another modification that is popular in military aircraft is the addition of an **afterburner** section between the turbine and the nozzle. Whenever a need for extra thrust arises, such as for short takeoffs or combat conditions, additional fuel is injected into the oxygen-rich combustion gases leaving the turbine. As a result of this added energy, the exhaust gases leave at a higher velocity, providing a greater thrust.

A **ramjet** engine is a properly shaped duct with no compressor or turbine, as shown in Fig. 9–55, and is sometimes used for high-speed propulsion of missiles and aircraft. The pressure rise in the engine is provided by the ram effect of the incoming high-speed air being rammed against a barrier. Therefore, a ramjet engine needs to be brought to a sufficiently high speed by an external source before it can be fired.

The ramjet performs best in aircraft flying above Mach 2 or 3 (two or three times the speed of sound). In a ramjet, the air is slowed down to about Mach 0.2, fuel is added to the air and burned at this low velocity, and the combustion gases are expended and accelerated in a nozzle.

A **scramjet** engine is essentially a ramjet in which air flows through at supersonic speeds (above the speed of sound). Ramjets that convert to scramjet configurations at speeds above Mach 6 are successfully tested at speeds of about Mach 8.

Finally, a **rocket** is a device where a solid or liquid fuel and an oxidizer react in the combustion chamber. The high-pressure combustion gases are then expanded in a nozzle. The gases leave the rocket at very high velocities, producing the thrust to propel the rocket.

9–12 ▪ SECOND-LAW ANALYSIS OF GAS POWER CYCLES

The ideal Carnot, Ericsson, and Stirling cycles are *totally reversible*; thus they do not involve any irreversibilities. The ideal Otto, Diesel, and Brayton cycles, however, are only *internally reversible,* and they may involve irreversibilities external to the system. A second-law analysis of these cycles reveals where the largest irreversibilities occur and where to start improvements.

Relations for *exergy* and *exergy destruction* for both closed and steady-flow systems are developed in Chap. 8. The exergy destruction for a closed system can be expressed as

$$X_{\text{dest}} = T_0 S_{\text{gen}} = T_0(\Delta S_{\text{sys}} - S_{\text{in}} + S_{\text{out}})$$
$$= T_0\left[(S_2 - S_1)_{\text{sys}} - \frac{Q_{\text{in}}}{T_{b,\text{in}}} + \frac{Q_{\text{out}}}{T_{b,\text{out}}}\right] \quad \text{(kJ)} \tag{9–30}$$

where $T_{b,\text{in}}$ and $T_{b,\text{out}}$ are the temperatures of the system boundary where heat is transferred into and out of the system, respectively. A similar relation for steady-flow systems can be expressed, in rate form, as

$$\dot{X}_{\text{dest}} = T_0 \dot{S}_{\text{gen}} = T_0(\dot{S}_{\text{out}} - \dot{S}_{\text{in}}) = T_0\left(\sum_{\text{out}} \dot{m}s - \sum_{\text{in}} \dot{m}s - \frac{\dot{Q}_{\text{in}}}{T_{b,\text{in}}} + \frac{\dot{Q}_{\text{out}}}{T_{b,\text{out}}}\right) \quad \text{(kW)} \tag{9–31}$$

or, on a unit–mass basis for a one-inlet, one-exit steady-flow device, as

$$X_{\text{dest}} = T_0 s_{\text{gen}} = T_0\left(s_e - s_i - \frac{q_{\text{in}}}{T_{b,\text{in}}} + \frac{q_{\text{out}}}{T_{b,\text{out}}}\right) \quad \text{(kJ/kg)} \tag{9–32}$$

where subscripts i and e denote the inlet and exit states, respectively.

The exergy destruction of a *cycle* is the sum of the exergy destructions of the processes that compose that cycle. The exergy destruction of a cycle can also be determined without tracing the individual processes by considering the entire cycle as a single process and using one of the relations above. Entropy is a property, and its value depends on the state only. For a cycle, reversible or actual, the initial and the final states are identical; thus $s_e = s_i$. Therefore, the exergy destruction of a cycle depends on the magnitude of the heat transfer with the high- and low-temperature reservoirs involved and on their temperatures. It can be expressed on a unit–mass basis as

$$x_{\text{dest}} = T_0\left(\sum \frac{q_{\text{out}}}{T_{b,\text{out}}} - \sum \frac{q_{\text{in}}}{T_{b,\text{in}}}\right) \quad \text{(kJ/kg)} \tag{9–33}$$

For a cycle that involves heat transfer only with a source at T_H and a sink at T_L, the exergy destruction becomes

$$x_{\text{dest}} = T_0\left(\frac{q_{\text{out}}}{T_L} - \frac{q_{\text{in}}}{T_H}\right) \quad \text{(kJ/kg)} \tag{9–34}$$

The exergies of a closed system ϕ and a fluid stream ψ at any state can be determined from

$$\phi = (u - u_0) - T_0(s - s_0) + P_0(v - v_0) + \frac{V^2}{2} + gz \quad \text{(kJ/kg)} \tag{9–35}$$

and

$$\psi = (h - h_0) - T_0(s - s_0) + \frac{V^2}{2} + gz \quad \text{(kJ/kg)} \tag{9–36}$$

where subscript "0" denotes the state of the surroundings.

EXAMPLE 9–10 Second-Law Analysis of an Otto Cycle

Determine the exergy destruction associated with the Otto cycle (all four processes as well as the cycle) discussed in Example 9–2, assuming that heat is transferred to the working fluid from a source at 1700 K and heat is rejected to the surroundings at 290 K. Also, determine the exergy of the exhaust gases when they are purged.

Solution The Otto cycle analyzed in Example 9–2 is reconsidered. For specified source and sink temperatures, the exergy destruction associated with the cycle and the exergy purged with the exhaust gases are to be determined.

Analysis In Example 9–2, various quantities of interest were given or determined to be

$$r = 8 \qquad P_2 = 1.7997 \text{ MPa}$$
$$T_0 = 290 \text{ K} \qquad P_3 = 4.345 \text{ MPa}$$
$$T_1 = 290 \text{ K} \qquad q_{\text{in}} = 800 \text{ kJ/kg}$$
$$T_2 = 652.4 \text{ K} \qquad q_{\text{out}} = 381.83 \text{ kJ/kg}$$
$$T_3 = 1575.1 \text{ K} \qquad w_{\text{net}} = 418.17 \text{ kJ/kg}$$

Processes 1-2 and 3-4 are isentropic ($s_1 = s_2$, $s_3 = s_4$) and therefore do not involve any internal or external irreversibilities; that is, $X_{\text{dest},12} = 0$ and $X_{\text{dest},34} = 0$.

Processes 2-3 and 4-1 are constant-volume heat-addition and heat-rejection processes, respectively, and are internally reversible. However, the heat transfer between the working fluid and the source or the sink takes place through a finite temperature difference, rendering both processes irreversible. The exergy destruction associated with each process is determined from Eq. 9–32. However, first we need to determine the entropy change of air during these processes:

$$\begin{aligned} s_3 - s_2 &= s_3^\circ - s_2^\circ - R \ln \frac{P_3}{P_2} \\ &= (3.5045 - 2.4975) \text{ kJ/kg}\cdot\text{K} - (0.287 \text{ kJ/kg}\cdot\text{K}) \ln \frac{4.345 \text{ MPa}}{1.7997 \text{ MPa}} \\ &= 0.7540 \text{ kJ/kg}\cdot\text{K} \end{aligned}$$

Also,

$$q_{\text{in}} = 800 \text{ kJ/kg} \quad \text{and} \quad T_{\text{source}} = 1700 \text{ K}$$

Thus,

$$\begin{aligned} x_{\text{dest},23} &= T_0\left[(s_3 - s_2)_{\text{sys}} - \frac{q_{\text{in}}}{T_{\text{source}}}\right] \\ &= (290 \text{ K})\left[0.7540 \text{ kJ/kg}\cdot\text{K} - \frac{800 \text{ kJ/kg}}{1700 \text{ K}}\right] \\ &= 82.2 \text{ kJ/kg} \end{aligned}$$

For process 4-1, $s_1 - s_4 = s_2 - s_3 = -0.7540$ kJ/kg · K, $q_{R,41} = q_{\text{out}} =$ 381.83 kJ/kg, and $T_{\text{sink}} = 290$ K. Thus,

$$x_{\text{dest},41} = T_0\left[(s_1 - s_4)_{\text{sys}} + \frac{q_{\text{out}}}{T_{\text{sink}}}\right]$$

$$= (290\text{ K})\left[-0.7540\text{ kJ/kg}\cdot\text{K} + \frac{381.83\text{ kJ/kg}}{290\text{ K}}\right]$$
$$= 163.2\text{ kJ/kg}$$

Therefore, the irreversibility of the cycle is

$$x_{\text{dest,cycle}} = x_{\text{dest,12}} + x_{\text{dest,23}} + x_{\text{dest,34}} + x_{\text{dest,41}}$$
$$= 0 + 82.2\text{ kJ/kg} + 0 + 163.2\text{ kJ/kg}$$
$$= \mathbf{245.4\text{ kJ/kg}}$$

The exergy destruction of the cycle could also be determined from Eq. 9–34. Notice that the largest exergy destruction in the cycle occurs during the heat-rejection process. Therefore, any attempt to reduce the exergy destruction should start with this process.

Disregarding any kinetic and potential energies, the exergy (work potential) of the working fluid before it is purged (state 4) is determined from Eq. 9–35:

$$\phi_4 = (u_4 - u_0) - T_0(s_4 - s_0) + P_0(v_4 - v_0)$$

where

$$s_4 - s_0 = s_4 - s_1 = 0.7540\text{ kJ/kg}\cdot\text{K}$$
$$u_4 - u_0 = u_4 - u_1 = q_{\text{out}} = 381.83\text{ kJ/kg}$$
$$v_4 - v_0 = v_4 - v_1 = 0$$

Thus,

$$\phi_4 = 381.83\text{ kJ/kg} - (290\text{ K})(0.7540\text{ kJ/kg}\cdot\text{K}) + 0 = \mathbf{163.2\text{ kJ/kg}}$$

which is equivalent to the exergy destruction for process 4-1. (Why?)

Discussion Note that 163.2 kJ/kg of work could be obtained from the exhaust gases if they were brought to the state of the surroundings in a reversible manner.

TOPIC OF SPECIAL INTEREST* Saving Fuel and Money by Driving Sensibly

Two-thirds of the oil used in the United States is used for transportation. Half of this oil is consumed by passenger cars and light trucks that are used to commute to and from work (38 percent), run a family business (35 percent), and for recreational, social, and religious activities (27 percent). The overall fuel efficiency of the vehicles has increased considerably over the years due to improvements primarily in aerodynamics, materials, and electronic controls. However, the average fuel consumption of new vehicles has not changed much from about 20 miles per gallon (mpg) because of the increasing consumer trend toward purchasing larger and less fuel-efficient cars, trucks, and sport utility vehicles. Motorists also continue to drive more each year: 11,725 miles in 1999 compared to 10,277 miles in 1990. Consequently, the annual gasoline

*This section can be skipped without a loss in continuity. Information in this section is based largely on the publications of the U.S. Department of Energy, Environmental Protection Agency, and the American Automotive Association.

use per vehicle in the United States has increased to 603 gallons in 1999 (worth $1206 at $2.00/gal) from 506 gallons in 1990 (Fig. 9–56).

Saving fuel is not limited to good driving habits. It also involves purchasing the right car, using it responsibly, and maintaining it properly. A car does not burn any fuel when it is not running, and thus a sure way to save fuel is not to drive the car at all—but this is not the reason we buy a car. We can reduce driving and thus fuel consumption by considering viable alternatives such as *living close to work and shopping areas, working at home, working longer hours in fewer days, joining a car pool or starting one, using public transportation, combining errands into a single trip and planning ahead, avoiding rush hours and roads with heavy traffic and many traffic lights,* and simply *walking* or *bicycling* instead of driving to nearby places, with the added benefit of good health and physical fitness. Driving only when necessary is the best way to save fuel, money, and the environment too.

Driving efficiently starts before buying a car, just like raising good children starts before getting married. The buying decision made now will affect the fuel consumption for many years. Under average driving conditions, the owner of a 30-mpg vehicle will spend $400 less each year on fuel than the owner of a 20-mpg vehicle (assuming a fuel cost of $2.00 per gallon and 12,000 miles of driving per year). If the vehicle is owned for 5 years, the 30-mpg vehicle will save $2000 during this period (Fig. 9–57). The fuel consumption of a car depends on many factors such as *the type of the vehicle, the weight, the transmission type, the size and efficiency of the engine,* and *the accessories* and *the options installed.* The most fuel-efficient cars are aerodynamically designed compact cars with a small engine, manual transmission, low frontal area (the height times the width of the car), and bare essentials.

At highway speeds, most fuel is used to overcome aerodynamic drag or air resistance to motion, which is the force needed to move the vehicle through the air. This resistance force is proportional to the drag coefficient and the frontal area. Therefore, for a given frontal area, a sleek-looking aerodynamically designed vehicle with contoured lines that coincide with the streamlines of air flow has a smaller drag coefficient and thus better fuel economy than a boxlike vehicle with sharp corners (Fig. 9–58). For the same overall shape, a compact car has a smaller frontal area and thus better fuel economy compared to a large car.

Moving around the *extra weight* requires more fuel, and thus it hurts fuel economy. Therefore, the lighter the vehicle, the more fuel-efficient it is. Also as a general rule, the larger the engine is, the greater its rate of fuel consumption is. So you can expect a car with a 1.8 L engine to be more fuel efficient than one with a 3.0 L engine. For a given engine size, *diesel engines* operate on much higher compression ratios than the gasoline engines, and thus they are inherently more fuel-efficient. *Manual transmissions* are usually more efficient than the automatic ones, but this is not always the case. A car with automatic transmission generally uses 10 percent more fuel than a car with manual transmission because of the losses associated with the hydraulic connection between the engine and the transmission, and the added weight. Transmissions with an *overdrive gear* (found in four-speed automatic transmissions and five-speed manual transmissions) save fuel and reduce

FIGURE 9–56

The average car in the United States is driven about 12,000 miles a year, uses about 600 gallons of gasoline, worth $1200 at $2.00/gal.

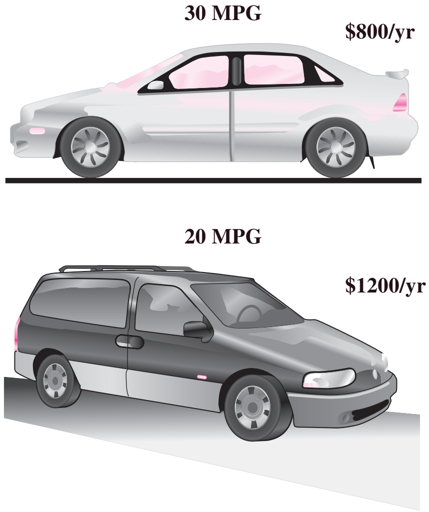

FIGURE 9–57

Under average driving conditions, the owner of a 30-mpg vehicle spends $400 less each year on gasoline than the owner of a 20-mpg vehicle (assuming $2.00/gal and 12,000 miles/yr).

noise and engine wear during highway driving by decreasing the engine rpm while maintaining the same vehicle speed.

Front wheel drive offers better traction (because of the engine weight on top of the front wheels), reduced vehicle weight and thus better fuel economy, with an added benefit of increased space in the passenger compartment. Four-wheel drive mechanisms provide better traction and braking thus safer driving on slippery roads and loose gravel by transmitting torque to all four wheels. However, the added safety comes with increased weight, noise, and cost, and decreased fuel economy. *Radial tires* usually reduce the fuel consumption by 5 to 10 percent by reducing the rolling resistance, but their pressure should be checked regularly since they can look normal and still be underinflated. *Cruise control* saves fuel during long trips on open roads by maintaining steady speed. *Tinted windows* and light interior and exterior colors reduce solar heat gain, and thus the need for air-conditioning.

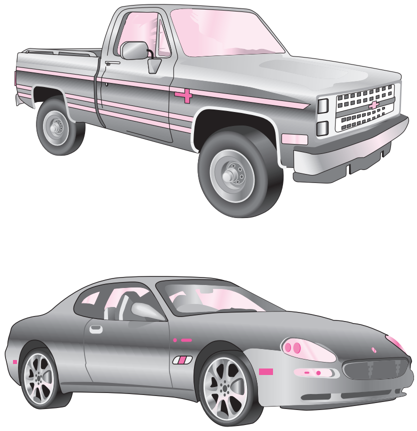

FIGURE 9–58

Aerodynamically designed vehicles have a smaller drag coefficient and thus better fuel economy than boxlike vehicles with sharp corners.

BEFORE DRIVING

Certain things done before driving can make a significant difference on the fuel cost of the vehicle while driving. Below we discuss some measures such as using the right kind of fuel, minimizing idling, removing extra weight, and keeping the tires properly inflated.

Use Fuel with the Minimum Octane Number Recommended by the Vehicle Manufacturer

Many motorists buy higher-priced premium fuel, thinking that it is better for the engine. Most of today's cars are designed to operate on regular unleaded fuel. If the owner's manual does not call for premium fuel, using anything other than regular gas is simply a waste of money. Octane number is not a measure of the "power" or "quality" of the fuel, it is simply a measure of fuel's resistance to engine knock caused by premature ignition. Despite the implications of flashy names like "premium," "super," or "power plus," a fuel with a higher octane number is not a better fuel; it is simply more expensive because of the extra processing involved to raise the octane number (Fig. 9–59). Older cars may need to go up one grade level from the recommended new car octane number if they start knocking.

FIGURE 9–59

Despite the implications of flashy names, a fuel with a higher octane number is not a better fuel; it is simply more expensive.

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Do Not Overfill the Gas Tank

Topping off the gas tank may cause the fuel to backflow during pumping. In hot weather, an overfilled tank may also cause the fuel to overflow due to thermal expansion. This wastes fuel, pollutes the environment, and may damage the car's paint. Also, fuel tank caps that do not close tightly allow some gasoline to be lost by evaporation. Buying fuel in cool weather such as early in the mornings minimizes evaporative losses. Each gallon of spilled or evaporated fuel emits as much hydrocarbon to the air as 7500 miles of driving.

Park in the Garage

The engine of a car parked in a garage overnight is warmer the next morning. This reduces the problems associated with the warming-up period such as starting, excessive fuel consumption, and environmental pollution. In hot weather, a garage blocks the direct sunlight and reduces the need for air-conditioning.

Start the Car Properly and Avoid Extended Idling

With today's cars, it is not necessary to prime the engine first by pumping the accelerator pedal repeatedly before starting. This only wastes fuel. Warming up the engine isn't necessary either. Keep in mind that an idling engine wastes fuel and pollutes the environment. Don't race a cold engine to warm it up. An engine warms up faster on the road under a light load, and the catalytic converter begins to function sooner. Start driving as soon as the engine is started, but avoid rapid acceleration and highway driving before the engine and thus the oil fully warms up to prevent engine wear.

In cold weather, the warm-up period is much longer, the fuel consumption during warm-up is much higher, and the exhaust emissions are much larger. At −20°C, for example, a car needs to be driven at least 3 miles to warm up fully. A gasoline engine uses up to 50 percent more fuel during warm-up than it does after it is warmed up. Exhaust emissions from a cold engine during warm-up are much higher since the catalytic converters do not function properly before reaching their normal operating temperature of about 390°C.

Don't Carry Unnecessary Weight in or on the Vehicle

Remove any snow or ice from the vehicle, and avoid carrying unneeded items, especially heavy ones (such as snow chains, old tires, books) in the passenger compartment, trunk, or the cargo area of the vehicle (Fig. 9–60). This wastes fuel since it requires extra fuel to carry around the extra weight. An extra 100 lbm decreases fuel economy of a car by about 1–2 percent.

Some people find it convenient to use a roof rack or carrier for additional cargo space. However, if you must carry some extra items, place them inside the vehicle rather than on roof racks to reduce drag. Any snow that accumulates on a vehicle and distorts its shape must be removed for the same reason. A loaded roof rack can increase fuel consumption by up to 5 percent in highway driving. Even the most streamlined empty rack increases aerodynamic drag and thus fuel consumption. Therefore, the roof rack should be removed when it is no longer needed.

FIGURE 9–60
A loaded roof rack can increase fuel consumption by up to 5 percent in highway driving.

Keep Tires Inflated to the Recommended Maximum Pressure

Keeping the tires inflated properly is one of the easiest and most important things one can do to improve fuel economy. If a range is recommended by the manufacturer, the higher pressure should be used to maximize fuel efficiency. Tire pressure should be checked when the tire is cold since tire pressure changes with temperature (it increases by 1 psi for every 10°F rise in temperature due to a rise in ambient temperature or just road friction). Underinflated tires run hot and jeopardize safety, cause the tires to wear prematurely, affect

FIGURE 9–61

Underinflated tires often cause fuel consumption of vehicles to increase by 5 or 6 percent.

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the vehicle's handling adversely, and hurt the fuel economy by increasing the rolling resistance. Overinflated tires cause unpleasant bumpy rides, and cause the tires to wear unevenly. Tires lose about 1 psi pressure per month due to air loss caused by the tire hitting holes, bumps, and curbs. Therefore, the tire pressure should be checked at least once a month. Just one tire underinflated by 2 psi results in a 1 percent increase in fuel consumption (Fig. 9–61). Underinflated tires often cause fuel consumption of vehicles to increase by 5 or 6 percent.

It is also important to keep the wheels aligned. Driving a vehicle with the front wheels out of alignment increases rolling resistance and thus fuel consumption while causing handling problems and uneven tire wear. Therefore, the wheels should be aligned properly whenever necessary.

WHILE DRIVING

The driving habits can make a significant difference in the amount of fuel used. Driving sensibly and practicing some fuel-efficient driving techniques such as those discussed below can improve fuel economy easily by more than 10 percent.

Avoid Quick Starts and Sudden Stops

Despite the attention they may get, the abrupt, aggressive "jackrabbit" starts waste fuel, wear the tires, jeopardize safety, and are harder on vehicle components and connectors. The squealing stops wear the brake pads prematurely, and may cause the driver to lose control of the vehicle. Easy starts and stops save fuel, reduce wear and tear, reduce pollution, and are safer and more courteous to other drivers.

Drive at Moderate Speeds

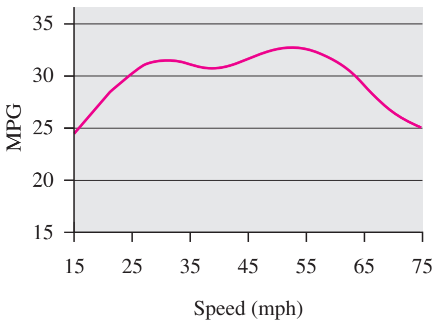

FIGURE 9–62

Aerodynamic drag increases and thus fuel economy decreases rapidly at speeds above 55 mph.

Source: EPA and U.S. Dept. of Energy.

Avoiding high speeds on open roads results in safer driving and better fuel economy. In highway driving, over 50 percent of the power produced by the engine is used to overcome aerodynamic drag (i.e., to push air out of the way). Aerodynamic drag and thus fuel consumption increase rapidly at speeds above 55 mph, as shown in Fig. 9–62. On average, a car uses about 15 percent more fuel at 65 mph and 25 percent more fuel at 70 mph than it does at 55 mph. (A car uses about 10 percent more fuel at 100 km/h and 20 percent more fuel at 110 km/h than it does at 90 km/h.)

The discussion above should not lead one to conclude that the lower the speed, the better the fuel economy—because it is not. The number of miles that can be driven per gallon of fuel drops sharply at speeds below 30 mph (or 50 km/h), as shown in the chart. Besides, speeds slower than the flow of traffic can create a traffic hazard. Therefore, a car should be driven at moderate speeds for safety and best fuel economy.

Maintain a Constant Speed

The fuel consumption remains at a minimum during steady driving at a moderate speed. Keep in mind that every time the accelerator is hard pressed, more fuel is pumped into the engine. The vehicle should be accelerated gradually and smoothly since extra fuel is squirted into the engine during quick acceleration. Using cruise control on highway trips can help maintain a constant speed and reduce fuel consumption. Steady driving is also safer, easier on the nerves, and better for the heart.

Anticipate Traffic Ahead and Avoid Tailgating

A driver can reduce fuel consumption by up to 10 percent by anticipating traffic conditions ahead and adjusting the speed accordingly, and avoiding tailgating and thus unnecessary braking and acceleration (Fig. 9–63). Accelerations and decelerations waste fuel. Braking and abrupt stops can be minimized, for example, by not following too closely, and slowing down gradually by releasing the gas pedal when approaching a red light, a stop sign, or slow traffic. This relaxed driving style is safer, saves fuel and money, reduces pollution, reduces wear on the tires and brakes, and is appreciated by other drivers. Allowing sufficient time to reach the destination makes it easier to resist the urge to tailgate.

FIGURE 9–63

Fuel consumption can be decreased by up to 10 percent by anticipating traffic conditions ahead and adjusting accordingly.

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Avoid Sudden Acceleration and Sudden Braking (Except in Emergencies)

Accelerate gradually and smoothly when passing other vehicles or merging with faster traffic. Pumping or hard pressing the accelerator pedal while driving causes the engine to switch to a “fuel enrichment mode” of operation that wastes fuel. In city driving, nearly half of the engine power is used for acceleration. When accelerating with stick-shifts, the RPM of the engine should be kept to a minimum. Braking wastes the mechanical energy produced by the engine and wears the brake pads.

Avoid Resting Feet on the Clutch or Brake Pedal while Driving

Resting the left foot on the brake pedal increases the temperature of the brake components, and thus reduces their effectiveness and service life while wasting fuel. Similarly, resting the left foot on the clutch pedal lessens the pressure on the clutch pads, causing them to slip and wear prematurely, wasting fuel.

Use Highest Gear (Overdrive) During Highway Driving

Overdrive improves fuel economy during highway driving by decreasing the vehicle’s engine speed (or RPM). The lower engine speed reduces fuel consumption per unit time as well as engine wear. Therefore, overdrive (the fifth gear in cars with overdrive manual transmission) should be used as soon as the vehicle’s speed is high enough.

Turn the Engine Off Rather Than Letting It Idle

Unnecessary idling during lengthy waits (such as waiting for someone or for service at a drive-up window, being stuck in traffic, etc.) wastes fuel, pollutes the air, and causes engine wear (more wear than driving) (Fig. 9–64). Therefore, the engine should be turned off rather than letting it idle. Idling for more than a minute consumes much more fuel than restarting the engine. Fuel consumption in the lines of drive-up windows and the pollution emitted can be avoided altogether by simply parking the car and going inside.

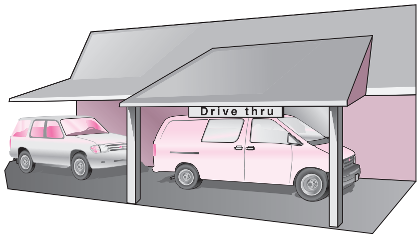

FIGURE 9–64

Unnecessary idling during lengthy waits wastes fuel, costs money, and pollutes the air.

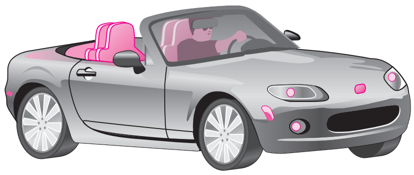

FIGURE 9–65

Air conditioning increases fuel consumption by 3 to 4 percent during highway driving, and by as much as 10 percent during city driving.

Use the Air Conditioner Sparingly

Air-conditioning consumes considerable power and thus increases fuel consumption by 3 to 4 percent during highway driving, and by as much as 10 percent during city driving (Fig. 9–65). The best alternative to air-conditioning is to supply fresh outdoor air to the car through the vents by turning on the flow-through ventilation system (usually by running the air conditioner in the "economy" mode) while keeping the windows and the sunroof closed. This measure is adequate to achieve comfort in pleasant weather, and it saves the most fuel since the compressor of the air conditioner is off. In warmer weather, however, ventilation cannot provide adequate cooling effect. In that case we can attempt to achieve comfort by rolling down the windows or opening the sunroof. This is certainly a viable alternative for city driving, but not so on highways since the aerodynamic drag caused by wide-open windows or sunroof at highway speeds consumes more fuel than does the air conditioner. Therefore, at highway speeds, the windows or the sunroof should be closed and the air conditioner should be turned on instead to save fuel. This is especially the case for the newer, aerodynamically designed cars.

Most air conditioners have a "maximum" or "recirculation" setting that reduces the amount of hot outside air that must be cooled, and thus the fuel consumption for air-conditioning. A passive measure to reduce the need for air conditioning is to park the vehicle in the shade, and to leave the windows slightly open to allow for air circulation.

AFTER DRIVING

You cannot be an efficient person and accomplish much unless you take good care of yourself (eating right, maintaining physical fitness, having checkups, etc.), and the cars are no exception. Regular maintenance improves performance, increases gas mileage, reduces pollution, lowers repair costs, and extends engine life. A little time and money saved now may cost a lot later in increased fuel, repair, and replacement costs.

Proper maintenance such as *checking the levels of fluids (engine oil, coolant, transmission, brake, power steering, windshield washer, etc.), the tightness of all belts, and formation of cracks or frays on hoses, belts, and wires, keeping tires properly inflated, lubricating the moving components, and replacing clogged air, fuel, or oil filters* maximizes fuel efficiency (Fig. 9–66). Clogged air filters increase fuel consumption (by up to 10 percent) and pollution by restricting airflow to the engine, and thus they should be replaced. The car should be tuned up regularly unless it has electronic controls and a fuel-injection system. High temperatures (which may be due to a malfunction of the cooling fan) should be avoided as they may cause the break down of the engine oil and thus excessive wear of the engine, and low temperatures (which may be due to a malfunction of the thermostat) may extend the engine's warm-up period and may prevent the engine from reaching the optimum operating conditions. Both effects reduce fuel economy.

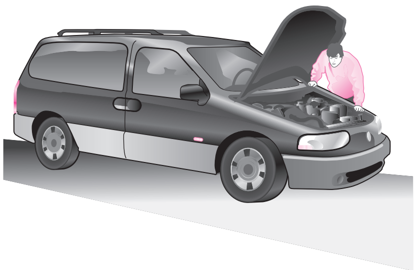

FIGURE 9–66

Proper maintenance maximizes fuel efficiency and extends engine life.

Clean oil extends engine life by reducing engine wear caused by friction, removes acids, sludge, and other harmful substances from the engine, improves performance, reduces fuel consumption, and decreases air pollution. Oil also helps to cool the engine, provides a seal between the cylinder walls and the pistons, and prevents the engine from rusting. Therefore, oil and oil filter should be changed as recommended by the vehicle manufacturer. Fuel-efficient oils (indicated by "Energy Efficient API" label) contain certain additives that reduce friction and increase a vehicle's fuel economy by 3 percent or more.

In summary, a person can save fuel, money, and the environment by *purchasing an energy-efficient vehicle, minimizing the amount of driving, being fuel-conscious while driving, and maintaining the car properly.* These measures have the added benefits of enhanced safety, reduced maintenance costs, and extended vehicle life.

SUMMARY

A cycle during which a net amount of work is produced is called a *power cycle,* and a power cycle during which the working fluid remains a gas throughout is called a *gas power cycle.* The most efficient cycle operating between a heat source at temperature T_H and a sink at temperature T_L is the Carnot cycle, and its thermal efficiency is given by

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

The actual gas cycles are rather complex. The approximations used to simplify the analysis are known as the *air-standard assumptions.* Under these assumptions, all the processes are assumed to be internally reversible; the working fluid is assumed to be air, which behaves as an ideal gas; and the combustion and exhaust processes are replaced by heat-addition and heat-rejection processes, respectively. The air-standard assumptions are called *cold-air-standard assumptions* if air is also assumed to have constant specific heats at room temperature.

In reciprocating engines, the *compression ratio* r and the *mean effective pressure* MEP are defined as

$$r = \frac{V_{\max}}{V_{\min}} = \frac{V_{\text{BDC}}}{V_{\text{TDC}}}$$

$$\text{MEP} = \frac{w_{\text{net}}}{v_{\max} - v_{\min}}$$

The *Otto cycle* is the ideal cycle for the spark-ignition reciprocating engines, and it consists of four internally reversible processes: isentropic compression, constant-volume heat addition, isentropic expansion, and constant-volume heat rejection. Under cold-air-standard assumptions, the thermal efficiency of the ideal Otto cycle is

$$\eta_{\text{th,Otto}} = 1 - \frac{1}{r^{k-1}}$$

where r is the compression ratio and k is the specific heat ratio c_p/c_v.

The *Diesel cycle* is the ideal cycle for the compression-ignition reciprocating engines. It is very similar to the Otto cycle, except that the constant-volume heat-addition process is replaced by a constant-pressure heat-addition process. Its thermal efficiency under cold-air-standard assumptions is

$$\eta_{\text{th,Diesel}} = 1 - \frac{1}{r^{k-1}}\left[\frac{r_c^k - 1}{k(r_c - 1)}\right]$$

where r_c is the *cutoff ratio,* defined as the ratio of the cylinder volumes after and before the combustion process.

Stirling and *Ericsson cycles* are two totally reversible cycles that involve an isothermal heat-addition process at T_H and an isothermal heat-rejection process at T_L. They differ from the Carnot cycle in that the two isentropic processes are replaced by two constant-volume regeneration processes in the Stirling cycle and by two constant-pressure regeneration processes in the Ericsson cycle. Both cycles utilize *regeneration,* a process during which heat is transferred to a thermal energy storage device (called a *regenerator*) during one part of the cycle that is then transferred back to the working fluid during another part of the cycle.

The ideal cycle for modern gas-turbine engines is the *Brayton cycle,* which is made up of four internally reversible processes: isentropic compression, constant-pressure heat addition, isentropic expansion, and constant-pressure heat rejection. Under cold-air-standard assumptions, its thermal efficiency is

$$\eta_{\text{th,Brayton}} = 1 - \frac{1}{r_p^{(k-1)/k}}$$

where $r_p = P_{\max}/P_{\min}$ is the pressure ratio and k is the specific heat ratio. The thermal efficiency of the simple Brayton cycle increases with the pressure ratio.

The deviation of the actual compressor and the turbine from the idealized isentropic ones can be accurately accounted for by utilizing their isentropic efficiencies, defined as

$$\eta_C = \frac{w_s}{w_a} \cong \frac{h_{2s} - h_1}{h_{2a} - h_1}$$

and

$$\eta_T = \frac{w_a}{w_s} \cong \frac{h_3 - h_{4a}}{h_3 - h_{4s}}$$

where states 1 and 3 are the inlet states, 2a and 4a are the actual exit states, and 2s and 4s are the isentropic exit states.

In gas-turbine engines, the temperature of the exhaust gas leaving the turbine is often considerably higher than the temperature of the air leaving the compressor. Therefore, the high-pressure air leaving the compressor can be heated by transferring heat to it from the hot exhaust gases in a counter-flow heat exchanger, which is also known as a *regenerator.* The extent to which a regenerator approaches an ideal regenerator is called the *effectiveness* ϵ and is defined as

$$\epsilon = \frac{q_{\text{regen,act}}}{q_{\text{regen,max}}}$$

Under cold-air-standard assumptions, the thermal efficiency of an ideal Brayton cycle with regeneration becomes

$$\eta_{\text{th,regen}} = 1 - \left(\frac{T_1}{T_3}\right)(r_p)^{(k-1)/k}$$

where T_1 and T_3 are the minimum and maximum temperatures, respectively, in the cycle.

The thermal efficiency of the Brayton cycle can also be increased by utilizing *multistage compression with intercooling, regeneration, and multistage expansion with reheating.* The work input to the compressor is minimized when equal pressure ratios are maintained across each stage. This procedure also maximizes the turbine work output.

Gas-turbine engines are widely used to power aircraft because they are light and compact and have a high power-to-weight ratio. The ideal *jet-propulsion cycle* differs from the simple ideal Brayton cycle in that the gases are partially expanded in the turbine. The gases that exit the turbine at a relatively high pressure are subsequently accelerated in a nozzle to provide the thrust needed to propel the aircraft.

The *net thrust* developed by the engine is

$$F = \dot{m}(V_{\text{exit}} - V_{\text{inlet}})$$

where $\dot{m}$ is the mass flow rate of gases, V_{exit} is the exit velocity of the exhaust gases, and V_{inlet} is the inlet velocity of the air, both relative to the aircraft.

The power developed from the thrust of the engine is called the *propulsive power* $\dot{W}_P$, and it is given by

$$\dot{W}_P = \dot{m}(V_{\text{exit}} - V_{\text{inlet}})V_{\text{aircraft}}$$

Propulsive efficiency is a measure of how efficiently the energy released during the combustion process is converted to propulsive energy, and it is defined as

$$\eta_P = \frac{\text{Propulsive power}}{\text{Energy input rate}} = \frac{\dot{W}_P}{\dot{Q}_{\text{in}}}$$

For an ideal cycle that involves heat transfer only with a source at T_H and a sink at T_L, the exergy destruction is

$$x_{\text{dest}} = T_0\left(\frac{q_{\text{out}}}{T_L} - \frac{q_{\text{in}}}{T_H}\right)$$

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PROBLEMS*

Actual and Ideal Cycles, Carnot Cycle, Air-Standard Assumptions, Reciprocating Engines

9–1C Why is the Carnot cycle not suitable as an ideal cycle for all power-producing cyclic devices?

9–2C How does the thermal efficiency of an ideal cycle, in general, compare to that of a Carnot cycle operating between the same temperature limits?

9–3C What does the area enclosed by the cycle represent on a *P*-*v* diagram? How about on a *T*-*s* diagram?

9–4C What is the difference between air-standard assumptions and the cold-air-standard assumptions?

9–5C How are the combustion and exhaust processes modeled under the air-standard assumptions?

9–6C What are the air-standard assumptions?

9–7C What is the difference between the clearance volume and the displacement volume of reciprocating engines?

9–8C Define the compression ratio for reciprocating engines.

9–9C How is the mean effective pressure for reciprocating engines defined?

9–10C Can the mean effective pressure of an automobile engine in operation be less than the atmospheric pressure?

9–11C As a car gets older, will its compression ratio change? How about the mean effective pressure?

9–12C What is the difference between spark-ignition and compression-ignition engines?

9–13C Define the following terms related to reciprocating engines: stroke, bore, top dead center, and clearance volume.

9–14 An air-standard cycle with variable specific heats is executed in a closed system and is composed of the following four processes:

1-2 Isentropic compression from 100 kPa and 27°C to 800 kPa
2-3 v = *constant* heat addition to 1800 K
3-4 Isentropic expansion to 100 kPa
4-1 P = *constant* heat rejection to initial state

(*a*) Show the cycle on *P*-*v* and *T*-*s* diagrams.
(*b*) Calculate the net work output per unit mass.
(*c*) Determine the thermal efficiency.

* 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.

9–15 Reconsider Problem 9–14. Using EES (or other) software, study the effect of varying the temperature after the constant-volume heat addition from 1500 K to 2500 K. Plot the net work output and thermal efficiency as a function of the maximum temperature of the cycle. Plot the *T*-*s* and *P*-*v* diagrams for the cycle when the maximum temperature of the cycle is 1800 K.

9–16 An air-standard cycle is executed in a closed system and is composed of the following four processes:

1-2 Isentropic compression from 100 kPa and 27°C to 1 MPa
2-3 P = *constant* heat addition in amount of 2800 kJ/kg
3-4 v = *constant* heat rejection to 100 kPa
4-1 P = *constant* heat rejection to initial state

(*a*) Show the cycle on *P*-*v* and *T*-*s* diagrams.
(*b*) Calculate the maximum temperature in the cycle.
(*c*) Determine the thermal efficiency.

Assume constant specific heats at room temperature.
Answers: (*b*) 3360 K, (*c*) 21.0 percent

9–17E An air-standard cycle with variable specific heats is executed in a closed system and is composed of the following four processes:

1-2 v = *constant* heat addition from 14.7 psia and 80°F in the amount of 300 Btu/lbm
2-3 P = *constant* heat addition to 3200 R
3-4 Isentropic expansion to 14.7 psia
4-1 P = *constant* heat rejection to initial state

(*a*) Show the cycle on *P*-*v* and *T*-*s* diagrams.
(*b*) Calculate the total heat input per unit mass.
(*c*) Determine the thermal efficiency.
Answers: (*b*) 612.4 Btu/lbm, (*c*) 24.2 percent

9–18E Repeat Problem 9–17E using constant specific heats at room temperature.

9–19 An air-standard cycle is executed in a closed system with 0.004 kg of air and consists of the following three processes:

1-2 Isentropic compression from 100 kPa and 27°C to 1 MPa
2-3 P = *constant* heat addition in the amount of 2.76 kJ
3-1 $P = c_1 v + c_2$ heat rejection to initial state (c_1 and c_2 are constants)

(*a*) Show the cycle on *P*-*v* and *T*-*s* diagrams.
(*b*) Calculate the heat rejected.
(*c*) Determine the thermal efficiency.

Assume constant specific heats at room temperature.
Answers: (*b*) 1.679 kJ, (*c*) 39.2 percent

9–20 An air-standard cycle with variable specific heats is executed in a closed system with 0.003 kg of air and consists of the following three processes:

1-2 v = *constant* heat addition from 95 kPa and 17°C to 380 kPa
2-3 Isentropic expansion to 95 kPa
3-1 P = *constant* heat rejection to initial state

(*a*) Show the cycle on P-v and T-s diagrams.
(*b*) Calculate the net work per cycle, in kJ.
(*c*) Determine the thermal efficiency.

9–21 Repeat Problem 9–20 using constant specific heats at room temperature.

9–22 Consider a Carnot cycle executed in a closed system with 0.003 kg of air. The temperature limits of the cycle are 300 and 900 K, and the minimum and maximum pressures that occur during the cycle are 20 and 2000 kPa. Assuming constant specific heats, determine the net work output per cycle.

9–23 An air-standard Carnot cycle is executed in a closed system between the temperature limits of 350 and 1200 K. The pressures before and after the isothermal compression are 150 and 300 kPa, respectively. If the net work output per cycle is 0.5 kJ, determine (*a*) the maximum pressure in the cycle, (*b*) the heat transfer to air, and (*c*) the mass of air. Assume variable specific heats for air. *Answers:* (*a*) 30,013 kPa, (*b*) 0.706 kJ, (*c*) 0.00296 kg

9–24 Repeat Problem 9–23 using helium as the working fluid.

9–25 Consider a Carnot cycle executed in a closed system with air as the working fluid. The maximum pressure in the cycle is 800 kPa while the maximum temperature is 750 K. If the entropy increase during the isothermal heat rejection process is 0.25 kJ/kg · K and the net work output is 100 kJ/kg, determine (a) the minimum pressure in the cycle, (b) the heat rejection from the cycle, and (c) the thermal efficiency of the cycle. (d) If an actual heat engine cycle operates between the same temperature limits and produces 5200 kW of power for an air flow rate of 90 kg/s, determine the second law efficiency of this cycle.

Otto Cycle

9–26C What four processes make up the ideal Otto cycle?

9–27C How do the efficiencies of the ideal Otto cycle and the Carnot cycle compare for the same temperature limits? Explain.

9–28C How is the rpm (revolutions per minute) of an actual four-stroke gasoline engine related to the number of thermodynamic cycles? What would your answer be for a two-stroke engine?

9–29C Are the processes that make up the Otto cycle analyzed as closed-system or steady-flow processes? Why?

9–30C How does the thermal efficiency of an ideal Otto cycle change with the compression ratio of the engine and the specific heat ratio of the working fluid?

9–31C Why are high compression ratios not used in spark-ignition engines?

9–32C An ideal Otto cycle with a specified compression ratio is executed using (*a*) air, (*b*) argon, and (*c*) ethane as the working fluid. For which case will the thermal efficiency be the highest? Why?

9–33C What is the difference between fuel-injected gasoline engines and diesel engines?

9–34 An ideal Otto cycle has a compression ratio of 8. At the beginning of the compression process, air is at 95 kPa and 27°C, and 750 kJ/kg of heat is transferred to air during the constant-volume heat-addition process. Taking into account the variation of specific heats with temperature, determine (*a*) the pressure and temperature at the end of the heat-addition process, (*b*) the net work output, (*c*) the thermal efficiency, and (*d*) the mean effective pressure for the cycle. *Answers:* (*a*) 3898 kPa, 1539 K, (*b*) 392.4 kJ/kg, (*c*) 52.3 percent, (*d*) 495 kPa

9–35 Reconsider Problem 9–34. Using EES (or other) software, study the effect of varying the compression ratio from 5 to 10. Plot the net work output and thermal efficiency as a function of the compression ratio. Plot the T-s and P-v diagrams for the cycle when the compression ratio is 8.

9–36 Repeat Problem 9–34 using constant specific heats at room temperature.

9–37 The compression ratio of an air-standard Otto cycle is 9.5. Prior to the isentropic compression process, the air is at 100 kPa, 35°C, and 600 cm^3. The temperature at the end of the isentropic expansion process is 800 K. Using specific heat values at room temperature, determine (*a*) the highest temperature and pressure in the cycle; (*b*) the amount of heat transferred in, in kJ; (*c*) the thermal efficiency; and (*d*) the mean effective pressure. *Answers:* (*a*) 1969 K, 6072 kPa, (*b*) 0.59 kJ, (c) 59.4 percent, (*d*) 652 kPa

9–38 Repeat Problem 9–37, but replace the isentropic expansion process by a polytropic expansion process with the polytropic exponent n = 1.35.

9–39E An ideal Otto cycle with air as the working fluid has a compression ratio of 8. The minimum and maximum temperatures in the cycle are 540 and 2400 R. Accounting for the variation of specific heats with temperature, determine (*a*) the amount of heat transferred to the air during the heat-addition process, (*b*) the thermal efficiency, and (*c*) the thermal efficiency of a Carnot cycle operating between the same temperature limits.

9–40E Repeat Problem 9–39E using argon as the working fluid.

9–41 A four-cylinder, four-stroke, 2.2-L gasoline engine operates on the Otto cycle with a compression ratio of 10. The air is at 100 kPa and 60°C at the beginning of the compression process, and the maximum pressure in the cycle is 8 MPa. The compression and expansion processes may be modeled as polytropic with a polytropic constant of 1.3. Using constant specific heats at 850 K, determine (*a*) the temperature at the end of the expansion process, (*b*) the net work output and the thermal efficiency, (*c*) the mean effective pressure, (*d*) the engine speed for a net power output of 70 kW, and (*e*) the specific fuel consumption, in g/kWh, defined as the ratio of the mass of the fuel consumed to the net work produced. The air–fuel ratio, defined as the amount of air divided by the amount of fuel intake, is 16.

DIESEL CYCLE

9–42C How does a diesel engine differ from a gasoline engine?

9–43C How does the ideal Diesel cycle differ from the ideal Otto cycle?

9–44C For a specified compression ratio, is a diesel or gasoline engine more efficient?

9–45C Do diesel or gasoline engines operate at higher compression ratios? Why?

9–46C What is the cutoff ratio? How does it affect the thermal efficiency of a Diesel cycle?

9–47 An air-standard Diesel cycle has a compression ratio of 16 and a cutoff ratio of 2. At the beginning of the compression process, air is at 95 kPa and 27°C. Accounting for the variation of specific heats with temperature, determine (*a*) the temperature after the heat-addition process, (*b*) the thermal efficiency, and (*c*) the mean effective pressure. *Answers:* (*a*) 1724.8 K, (*b*) 56.3 percent, (*c*) 675.9 kPa

9–48 Repeat Problem 9–47 using constant specific heats at room temperature.

9–49E An air-standard Diesel cycle has a compression ratio of 18.2. Air is at 80°F and 14.7 psia at the beginning of the compression process and at 3000 R at the end of the heat-addition process. Accounting for the variation of specific heats with temperature, determine (*a*) the cutoff ratio, (*b*) the heat rejection per unit mass, and (*c*) the thermal efficiency.

9–50E Repeat Problem 9–49E using constant specific heats at room temperature.

9–51 An ideal diesel engine has a compression ratio of 20 and uses air as the working fluid. The state of air at the beginning of the compression process is 95 kPa and 20°C. If the maximum temperature in the cycle is not to exceed 2200 K, determine (*a*) the thermal efficiency and (*b*) the mean effective pressure. Assume constant specific heats for air at room temperature. *Answers:* (*a*) 63.5 percent, (*b*) 933 kPa

9–52 Repeat Problem 9–51, but replace the isentropic expansion process by polytropic expansion process with the polytropic exponent $n = 1.35$.

9–53 Reconsider Problem 9–52. Using EES (or other) software, study the effect of varying the compression ratio from 14 to 24. Plot the net work output, mean effective pressure, and thermal efficiency as a function of the compression ratio. Plot the T-s and P-v diagrams for the cycle when the compression ratio is 20.

9–54 A four-cylinder two-stroke 2.4-L diesel engine that operates on an ideal Diesel cycle has a compression ratio of 17 and a cutoff ratio of 2.2. Air is at 55°C and 97 kPa at the beginning of the compression process. Using the cold-air-standard assumptions, determine how much power the engine will deliver at 1500 rpm.

9–55 Repeat Problem 9–54 using nitrogen as the working fluid.

9–56 The compression ratio of an ideal dual cycle is 14. Air is at 100 kPa and 300 K at the beginning of the compression process and at 2200 K at the end of the heat-addition process. Heat transfer to air takes place partly at constant volume and partly at constant pressure, and it amounts to 1520.4 kJ/kg. Assuming variable specific heats for air, determine (*a*) the fraction of heat transferred at constant volume and (*b*) the thermal efficiency of the cycle.

9–57 Reconsider Problem 9–56. Using EES (or other) software, study the effect of varying the compression ratio from 10 to 18. For the compression ratio equal to 14, plot the T-s and P-v diagrams for the cycle.

9–58 Repeat Problem 9–56 using constant specific heats at room temperature. Is the constant specific heat assumption reasonable in this case?

9–59 A six-cylinder, four-stroke, 4.5-L compression-ignition engine operates on the ideal diesel cycle with a compression ratio of 17. The air is at 95 kPa and 55°C at the beginning of the compression process and the engine speed is 2000 rpm. The engine uses light diesel fuel with a heating value of 42,500 kJ/kg, an air–fuel ratio of 24, and a combustion efficiency of 98 percent. Using constant specific heats at 850 K, determine (*a*) the maximum temperature in the cycle and the cutoff ratio (*b*) the net work output per cycle and the thermal efficiency, (*c*) the mean effective pressure, (*d*) the net power output, and (*e*) the specific fuel consumption, in g/kWh, defined as the ratio of the mass of the fuel consumed to the net work produced. *Answers:* (*a*) 2383 K, 2.7 (*b*) 4.36 kJ, 0.543, (*c*) 969 kPa, (*d*) 72.7 kW, (*e*) 159 g/kWh

Stirling and Ericsson Cycles

9–60C Consider the ideal Otto, Stirling, and Carnot cycles operating between the same temperature limits. How would you compare the thermal efficiencies of these three cycles?

9–61C Consider the ideal Diesel, Ericsson, and Carnot cycles operating between the same temperature limits. How would you compare the thermal efficiencies of these three cycles?

9–62C What cycle is composed of two isothermal and two constant-volume processes?

9–63C How does the ideal Ericsson cycle differ from the Carnot cycle?

9–64E An ideal Ericsson engine using helium as the working fluid operates between temperature limits of 550 and 3000 R and pressure limits of 25 and 200 psia. Assuming a mass flow rate of 14 lbm/s, determine (*a*) the thermal efficiency of the cycle, (*b*) the heat transfer rate in the regenerator, and (*c*) the power delivered.

9–65 Consider an ideal Ericsson cycle with air as the working fluid executed in a steady-flow system. Air is at 27°C and 120 kPa at the beginning of the isothermal compression process, during which 150 kJ/kg of heat is rejected. Heat transfer to air occurs at 1200 K. Determine (*a*) the maximum pressure in the cycle, (*b*) the net work output per unit mass of air, and (*c*) the thermal efficiency of the cycle. *Answers:* (*a*) 685 kPa, (*b*) 450 kJ/kg, (*c*) 75 percent

9–66 An ideal Stirling engine using helium as the working fluid operates between temperature limits of 300 and 2000 K and pressure limits of 150 kPa and 3 MPa. Assuming the mass of the helium used in the cycle is 0.12 kg, determine (*a*) the thermal efficiency of the cycle, (*b*) the amount of heat transfer in the regenerator, and (*c*) the work output per cycle.

Ideal and Actual Gas-Turbine (Brayton) Cycles

9–67C Why are the back work ratios relatively high in gas-turbine engines?

9–68C What four processes make up the simple ideal Brayton cycle?

9–69C For fixed maximum and minimum temperatures, what is the effect of the pressure ratio on (*a*) the thermal efficiency and (*b*) the net work output of a simple ideal Brayton cycle?

9–70C What is the back work ratio? What are typical back work ratio values for gas-turbine engines?

9–71C How do the inefficiencies of the turbine and the compressor affect (*a*) the back work ratio and (*b*) the thermal efficiency of a gas-turbine engine?

9–72E A simple ideal Brayton cycle with air as the working fluid has a pressure ratio of 10. The air enters the compressor at 520 R and the turbine at 2000 R. Accounting for the variation of specific heats with temperature, determine (*a*) the air temperature at the compressor exit, (*b*) the back work ratio, and (*c*) the thermal efficiency.

9–73 A simple Brayton cycle using air as the working fluid has a pressure ratio of 8. The minimum and maximum temperatures in the cycle are 310 and 1160 K. Assuming an isentropic efficiency of 75 percent for the compressor and 82 percent for the turbine, determine (*a*) the air temperature at the turbine exit, (*b*) the net work output, and (*c*) the thermal efficiency.

9–74 Reconsider Problem 9–73. Using EES (or other) software, allow the mass flow rate, pressure ratio, turbine inlet temperature, and the isentropic efficiencies of the turbine and compressor to vary. Assume the compressor inlet pressure is 100 kPa. Develop a general solution for the problem by taking advantage of the diagram window method for supplying data to EES software.

9–75 Repeat Problem 9–73 using constant specific heats at room temperature.

9–76 Air is used as the working fluid in a simple ideal Brayton cycle that has a pressure ratio of 12, a compressor inlet temperature of 300 K, and a turbine inlet temperature of 1000 K. Determine the required mass flow rate of air for a net power output of 70 MW, assuming both the compressor and the turbine have an isentropic efficiency of (*a*) 100 percent and (*b*) 85 percent. Assume constant specific heats at room temperature. *Answers:* (*a*) 352 kg/s, (*b*) 1037 kg/s

9–77 A stationary gas-turbine power plant operates on a simple ideal Brayton cycle with air as the working fluid. The air enters the compressor at 95 kPa and 290 K and the turbine at 760 kPa and 1100 K. Heat is transferred to air at a rate of 35,000 kJ/s. Determine the power delivered by this plant (*a*) assuming constant specific heats at room temperature and (*b*) accounting for the variation of specific heats with temperature.

9–78 Air enters the compressor of a gas-turbine engine at 300 K and 100 kPa, where it is compressed to 700 kPa and 580 K. Heat is transferred to air in the amount of 950 kJ/kg before it enters the turbine. For a turbine efficiency of 86 percent, determine (*a*) the fraction of the turbine work output used to drive the compressor and (*b*) the thermal efficiency. Assume variable specific heats for air.

9–79 Repeat Problem 9–78 using constant specific heats at room temperature.

9–80E A gas-turbine power plant operates on a simple Brayton cycle with air as the working fluid. The air enters the turbine at 120 psia and 2000 R and leaves at 15 psia and 1200 R. Heat is rejected to the surroundings at a rate of 6400 Btu/s, and air flows through the cycle at a rate of 40 lbm/s. Assuming the turbine to be isentropic and the compresssor to have an isentropic efficiency of 80 percent, determine the net power output of the plant. Account for the variation of specific heats with temperature. *Answer:* 3373 kW

9–81E For what compressor efficiency will the gas-turbine power plant in Problem 9–80E produce zero net work?

9–82 A gas-turbine power plant operates on the simple Brayton cycle with air as the working fluid and delivers 32 MW of power. The minimum and maximum temperatures in the cycle are 310 and 900 K, and the pressure of air at the compressor exit is 8 times the value at the compressor inlet. Assuming an isentropic efficiency of 80 percent for the compressor and 86 percent for the turbine, determine the mass flow rate of air through the cycle. Account for the variation of specific heats with temperature.

9–83 Repeat Problem 9–82 using constant specific heats at room temperature.

9–84 A gas-turbine power plant operates on the simple Brayton cycle between the pressure limits of 100 and 1200 kPa. The working fluid is air, which enters the compressor at 30°C at a rate of 150 m^3/min and leaves the turbine at 500°C. Using variable specific heats for air and assuming a compressor isentropic efficiency of 82 percent and a turbine isentropic efficiency of 88 percent, determine (*a*) the net power output, (*b*) the back work ratio, and (*c*) the thermal efficiency. *Answers:* (*a*) 659 kW, (*b*) 0.625, (*c*) 0.319

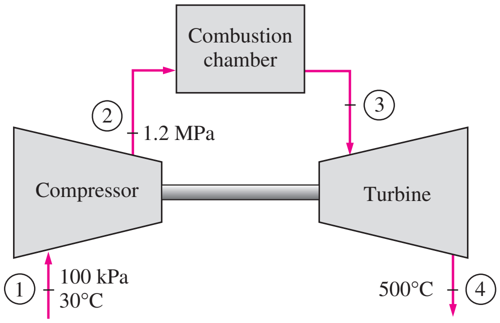

FIGURE P9–84

Brayton Cycle with Regeneration

9–85C How does regeneration affect the efficiency of a Brayton cycle, and how does it accomplish it?

9–86C Somebody claims that at very high pressure ratios, the use of regeneration actually decreases the thermal efficiency of a gas-turbine engine. Is there any truth in this claim? Explain.

9–87C Define the effectiveness of a regenerator used in gas-turbine cycles.

9–88C In an ideal regenerator, is the air leaving the compressor heated to the temperature at (*a*) turbine inlet, (*b*) turbine exit, (*c*) slightly above turbine exit?

9–89C In 1903, Aegidius Elling of Norway designed and built an 11-hp gas turbine that used steam injection between the combustion chamber and the turbine to cool the combustion gases to a safe temperature for the materials available at the time. Currently there are several gas-turbine power plants that use steam injection to augment power and improve thermal efficiency. For example, the thermal efficiency of the General Electric LM5000 gas turbine is reported to increase from 35.8 percent in simple-cycle operation to 43 percent when steam injection is used. Explain why steam injection increases the power output and the efficiency of gas turbines. Also, explain how you would obtain the steam.

9–90E The idea of using gas turbines to power automobiles was conceived in the 1930s, and considerable research was done in the 1940s and 1950s to develop automotive gas turbines by major automobile manufacturers such as the Chrysler and Ford corporations in the United States and Rover in the United Kingdom. The world's first gas-turbine-powered automobile, the 200-hp Rover Jet 1, was built in 1950 in the United Kingdom. This was followed by the production of the Plymouth Sport Coupe by Chrysler in 1954 under the leadership of G. J. Huebner. Several hundred gas-turbine-powered Plymouth cars were built in the early 1960s for demonstration purposes and were loaned to a select group of people to gather field experience. The users had no complaints other than slow acceleration. But the cars were never mass-produced because of the high production (especially material) costs and the failure to satisfy the provisions of the 1966 Clean Air Act.

A gas-turbine-powered Plymouth car built in 1960 had a turbine inlet temperature of 1700°F, a pressure ratio of 4, and a regenerator effectiveness of 0.9. Using isentropic efficiencies of 80 percent for both the compressor and the turbine, determine the thermal efficiency of this car. Also, determine the mass flow rate of air for a net power output of 95 hp. Assume the ambient air to be at 540 R and 14.5 psia.

9–91 The 7FA gas turbine manufactured by General Electric is reported to have an efficiency of 35.9 percent in the simple-cycle mode and to produce 159 MW of net power. The pressure ratio is 14.7 and the turbine inlet temperature is 1288°C. The mass flow rate through the turbine is 1,536,000 kg/h. Taking the ambient conditions to be 20°C and 100 kPa, determine the isentropic efficiency of the turbine and the compressor. Also, determine the thermal efficiency of this gas turbine if a regenerator with an effectiveness of 80 percent is added.

9–92 Reconsider Problem 9–91. Using EES (or other) software, develop a solution that allows different isentropic efficiencies for the compressor and turbine and study the effect of the isentropic efficiencies on net work done and the heat supplied to the cycle. Plot the *T-s* diagram for the cycle.

9–93 An ideal Brayton cycle with regeneration has a pressure ratio of 10. Air enters the compressor at 300 K and the turbine at 1200 K. If the effectiveness of the regenerator is 100 percent, determine the net work output and the thermal efficiency of the cycle. Account for the variation of specific heats with temperature.

9–94 Reconsider Problem 9–93. Using EES (or other) software, study the effects of varying the isentropic efficiencies for the compressor and turbine and regenerator effectiveness on net work done and the heat supplied to the cycle for the variable specific heat case. Plot the *T-s* diagram for the cycle.

9–95 Repeat Problem 9–93 using constant specific heats at room temperature.

9–96 A Brayton cycle with regeneration using air as the working fluid has a pressure ratio of 7. The minimum and maximum temperatures in the cycle are 310 and 1150 K. Assuming an isentropic efficiency of 75 percent for the compressor and

82 percent for the turbine and an effectiveness of 65 percent for the regenerator, determine (*a*) the air temperature at the turbine exit, (*b*) the net work output, and (*c*) the thermal efficiency. *Answers:* (*a*) 783 K, (*b*) 108.1 kJ/kg, (*c*) 22.5 percent

9–97 A stationary gas-turbine power plant operates on an ideal regenerative Brayton cycle (ϵ = 100 percent) with air as the working fluid. Air enters the compressor at 95 kPa and 290 K and the turbine at 760 kPa and 1100 K. Heat is transferred to air from an external source at a rate of 75,000 kJ/s. Determine the power delivered by this plant (*a*) assuming constant specific heats for air at room temperature and (*b*) accounting for the variation of specific heats with temperature.

9–98 Air enters the compressor of a regenerative gas-turbine engine at 300 K and 100 kPa, where it is compressed to 800 kPa and 580 K. The regenerator has an effectiveness of 72 percent, and the air enters the turbine at 1200 K. For a turbine efficiency of 86 percent, determine (*a*) the amount of heat transfer in the regenerator and (*b*) the thermal efficiency. Assume variable specific heats for air. *Answers:* (*a*) 152.5 kJ/kg, (*b*) 36.0 percent

9–99 Repeat Problem 9–98 using constant specific heats at room temperature.

9–100 Repeat Problem 9–98 for a regenerator effectiveness of 70 percent.

Brayton Cycle with Intercooling, Reheating, and Regeneration

9–101C Under what modifications will the ideal simple gas-turbine cycle approach the Ericsson cycle?

9–102C The single-stage compression process of an ideal Brayton cycle without regeneration is replaced by a multistage compression process with intercooling between the same pressure limits. As a result of this modification,

(*a*) Does the compressor work increase, decrease, or remain the same?

(*b*) Does the back work ratio increase, decrease, or remain the same?

(*c*) Does the thermal efficiency increase, decrease, or remain the same?

9–103C The single-stage expansion process of an ideal Brayton cycle without regeneration is replaced by a multistage expansion process with reheating between the same pressure limits. As a result of this modification,

(*a*) Does the turbine work increase, decrease, or remain the same?

(*b*) Does the back work ratio increase, decrease, or remain the same?

(*c*) Does the thermal efficiency increase, decrease, or remain the same?

9–104C A simple ideal Brayton cycle without regeneration is modified to incorporate multistage compression with intercooling and multistage expansion with reheating, without changing the pressure or temperature limits of the cycle. As a result of these two modifications,

(*a*) Does the net work output increase, decrease, or remain the same?

(*b*) Does the back work ratio increase, decrease, or remain the same?

(*c*) Does the thermal efficiency increase, decrease, or remain the same?

(*d*) Does the heat rejected increase, decrease, or remain the same?

9–105C A simple ideal Brayton cycle is modified to incorporate multistage compression with intercooling, multistage expansion with reheating, and regeneration without changing the pressure limits of the cycle. As a result of these modifications,

(*a*) Does the net work output increase, decrease, or remain the same?

(*b*) Does the back work ratio increase, decrease, or remain the same?

(*c*) Does the thermal efficiency increase, decrease, or remain the same?

(*d*) Does the heat rejected increase, decrease, or remain the same?

9–106C For a specified pressure ratio, why does multistage compression with intercooling decrease the compressor work, and multistage expansion with reheating increase the turbine work?

9–107C In an ideal gas-turbine cycle with intercooling, reheating, and regeneration, as the number of compression and expansion stages is increased, the cycle thermal efficiency approaches (*a*) 100 percent, (*b*) the Otto cycle efficiency, or (*c*) the Carnot cycle efficiency.

9–108 Consider an ideal gas-turbine cycle with two stages of compression and two stages of expansion. The pressure ratio across each stage of the compressor and turbine is 3. The air enters each stage of the compressor at 300 K and each stage of the turbine at 1200 K. Determine the back work ratio and the thermal efficiency of the cycle, assuming (*a*) no regenerator is used and (*b*) a regenerator with 75 percent effectiveness is used. Use variable specific heats.

9–109 Repeat Problem 9–108, assuming an efficiency of 80 percent for each compressor stage and an efficiency of 85 percent for each turbine stage.

9–110 Consider a regenerative gas-turbine power plant with two stages of compression and two stages of expansion. The overall pressure ratio of the cycle is 9. The air enters each stage of the compressor at 300 K and each stage of the turbine at 1200 K. Accounting for the variation of specific heats with temperature, determine the minimum mass flow rate of air needed to develop a net power output of 110 MW. *Answer:* 250 kg/s

9–111 Repeat Problem 9–110 using argon as the working fluid.

Jet-Propulsion Cycles

9–112C What is propulsive power? How is it related to thrust?

9–113C What is propulsive efficiency? How is it determined?

9–114C Is the effect of turbine and compressor irreversibilities of a turbojet engine to reduce (*a*) the net work, (*b*) the thrust, or (*c*) the fuel consumption rate?

9–115E A turbojet is flying with a velocity of 900 ft/s at an altitude of 20,000 ft, where the ambient conditions are 7 psia and 10°F. The pressure ratio across the compressor is 13, and the temperature at the turbine inlet is 2400 R. Assuming ideal operation for all components and constant specific heats for air at room temperature, determine (*a*) the pressure at the turbine exit, (*b*) the velocity of the exhaust gases, and (*c*) the propulsive efficiency.

9–116E Repeat Problem 9–115E accounting for the variation of specific heats with temperature.

9–117 A turbojet aircraft is flying with a velocity of 320 m/s at an altitude of 9150 m, where the ambient conditions are 32 kPa and −32°C. The pressure ratio across the compressor is 12, and the temperature at the turbine inlet is 1400 K. Air enters the compressor at a rate of 60 kg/s, and the jet fuel has a heating value of 42,700 kJ/kg. Assuming ideal operation for all components and constant specific heats for air at room temperature, determine (*a*) the velocity of the exhaust gases, (*b*) the propulsive power developed, and (*c*) the rate of fuel consumption.

9–118 Repeat Problem 9–117 using a compressor efficiency of 80 percent and a turbine efficiency of 85 percent.

9–119 Consider an aircraft powered by a turbojet engine that has a pressure ratio of 12. The aircraft is stationary on the ground, held in position by its brakes. The ambient air is at 27°C and 95 kPa and enters the engine at a rate of 10 kg/s. The jet fuel has a heating value of 42,700 kJ/kg, and it is burned completely at a rate of 0.2 kg/s. Neglecting the effect of the diffuser and disregarding the slight increase in mass at the engine exit as well as the inefficiencies of engine components, determine the force that must be applied on the brakes to hold the plane stationary. *Answer:* 9089 N

9–120 Reconsider Problem 9–119. In the problem statement, replace the inlet mass flow rate by an inlet volume flow rate of 9.063 m^3/s. Using EES (or other) software, investigate the effect of compressor inlet temperature in the range of –20 to 30°C on the force that must be applied to the brakes to hold the plane stationary. Plot this force as a function in compressor inlet temperature.

9–121 Air at 7°C enters a turbojet engine at a rate of 16 kg/s and at a velocity of 300 m/s (relative to the engine). Air is heated in the combustion chamber at a rate 15,000 kJ/s and it leaves the engine at 427°C. Determine the thrust produced by this turbojet engine. (*Hint:* Choose the entire engine as your control volume.)

Second-Law Analysis of Gas Power Cycles

9–122 Determine the total exergy destruction associated with the Otto cycle described in Problem 9–34, assuming a source temperature of 2000 K and a sink temperature of 300 K. Also, determine the exergy at the end of the power stroke. *Answers:* 245.12 kJ/kg, 145.2 kJ/kg

9–123 Determine the total exergy destruction associated with the Diesel cycle described in Problem 9–47, assuming a source temperature of 2000 K and a sink temperature of 300 K. Also, determine the exergy at the end of the isentropic compression process. *Answers:* 292.7 kJ/kg, 348.6 kJ/kg

9–124E Determine the exergy destruction associated with the heat rejection process of the Diesel cycle described in Problem 9–49E, assuming a source temperature of 3500 R and a sink temperature of 540 R. Also, determine the exergy at the end of the isentropic expansion process.

9–125 Calculate the exergy destruction associated with each of the processes of the Brayton cycle described in Problem 9–73, assuming a source temperature of 1600 K and a sink temperature of 290 K.

9–126 Determine the total exergy destruction associated with the Brayton cycle described in Problem 9–93, assuming a source temperature of 1800 K and a sink temperature of 300 K. Also, determine the exergy of the exhaust gases at the exit of the regenerator.

9–127 Reconsider Problem 9–126. Using EES (or other) software, investigate the effect of varying the cycle pressure ratio from 6 to 14 on the total exergy destruction for the cycle and the exergy of the exhaust gas leaving the regenerator. Plot these results as functions of pressure ratio. Discuss the results.

9–128 Determine the exergy destruction associated with each of the processes of the Brayton cycle described in Problem 9–98, assuming a source temperature of 1260 K and a sink temperature of 300 K. Also, determine the exergy of the exhaust gases at the exit of the regenerator. Take $P_{exhaust} = P_0 = 100$ kPa.

9–129 A gas-turbine power plant operates on the simple Brayton cycle between the pressure limits of 100 and 700 kPa. Air enters the compressor at 30°C at a rate of 12.6 kg/s and leaves at 260°C. A diesel fuel with a heating value of 42,000 kJ/kg is burned in the combustion chamber with an air–fuel ratio of 60 and a combustion efficiency of 97 percent. Combustion gases leave the combustion chamber and enter the turbine whose isentropic efficiency is 85 percent. Treating the combustion gases as air and using constant specific heats at 500°C, determine (*a*) the isentropic efficiency

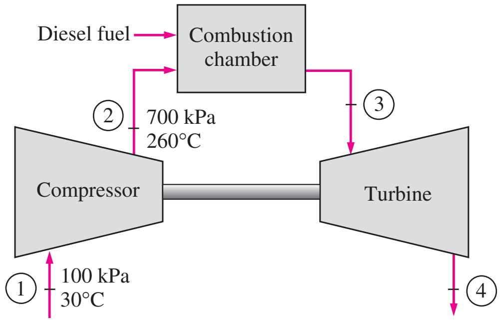

FIGURE P9–129

of the compressor, (*b*) the net power output and the back work ratio, (*c*) the thermal efficiency, and (*d*) the second-law efficiency.

9–130 A four-cylinder, four-stroke, 2.8-liter modern, high-speed compression-ignition engine operates on the ideal dual cycle with a compression ratio of 14. The air is at 95 kPa and 55°C at the beginning of the compression process and the engine speed is 3500 rpm. Equal amounts of fuel are burned at constant volume and at constant pressure. The maximum allowable pressure in the cycle is 9 MPa due to material strength limitations. Using constant specific heats at 850 K, determine (*a*) the maximum temperature in the cycle, (*b*) the net work output and the thermal efficiency, (*c*) the mean effective pressure, and (*d*) the net power output. Also, determine (*e*) the second-law efficiency of the cycle and the rate of exergy output with the exhaust gases when they are purged. *Answers:* (*a*) 3254 K, (*b*) 1349 kJ/kg, 0.587, (*c*) 1466 kPa, (*d*) 120 kW, (*e*) 0.646, 50.4 kW

9–131 A gas-turbine power plant operates on the regenerative Brayton cycle between the pressure limits of 100 and 700 kPa. Air enters the compressor at 30°C at a rate of 12.6 kg/s and leaves at 260°C. It is then heated in a regenerator to 400°C by the hot combustion gases leaving the turbine. A diesel fuel with a heating value of 42,000 kJ/kg is burned in the combustion chamber with a combustion efficiency of 97 percent. The combustion gases leave the combustion chamber at 871°C and enter the turbine whose isentropic efficiency is 85 percent. Treating combustion gases as air and using constant specific heats at 500°C, determine (*a*) the isentropic efficiency of the compressor, (*b*) the effectiveness of the regenerator, (*c*) the air–fuel ratio in the combustion chamber, (*d*) the net power output and the back work ratio, (*e*) the thermal efficiency, and (*f*) the second-law efficiency of the plant. Also determine (*g*) the second-law (exergetic) efficiencies of the compressor, the turbine, and the regenerator, and (*h*) the rate of the exergy flow with the combustion gases at the regenerator exit. *Answers:* (*a*) 0.881, (*b*) 0.632, (*c*) 78.1, (*d*) 2267 kW, 0.583, (*e*) 0.345, (*f*) 0.469, (*g*) 0.929, 0.932, 0.890, (*h*) 1351 kW

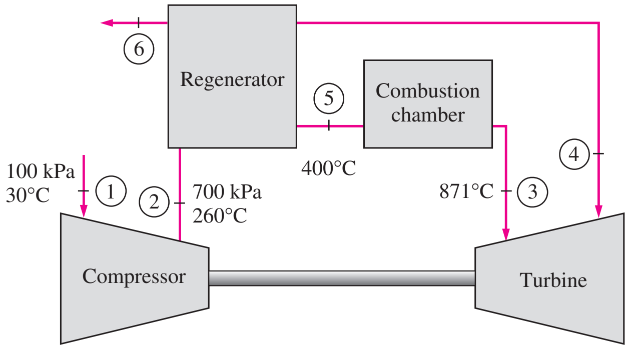

FIGURE P9–131

Review Problems

9–132 A four-stroke turbocharged V-16 diesel engine built by GE Transportation Systems to power fast trains produces 3500 hp at 1200 rpm. Determine the amount of power produced per cylinder per (*a*) mechanical cycle and (*b*) thermodynamic cycle.

9–133 Consider a simple ideal Brayton cycle operating between the temperature limits of 300 and 1500 K. Using constant specific heats at room temperature, determine the pressure ratio for which the compressor and the turbine exit temperatures of air are equal.

9–134 An air-standard cycle with variable coefficients is executed in a closed system and is composed of the following four processes:

1-2 v = *constant* heat addition from 100 kPa and 27°C to 300 kPa
2-3 P = *constant* heat addition to 1027°C
3-4 Isentropic expansion to 100 kPa
4-1 P = *constant* heat rejection to initial state

(*a*) Show the cycle on P-v and T-s diagrams.
(*b*) Calculate the net work output per unit mass.
(*c*) Determine the thermal efficiency.

9–135 Repeat Problem 9–134 using constant specific heats at room temperature.

9–136 An air-standard cycle with variable specific heats is executed in a closed system with 0.003 kg of air, and it consists of the following three processes:

1-2 Isentropic compression from 100 kPa and 27°C to 700 kPa
2-3 P = *constant* heat addition to initial specific volume
3-1 v = *constant* heat rejection to initial state

(*a*) Show the cycle on P-v and T-s diagrams.
(*b*) Calculate the maximum temperature in the cycle.
(*c*) Determine the thermal efficiency.
Answers: (*b*) 2100 K, (*c*) 15.8 percent

9–137 Repeat Problem 9–136 using constant specific heats at room temperature.

9–138 A Carnot cycle is executed in a closed system and uses 0.0025 kg of air as the working fluid. The cycle efficiency is 60 percent, and the lowest temperature in the cycle is 300 K. The pressure at the beginning of the isentropic expansion is 700 kPa, and at the end of the isentropic compression it is 1 MPa. Determine the net work output per cycle.

9–139 A four-cylinder spark-ignition engine has a compression ratio of 8, and each cylinder has a maximum volume of 0.6 L. At the beginning of the compression process, the air is at 98 kPa and 17°C, and the maximum temperature in the cycle is 1800 K. Assuming the engine to operate on the ideal Otto cycle, determine (*a*) the amount of heat supplied per cylinder, (*b*) the thermal efficiency, and (*c*) the number of revolutions per minute required for a net power output of 60 kW. Assume variable specific heats for air.

9–140 Reconsider Problem 9–139. Using EES (or other) software, study the effect of varying the compression ratio from 5 to 11 on the net work done and the efficiency of the cycle. Plot the *P*-v and *T*-*s* diagrams for the cycle, and discuss the results.

9–141 An ideal Otto cycle has a compression ratio of 9.2 and uses air as the working fluid. At the beginning of the compression process, air is at 98 kPa and 27°C. The pressure is doubled during the constant-volume heat-addition process. Accounting for the variation of specific heats with temperature, determine (*a*) the amount of heat transferred to the air, (*b*) the net work output, (*c*) the thermal efficiency, and (*d*) the mean effective pressure for the cycle.

9–142 Repeat Problem 9–141 using constant specific heats at room temperature.

9–143 Consider an engine operating on the ideal Diesel cycle with air as the working fluid. The volume of the cylinder is 1200 cm^3 at the beginning of the compression process, 75 cm^3 at the end, and 150 cm^3 after the heat-addition process. Air is at 17°C and 100 kPa at the beginning of the compression process. Determine (*a*) the pressure at the beginning of the heat-rejection process, (*b*) the net work per cycle, in kJ, and (*c*) the mean effective pressure.

9–144 Repeat Problem 9–143 using argon as the working fluid.

9–145E An ideal dual cycle has a compression ratio of 12 and uses air as the working fluid. At the beginning of the compression process, air is at 14.7 psia and 90°F, and occupies a volume of 75 in^3. During the heat-addition process, 0.3 Btu of heat is transferred to air at constant volume and 1.1 Btu at constant pressure. Using constant specific heats evaluated at room temperature, determine the thermal efficiency of the cycle.

9–146 Consider an ideal Stirling cycle using air as the working fluid. Air is at 350 K and 200 kPa at the beginning of the isothermal compression process, and heat is supplied to air from a source at 1800 K in the amount of 900 kJ/kg. Determine (*a*) the maximum pressure in the cycle and (*b*) the net work output per unit mass of air. *Answers:* (*a*) 5873 kPa, (*b*) 725 kJ/kg

9–147 Consider a simple ideal Brayton cycle with air as the working fluid. The pressure ratio of the cycle is 6, and the minimum and maximum temperatures are 300 and 1300 K, respectively. Now the pressure ratio is doubled without changing the minimum and maximum temperatures in the cycle. Determine the change in (*a*) the net work output per unit mass and (*b*) the thermal efficiency of the cycle as a result of this modification. Assume variable specific heats for air. *Answers:* (*a*) 41.5 kJ/kg, (*b*) 10.6 percent

9–148 Repeat Problem 9–147 using constant specific heats at room temperature.

9–149 Helium is used as the working fluid in a Brayton cycle with regeneration. The pressure ratio of the cycle is 8, the compressor inlet temperature is 300 K, and the turbine inlet temperature is 1800 K. The effectiveness of the regenerator is 75 percent. Determine the thermal efficiency and the required mass flow rate of helium for a net power output of 60 MW, assuming both the compressor and the turbine have an isentropic efficiency of (*a*) 100 percent and (*b*) 80 percent.

9–150 A gas-turbine engine with regeneration operates with two stages of compression and two stages of expansion. The pressure ratio across each stage of the compressor and turbine is 3.5. The air enters each stage of the compressor at 300 K and each stage of the turbine at 1200 K. The compressor and turbine efficiencies are 78 and 86 percent, respectively, and the effectiveness of the regenerator is 72 percent. Determine the back work ratio and the thermal efficiency of the cycle, assuming constant specific heats for air at room temperature. *Answers:* 53.2 percent, 39.2 percent

9–151 Reconsider Problem 9–150. Using EES (or other) software, study the effects of varying the isentropic efficiencies for the compressor and turbine and regenerator effectiveness on net work done and the heat supplied to the cycle for the variable specific heat case. Let the isentropic efficiencies and the effectiveness vary from 70 percent to 90 percent. Plot the *T*-*s* diagram for the cycle.

9–152 Repeat Problem 9–150 using helium as the working fluid.

9–153 Consider the ideal regenerative Brayton cycle. Determine the pressure ratio that maximizes the thermal efficiency of the cycle and compare this value with the pressure ratio that maximizes the cycle net work. For the same maximum-to-minimum temperature ratios, explain why the pressure ratio for maximum efficiency is less than the pressure ratio for maximum work.

9–154 Consider an ideal gas-turbine cycle with one stage of compression and two stages of expansion and regeneration.

The pressure ratio across each turbine stage is the same. The high-pressure turbine exhaust gas enters the regenerator and then enters the low-pressure turbine for expansion to the compressor inlet pressure. Determine the thermal efficiency of this cycle as a function of the compressor pressure ratio and the high-pressure turbine to compressor inlet temperature ratio. Compare your result with the efficiency of the standard regenerative cycle.

9–155 A four-cylinder, four-stroke spark-ignition engine operates on the ideal Otto cycle with a compression ratio of 11 and a total displacement volume of 1.8 liter. The air is at 90 kPa and 50°C at the beginning of the compression process. The heat input is 1.5 kJ per cycle per cylinder. Accounting for the variation of specific heats of air with temperature, determine (*a*) the maximum temperature and pressure that occur during the cycle, (*b*) the net work per cycle per cyclinder and the thermal efficiency of the cycle, (*c*) the mean effective pressure, and (*d*) the power output for an engine speed of 3000 rpm.

9–156 A gas-turbine plant operates on the regenerative Brayton cycle with two stages of reheating and two-stages of intercooling between the pressure limits of 100 and 1200 kPa. The working fluid is air. The air enters the first and the second stages of the compressor at 300 K and 350 K, respectively, and the first and the second stages of the turbine at 1400 K and 1300 K, respectively. Assuming both the compressor and the turbine have an isentropic efficiency of 80 percent and the regenerator has an effectiveness of 75 percent and using variable specific heats, determine (*a*) the back work ratio and the net work output, (*b*) the thermal efficiency, and (*c*) the second-law efficiency of the cycle. Also determine (*d*) the exergies at the exits of the combustion chamber (state 6) and the regenerator (state 10) (See Figure 9–43 in the text). *Answers:* (*a*) 0.523, 317 kJ/kg, (*b*) 0.553, (*c*) 0.704, (*d*) 931 kJ/kg, 129 kJ/kg

9–157 Electricity and process heat requirements of a manufacturing facility are to be met by a cogeneration plant consisting of a gas turbine and a heat exchanger for steam production. The plant operates on the simple Brayton cycle between the pressure limits of 100 and 1200 kPa with air as the working fluid. Air enters the compressor at 30°C. Combustion gases leave the turbine and enter the heat exchanger at 500°C, and leave the heat exchanger of 350°C, while the liquid water enters the heat exchanger at 25°C and leaves at 200°C as a saturated vapor. The net power produced by the gas-turbine cycle is 800 kW. Assuming a compressor isentropic efficiency of 82 percent and a turbine isentropic efficiency of 88 percent and using variable specific heats, determine (*a*) the mass flow rate of air, (*b*) the back work ratio and the thermal efficiency, and (*c*) the rate at which steam is produced in the heat exchanger. Also determine (*d*) the utilization efficiency of the cogeneration plant, defined as the ratio of the total energy utilized to the energy supplied to the plant.

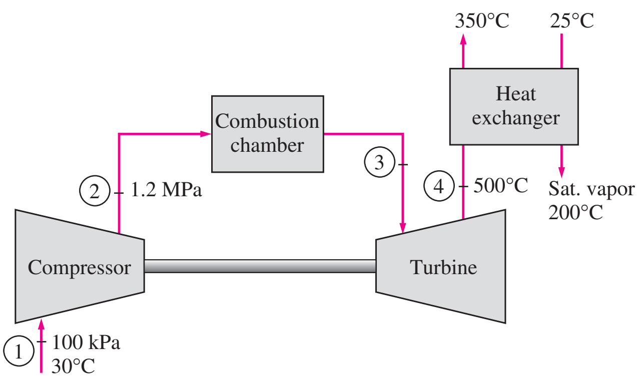

FIGURE P9–157

9–158 A turbojet aircraft flies with a velocity of 900 km/h at an altitude where the air temperature and pressure are −35°C and 40 kPa. Air leaves the diffuser at 50 kPa with a velocity of 15 m/s, and combustion gases enter the turbine at 450 kPa and 950°C. The turbine produces 500 kW of power, all of which is used to drive the compressor. Assuming an isentropic efficiency of 83 percent for the compressor, turbine, and nozzle, and using variable specific heats, determine (*a*) the pressure of combustion gases at the turbine exit, (*b*) the mass flow rate of air through the compressor, (*c*) the velocity of the gases at the nozzle exit, and (*d*) the propulsive power and the propulsive efficiency for this engine. *Answers:* (*a*) 147 kPa, (*b*) 1.76 kg/s, (*c*) 719 m/s, (*d*) 206 kW, 0.156

9–159 Using EES (or other) software, study the effect of variable specific heats on the thermal efficiency of the ideal Otto cycle using air as the working fluid. At the beginning of the compression process, air is at 100 kPa and 300 K. Determine the percentage of error involved in using constant specific heat values at room temperature for the following combinations of compression ratios and maximum cycle temperatures: r = 6, 8, 10, 12, and T_{max} = 1000, 1500, 2000, 2500 K.

9–160 Using EES (or other) software, determine the effects of compression ratio on the net work output and the thermal efficiency of the Otto cycle for a maximum cycle temperature of 2000 K. Take the working fluid to be air that is at 100 kPa and 300 K at the beginning of the compression process, and assume variable specific heats. Vary the compression ratio from 6 to 15 with an increment of 1. Tabulate and plot your results against the compression ratio.

9–161 Using EES (or other) software, determine the effects of pressure ratio on the net work output and the thermal efficiency of a simple Brayton cycle for a maximum cycle temperature of 1800 K. Take the working fluid to be air that is at 100 kPa and 300 K at the beginning of the compression process, and assume variable specific heats. Vary the pressure ratio from 5 to 24 with an increment of 1. Tabulate and plot your results against the pressure ratio. At what pressure ratio does the net work output become a maximum? At what pressure ratio does the thermal efficiency become a maximum?

9–162 Repeat Problem 9–161 assuming isentropic efficiencies of 85 percent for both the turbine and the compressor.

9–163 Using EES (or other) software, determine the effects of pressure ratio, maximum cycle temperature, and compressor and turbine efficiencies on the net work output per unit mass and the thermal efficiency of a simple Brayton cycle with air as the working fluid. Air is at 100 kPa and 300 K at the compressor inlet. Also, assume constant specific heats for air at room temperature. Determine the net work output and the thermal efficiency for all combinations of the following parameters, and draw conclusions from the results.

Pressure ratio:	5, 8, 14
Maximum cycle temperature:	800, 1200, 1600 K
Compressor isentropic efficiency:	80, 100 percent
Turbine isentropic efficiency:	80, 100 percent

9–164 Repeat Problem 9–163 by considering the variation of specific heats of air with temperature.

9–165 Repeat Problem 9–163 using helium as the working fluid.

9–166 Using EES (or other) software, determine the effects of pressure ratio, maximum cycle temperature, regenerator effectiveness, and compressor and turbine efficiencies on the net work output per unit mass and on the thermal efficiency of a regenerative Brayton cycle with air as the working fluid. Air is at 100 kPa and 300 K at the compressor inlet. Also, assume constant specific heats for air at room temperature. Determine the net work output and the thermal efficiency for all combinations of the following parameters.

Pressure ratio:	6, 10
Maximum cycle temperature:	1500, 2000 K
Compressor isentropic efficiency:	80, 100 percent
Turbine isentropic efficiency:	80, 100 percent
Regenerator effectiveness:	70, 90 percent

9–167 Repeat Problem 9–166 by considering the variation of specific heats of air with temperature.

9–168 Repeat Problem 9–166 using helium as the working fluid.

9–169 Using EES (or other) software, determine the effect of the number of compression and expansion stages on the thermal efficiency of an ideal regenerative Brayton cycle with multistage compression and expansion. Assume that the overall pressure ratio of the cycle is 12, and the air enters each stage of the compressor at 300 K and each stage of the turbine at 1200 K. Using constant specific heats for air at room temperature, determine the thermal efficiency of the cycle by varying the number of stages from 1 to 22 in increments of 3. Plot the thermal efficiency versus the number of stages. Compare your results to the efficiency of an Ericsson cycle operating between the same temperature limits.

9–170 Repeat Problem 9–169 using helium as the working fluid.

Fundamentals of Engineering (FE) Exam Problems

9–171 An Otto cycle with air as the working fluid has a compression ratio of 8.2. Under cold-air-standard conditions, the thermal efficiency of this cycle is

(*a*) 24 percent (*b*) 43 percent (*c*) 52 percent
(*d*) 57 percent (*e*) 75 percent

9–172 For specified limits for the maximum and minimum temperatures, the ideal cycle with the lowest thermal efficiency is

(*a*) Carnot (*b*) Stirling (*c*) Ericsson
(*d*) Otto (*e*) All are the same

9–173 A Carnot cycle operates between the temperature limits of 300 and 2000 K, and produces 600 kW of net power. The rate of entropy change of the working fluid during the heat addition process is

(*a*) 0 (*b*) 0.300 kW/K (*c*) 0.353 kW/K
(*d*) 0.261 kW/K (*e*) 2.0 kW/K

9–174 Air in an ideal Diesel cycle is compressed from 3 to 0.15 L, and then it expands during the constant pressure heat addition process to 0.30 L. Under cold air standard conditions, the thermal efficiency of this cycle is

(*a*) 35 percent (*b*) 44 percent (*c*) 65 percent
(*d*) 70 percent (*e*) 82 percent

9–175 Helium gas in an ideal Otto cycle is compressed from 20°C and 2.5 to 0.25 L, and its temperature increases by an additional 700°C during the heat addition process. The temperature of helium before the expansion process is

(*a*) 1790°C (*b*) 2060°C (*c*) 1240°C
(*d*) 620°C (*e*) 820°C

9–176 In an ideal Otto cycle, air is compressed from 1.20 kg/m^3 and 2.2 to 0.26 L, and the net work output of the cycle is 440 kJ/kg. The mean effective pressure (MEP) for this cycle is

(*a*) 612 kPa (*b*) 599 kPa (*c*) 528 kPa
(*d*) 416 kPa (*e*) 367 kPa

9–177 In an ideal Brayton cycle, air is compressed from 95 kPa and 25°C to 800 kPa. Under cold-air-standard conditions, the thermal efficiency of this cycle is

(*a*) 46 percent (*b*) 54 percent (*c*) 57 percent
(*d*) 39 percent (*e*) 61 percent

9–178 Consider an ideal Brayton cycle executed between the pressure limits of 1200 and 100 kPa and temperature limits of 20 and 1000°C with argon as the working fluid. The net work output of the cycle is

(*a*) 68 kJ/kg (*b*) 93 kJ/kg (*c*) 158 kJ/kg
(*d*) 186 kJ/kg (*e*) 310 kJ/kg

9–179 An ideal Brayton cycle has a net work output of 150 kJ/kg and a back work ratio of 0.4. If both the turbine and the compressor had an isentropic efficiency of 85 percent, the net work output of the cycle would be

(*a*) 74 kJ/kg (*b*) 95 kJ/kg (*c*) 109 kJ/kg
(*d*) 128 kJ/kg (*e*) 177 kJ/kg

9–180 In an ideal Brayton cycle, air is compressed from 100 kPa and 25°C to 1 MPa, and then heated to 1200°C before entering the turbine. Under cold-air-standard conditions, the air temperature at the turbine exit is

(*a*) 490°C (*b*) 515°C (*c*) 622°C
(*d*) 763°C (*e*) 895°C

9–181 In an ideal Brayton cycle with regeneration, argon gas is compressed from 100 kPa and 25°C to 400 kPa, and then heated to 1200°C before entering the turbine. The highest temperature that argon can be heated in the regenerator is

(*a*) 246°C (*b*) 846°C (*c*) 689°C
(*d*) 368°C (*e*) 573°C

9–182 In an ideal Brayton cycle with regeneration, air is compressed from 80 kPa and 10°C to 400 kPa and 175°C, is heated to 450°C in the regenerator, and then further heated to 1000°C before entering the turbine. Under cold-air-standard conditions, the effectiveness of the regenerator is

(*a*) 33 percent (*b*) 44 percent (*c*) 62 percent
(*d*) 77 percent (*e*) 89 percent

9–183 Consider a gas turbine that has a pressure ratio of 6 and operates on the Brayton cycle with regeneration between the temperature limits of 20 and 900°C. If the specific heat ratio of the working fluid is 1.3, the highest thermal efficiency this gas turbine can have is

(*a*) 38 percent (*b*) 46 percent (*c*) 62 percent
(*d*) 58 percent (*e*) 97 percent

9–184 An ideal gas turbine cycle with many stages of compression and expansion and a regenerator of 100 percent effectiveness has an overall pressure ratio of 10. Air enters every stage of compressor at 290 K, and every stage of turbine at 1200 K. The thermal efficiency of this gas-turbine cycle is

(*a*) 36 percent (*b*) 40 percent (*c*) 52 percent
(*d*) 64 percent (*e*) 76 percent

9–185 Air enters a turbojet engine at 260 m/s at a rate of 30 kg/s, and exits at 800 m/s relative to the aircraft. The thrust developed by the engine is

(*a*) 8 kN (*b*) 16 kN (*c*) 24 kN
(*d*) 20 kN (*e*) 32 kN

Design and Essay Problems

9–186 Design a closed-system air-standard gas power cycle composed of three processes and having a minimum thermal efficiency of 20 percent. The processes may be isothermal, isobaric, isochoric, isentropic, polytropic, or pressure as a linear function of volume. Prepare an engineering report describing your design, showing the system, P-v and T-s diagrams, and sample calculations.

9–187 Design a closed-system air-standard gas power cycle composed of three processes and having a minimum thermal efficiency of 20 percent. The processes may be isothermal, isobaric, isochoric, isentropic, polytropic, or pressure as a linear function of volume; however, the Otto, Diesel, Ericsson, and Stirling cycles may not be used. Prepare an engineering report describing your design, showing the system, P-v and T-s diagrams, and sample calculations.

9–188 Write an essay on the most recent developments on the two-stroke engines, and find out when we might be seeing cars powered by two-stroke engines in the market. Why do the major car manufacturers have a renewed interest in two-stroke engines?

9–189 In response to concerns about the environment, some major car manufacturers are currently marketing electric cars. Write an essay on the advantages and disadvantages of electric cars, and discuss when it is advisable to purchase an electric car instead of a traditional internal combustion car.

9–190 Intense research is underway to develop adiabatic engines that require no cooling of the engine block. Such engines are based on ceramic materials because of the ability of such materials to withstand high temperatures. Write an essay on the current status of adiabatic engine development. Also determine the highest possible efficiencies with these engines, and compare them to the highest possible efficiencies of current engines.

9–191 Since its introduction in 1903 by Aegidius Elling of Norway, steam injection between the combustion chamber and the turbine is used even in some modern gas turbines currently in operation to cool the combustion gases to a metallurgical-safe temperature while increasing the mass flow rate through the turbine. Currently there are several gas-turbine power plants that use steam injection to augment power and improve thermal efficiency.

Consider a gas-turbine power plant whose pressure ratio is 8. The isentropic efficiencies of the compressor and the turbine are 80 percent, and there is a regenerator with an effectiveness of 70 percent. When the mass flow rate of air through the compressor is 40 kg/s, the turbine inlet temperature becomes 1700 K. But the turbine inlet temperature is limited to 1500 K, and thus steam injection into the combustion gases is being considered. However, to avoid the complexities associated with steam injection, it is proposed to use excess air (that is, to take in much more air than needed for complete combustion) to lower the combustion and thus turbine inlet temperature while increasing the mass flow rate and thus power output of the turbine. Evaluate this proposal, and compare the thermodynamic performance of "high air flow" to that of a "steam-injection" gas-turbine power plant under the following design conditions: the ambient air is at 100 kPa and 25°C, adequate water supply is available at 20°C, and the amount of fuel supplied to the combustion chamber remains constant.