CHAPTER 6

RECIPROCATING INTERNAL COMBUSTION ENGINES

6.1 Introduction

Perhaps the best-known engine in the world is the reciprocating internal combustion (IC) engine. Virtually every person who has driven an automobile or pushed a power lawnmower has used one. By far the most widely used IC engine is the *spark-ignition gasoline engine,* which takes us to school and work and on pleasure jaunts. Although others had made significant contributions, Niklaus Otto is generally credited with the invention of the engine and with the statement of its theoretical cycle.

Another important engine is the reciprocating engine that made the name of Rudolf Diesel famous. The Diesel engine, the workhorse of the heavy truck industry, is widely used in industrial power and marine applications. It replaced the reciprocating steam engine in railroad locomotives about fifty years ago and remains dominant in that role today.

The piston, cylinder, crank, and connecting rod provide the geometric basis of the reciprocating engine. While two-stroke-cycle engines are in use and of continuing interest, the discussion here will emphasize the more widely applied four-stroke-cycle engine. In this engine the piston undergoes two mechanical cycles for each thermodynamic cycle. The intake and compression processes occur in the first two strokes, and the power and exhaust processes in the last two. These processes are made possible by the crank-slider mechanism, discussed next.

6.2 The Crank-Slider Mechanism

Common to most *reciprocating engines* is a linkage known as a crank-slider mechanism. Diagramed in Figure 6.1, this mechanism is one of several capable of producing the straight-line, backward-and-forward motion known as reciprocating. Fundamentally, the crank-slider converts rotational motion into linear motion, or vice-versa. With a piston as the slider moving inside a fixed cylinder, the mechanism provides the vital capability of a gas engine: the ability to compress and expand a gas. Before delving into this aspect of the engine, however, let us examine the crank-slider mechanism more closely.

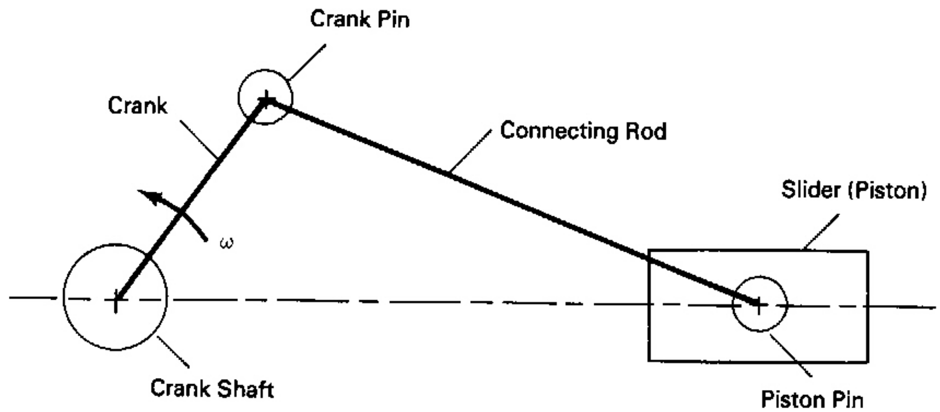

FIGURE 6.1 Crank-slider mechanism.

It is evident from Figure 6.2 that, while the crank arm rotates through 180°, the piston moves from the position known as *top-center* (TC) to the other extreme, called *bottom-center* (BC). During this period the piston travels a distance, S, called the *stroke,* that is twice the length of the crank.

For an angular velocity of the crank, ω, the crank pin A has a tangential velocity component $\omega S/2$. It is evident that, at TC and at BC, the crank pin velocity component in the piston direction, and hence the piston velocity, is zero. At these points, corresponding to crank angle $\theta = 0°$ and 180°, the piston reverses direction. Thus as θ varies from 0° to 180°, the piston velocity accelerates from 0 to a maximum and then returns to 0. A similar behavior exists between 180° and 360°.

The connecting rod is a two-force member; hence it is evident that there are both axial and lateral forces on the piston at crank angles other than 0° and 180°. These lateral forces are, of course, opposed by the cylinder walls. The resulting lateral force component normal to the cylinder wall gives rise to frictional forces between the piston rings and cylinder. It is evident that the normal force, and thus the frictional force, alternates from one side of the piston to the other during each cycle. Thus the piston motion presents a challenging lubrication problem for the control and reduction of both wear and energy loss.

The position of the piston with respect to the crank centerline is given by

$$x = (S/2)\cos\theta + L\cos\phi \qquad [\text{ft} \mid \text{m}] \qquad (6.1)$$

where $y_A = (S/2)\sin\theta = L\sin\phi$ can be used to eliminate ϕ to obtain

$$x/L = (S/2L)\cos\theta + [1 - (S/2L)^2 \sin^2\theta]^{1/2} \qquad [\text{dl}] \qquad (6.2)$$

Thus, while the axial component of the motion of the crank pin is simple harmonic, $x_A = (S/2)\cos\theta$, the motion of the piston and piston pin is more complex. It may be

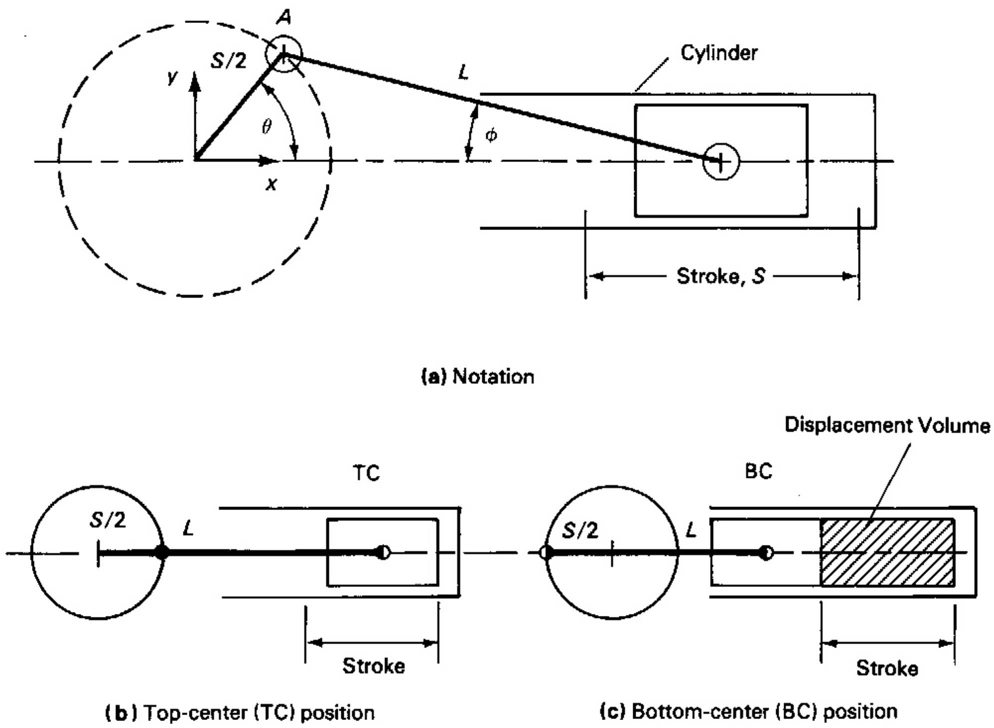

FIGURE 6.2 Geometry and notation for the crank-slider.

seen from Equation (6.2), however, that as *S/L* becomes small, the piston motion approaches simple harmonic. This becomes physically evident when it is recognized that, in this limit, the connecting rod angle, ϕ, approaches 0 and the piston motion approaches the axial motion of the crank pin. Equations (6.1) and (6.2) may be used to predict component velocities, accelerations, and forces in the engine.

The volume swept by the piston as it passes from TC to BC is called the *piston displacement,* disp. *Engine displacement,* DISP, is then the product of the piston displacement and the number of cylinders, DISP = (n)(disp). The piston displacement is the product of the piston cross-sectional area and the stroke. The cylinder inside diameter (and, approximately, also the piston diameter) is called its *bore*. Cylinder bore, stroke, and number of cylinders are usually quoted in engine specifications along with or instead of engine displacement. It will be seen later that the power output of a reciprocating engine is proportional to its displacement. An engine of historical interest that also used the crank-slider mechanism is discussed in the next section.

6.3 The Lenoir Cycle

An early form of the reciprocating internal combustion engine is credited to Etienne Lenoir. His engine, introduced in 1860, used a crank-slider-piston-cylinder arrangement

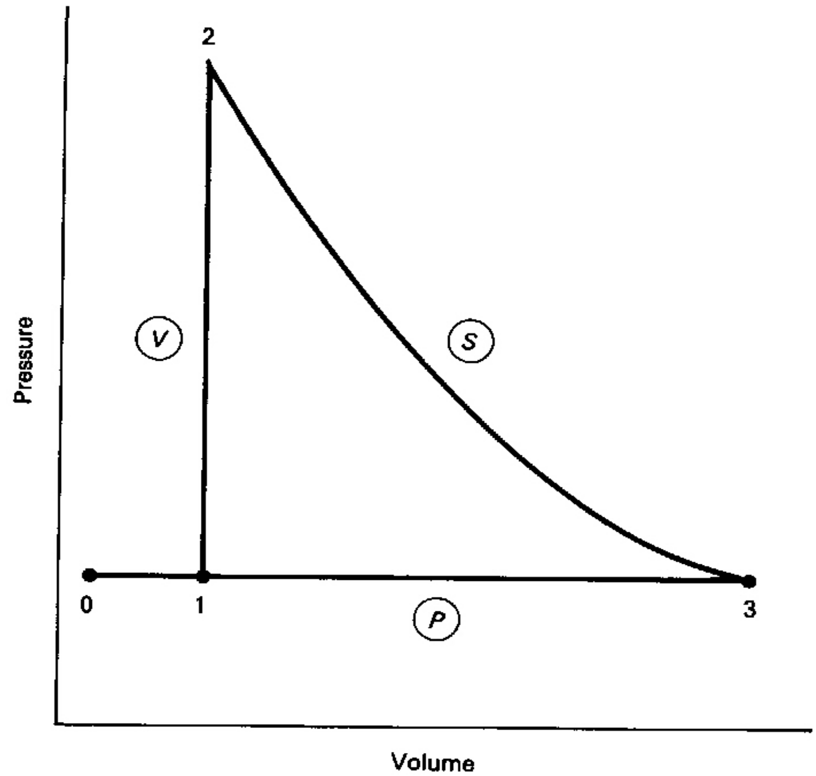

FIGURE 6.3 The Lenoir cycle.

in which a combustible mixture confined between the piston and cylinder is ignited after TC. The resulting combustion gas pressure forces acting on the piston deliver work by way of the connecting rod to the rotating crank. When the piston is at BC, combustion gases are allowed to escape. The rotational momentum of the crank system drives the piston toward TC, expelling additional gases as it goes. A fresh combustible mixture is again admitted to the combustion chamber (cylinder) and the cycle is repeated.

The theoretical *Lenoir cycle*, shown in Figure 6.3 on a pressure-volume diagram, consists of the intake of the working fluid (a combustible mixture) from state 0 to state 1, a constant-volume temperature and pressure rise from state 1 to state 2, approximating the combustion process, an isentropic expansion of the combustion gases to state 3, and a constant-pressure expulsion of residual gases back to state 0. Note that a portion of the piston displacement, from state 0 to state 1, is used to take in the combustible mixture and does not participate in the power stroke from state 2 to state 3. The engine has been called an explosion engine because the power delivered is due only to the extremely rapid combustion pressure rise or explosion of the mixture in the confined space of the cylinder.

Hundreds of Lenoir engines were used in the nineteenth century, but the engine is quite inefficient by todays standards. In 1862, Beau de Rochas pointed out that the

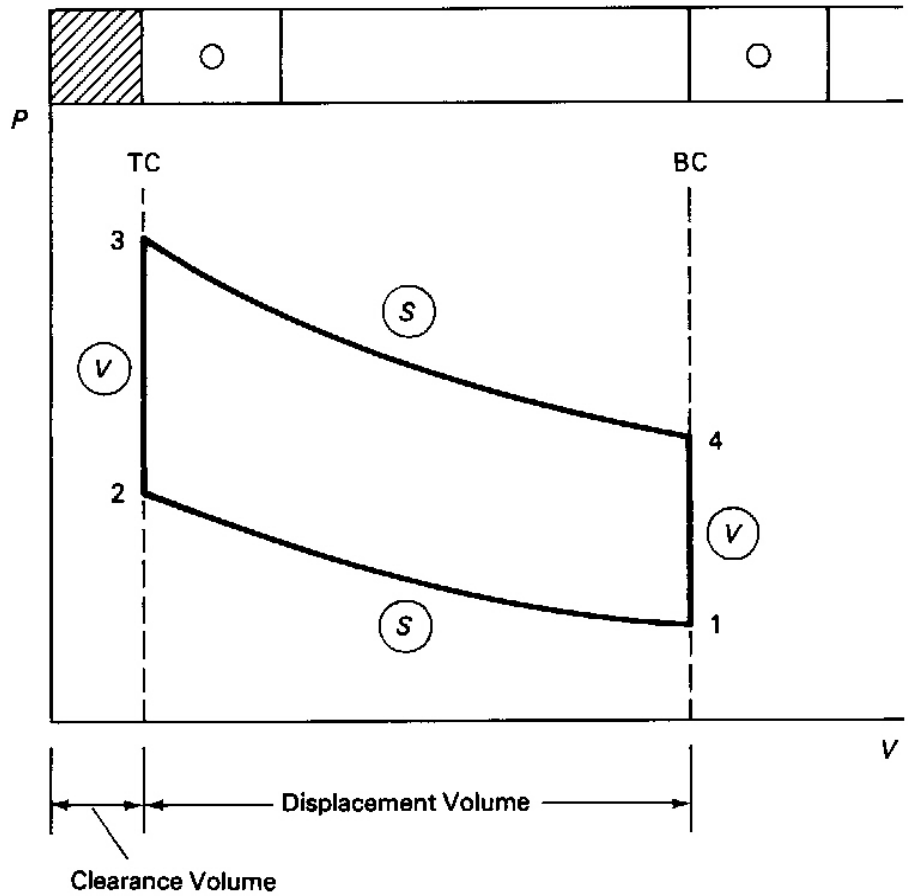

FIGURE 6.4 The Otto cycle.

efficiency of internal combustion could be markedly improved in reciprocating engines by compression of the air-fuel mixture prior to combustion. In 1876 Niklaus Otto (who is thought to have been unaware of Rochas' suggestion) demonstrated an engine that incorporated this important feature, as described next.

6.4 The Otto Cycle

The Otto cycle is the theoretical cycle commonly used to represent the processes in the spark ignition (SI) internal combustion engine. It is assumed that a fixed mass of working fluid is confined in the cylinder by a piston that moves from BC to TC and back, as shown in Figure 6.4. The cycle consists of isentropic compression of an air-fuel mixture from state 1 to state 2, constant-volume combustion to state 3, isentropic expansion of the combustion gases to state 4, and a constant-volume heat rejection back to state 1. The constant-volume heat rejection is a simple expedient to close the cycle. It obviates the need to represent the complex expansion and outflow of combustion gases from the cylinder at the end of the cycle. Note that the Otto cycle is not concerned with the induction of the air-fuel mixture or with the expulsion of residual combustion gases. Thus only two mechanical strokes of the crank-slider are needed in the Otto cycle, even when it is used to represent an ideal four-stroke-cycle Otto engine. In this case the remaining strokes are used to execute the necessary intake and exhaust functions. Because it involves only two strokes, the Otto cycle may also represent a two-stroke-cycle engine. The two-stroke-cycle engine is in principle capable of as much work in one rotation of the crank as the four-stroke engine is in two. However, it is difficult to implement because of the necessity of making the intake and exhaust functions a part of those two strokes. It is therefore not as highly developed or widely used as the four-stroke-cycle engine. We will focus on the four-stroke-cycle here.

The simplest analysis of the Otto cycle assumes calorically perfect air as the working fluid in what is called the *Air Standard cycle analysis*. Following the notation of Figure 6.4, the compression process can be represented by the isentropic relation for a calorically perfect gas, Equation (1.21), as

$$p_2/p_1 = (V_1/V_2)^k \qquad \text{[dl]} \qquad (6.3)$$

where the *compression ratio*, CR = V_1/V_2, is a fundamental parameter of all reciprocating engines. The diagram shows that the expansion ratio for the engine, V_4/V_3, has the same value, V_1/V_2. The *clearance volume*, V_2, is the volume enclosed between the cylinder head and the piston at TC. Thus the compression ratio may be expressed as the ratio of the sum of the clearance and displacement volumes to the clearance volume:

$$\mathrm{CR} = [V_2 + (V_1 - V_2)]/V_2$$

Thus, for a given displacement, the compression ratio may be increased by reducing the clearance volume.

The efficiency of the cycle can be most easily determined by considering constant-volume-process heat transfers and the First Law cyclic integral relation, Equation (1.3). The heat transferred in the processes 2→3 and 4→1 are

$$q_{2\rightarrow3} = c_v(T_3 - T_2) \qquad \text{[Btu/lb}_m \text{ | kj/kg]} \qquad (6.4)$$

and

$$q_{4\rightarrow1} = c_v(T_1 - T_4) \qquad \text{[Btu/lb}_m \text{ | kJ/kg]} \qquad (6.5)$$

Both the expansion process, 3→4, and the compression process, 1→2, are assumed to be isentropic. Thus, by definition, they are both adiabatic. From the cyclic integral, the net work per unit mass is then:

$$w = q_{2\rightarrow3} + q_{4\rightarrow1} = c_v(T_3 - T_2 + T_1 - T_4) \qquad \text{[Btu/lb}_m \text{ | kJ/kg]} \qquad (6.6)$$

As before, the cycle thermal efficiency is the ratio of the net work to the external heat supplied:

$$\eta_{\text{Otto}} = w/q_{2\to3} = c_v\,(T_3 - T_2 + T_1 - T_4)\,/\,[c_v\,(T_3 - T_2)]$$

$$= 1 + (T_1 - T_4)\,/\,(T_3 - T_2)$$

$$= 1 - T_1/T_2 = 1 - 1\,/\,\text{CR}^{\,k-1} \qquad [\text{dl}] \qquad (6.7)$$

where Equation (1.20) has been used to eliminate the temperatures. Equation (6.7) shows that increasing compression ratio increases the cycle thermal efficiency. This is true for real engines as well as for the idealized Otto engine. The ways in which real spark ignition engine cycles deviate from the theoretical Otto cycle are discussed later.

EXAMPLE 6.1

An Otto engine takes in an air-fuel mixture at 80°F and standard atmosphere presssure. It has a compression ratio of 8. Using Air Standard cycle analysis, a heating value of 20,425 Btu/lb_m, and A/F = 15, determine:

(a) The temperature and pressure at the end of compression, after combustion, and at the end of the power stroke.

(b) The net work per pound of working fluid.

(c) The thermal efficiency.

Solution

We use the notation of Figure 6.4:

(a) $p_2 \;=\; p_1(V_1/V_2)^k = 1(8)^{1.4} = 18.38$ atm

$$T_2 = T_1(V_1/V_2)^{k-1} = (540)(8)^{0.4} = 1240.6°\text{R}$$

$$T_3 = T_2 + q_a/c_v = T_2 + (\text{F/A})(\text{HV})k/c_p = 1240.6 + 1.4\cdot 20{,}425/15\cdot 0.24 = 9184°\text{R}$$

$$p_3 = p_2 T_3/T_2 = 18.38(9184/1240.6) = 136.1 \text{ atm}$$

$$T_4 = T_3/\text{CR}^{k-1} = 9184/\,8^{0.4} = 3997.2°\text{R}$$

$$p_4 = p_3/\text{CR}^k = 136.1/8^{1.4} = 7.4 \text{ atm}$$

(b) The constant-volume heat addition is governed by the fuel-air ratio and the fuel heating value:

$$q_a = \text{HV(F/A)} = 20{,}425/15 = 1361.7 \text{ Btu/lb}_m \text{ of air}$$

$q_r = c_v\,(T_1 - T_4) = (0.24/1.4)(\,540 - 3997.4) = -592.7$ Btu/lb$_m$

$w = q_a + q_r = 1361.7 + (-592.7) = 769$ Btu/lb$_m$

(c) The cycle termal efficiency may then be determined from the definition of the heat engine thermal efficiency or Equation (6.7):

$\eta_{th} = w/q_a = 769/1361.7 = 0.565$

$\eta_{th} = 1 - 1/8^{0.4} = 0.565$

In view of the discussion of gas properties and dissociation in Chapter 3, the values of T_3 and T_4 in Example 6.1 are unrealistically high. Much of the energy released by the fuel would go into vibration and dissociation of the gas molecules rather than into the translational and rotational degrees of freedom represented by the temperature. As a result, significantly lower temperatures would be obtained. Thus, while the analysis is formally correct, the use of constant-low-temperature heat capacities in the Air Standard cycle makes it a poor model for predicting temperature extremes when high energy releases occur. Some improvement is achieved by using constant-high-temperature heat capacities, but the best results would be achieved by the use of real gas properties, as discussed in several of the references.

6.5 Combustion in a Reciprocating Engine

The constant-volume heat transfer process at TC in the Otto cycle is an artifice to avoid the difficulties of modeling the complex processes that take place in the combustion chamber of the SI engine. These processes, in reality, take place over a crank angle span of 30° or more around TC. Let us consider aspects of these processes and their implementation in more detail.

Normally, the mixture in the combustion chamber must have an air-fuel ratio in the neighborhood of the stoichiometric value for satisfactory combustion. A more or less homogeneous mixture may be produced outside the cylinder in a carburetor, by injection into the intake manifold, or by throttle-body injection into a header serving several intake manifolds. In the case of the carburetor, fuel is drawn into the engine from the carburetor by the low pressure created in a venturi through which the combustion air flows. As a result, increased air flow causes lower venturi pressure and hence increased fuel flow. The fuel system thus serves to provide an air-fuel mixture that remains close to the stoichiometric ratio for a range of air flow rates. Various devices designed into the carburetor further adjust the fuel flow for the special operating conditions encountered, such as idling and rapid acceleration.

Maximum fuel economy is usually attained with excess air to ensure that all of the fuel is burned. A mixture with excess air is called a *lean mixture*. The carburetor usually produces this condition in automobiles during normal constant-speed driving.

On the other hand, maximum power is achieved with excess fuel to assure that all of the oxygen in the air in the combustion chamber is reacted. It is a matter of exploiting the full power-producing capability of the displacement volume. A mixture with excess fuel is called a *rich mixture*. The automotive carburetor produces a rich mixture during acceleration by supplying extra fuel to the air entering the intake manifold.

The equivalence ratio is sometimes used to characterize the mixture ratio, whether rich or lean. The *equivalence ratio*, Φ, is defined as the ratio of the actual fuel-air ratio to the stoichiometric fuel-air ratio. Thus $\Phi > 1$ represents a rich mixture and $\Phi < 1$ represents a lean mixture. In terms of air-fuel ratio, $\Phi = (A/F)_{stoich}/(A/F)$.

Homogeneous air-fuel mixtures close to stoichiometric may ignite spontaneously (that is, without a spark or other local energy source) if the mixture temperature exceeds a temperature called the *autoignition temperature*. If the mixture is brought to and held at a temperature higher than the autoignition temperature, there is a period of delay before spontaneous ignition or autoignition This time interval is called the *ignition delay*, or *ignition lag*. The ignition delay depends on the characteristics of the fuel and the equivalence ratio and usually decreases with increasing temperature.

In spark-ignition engines, compression ratios and therefore the temperatures at the end of compression are low enough that the air-fuel mixture is ignited by the spark plug before spontaneous ignition can occur. SI engines are designed so that a flame front will propagate smoothly from the spark plug into the unburned mixture until all of the mixture has been ignitied. However, as the flame front progresses, the temperature and pressure of the combustion gases behind it rise due to the release of the chemical energy of the fuel. As the front propagates, it compresses and heats the unburned mixture, sometimes termed the *end-gas*. Combustion is completed as planned when the front smoothly passes completely through the end-gas without autoignition. However, if the end-gas autoignites, a pinging or low-pitched sound called *knock* is heard.

The avoidance of knock due to autoignition of the end-gas is a major constraint on the *design compression ratio* of an SI engine. If hot spots or thermally induced compression of the end-gas ignite it before the flame front does, there is a more rapid release of chemical energy from the end-gas than during normal combustion. Knock is sometimes thought of as an explosion of the end gas that creates an abrupt pulse and pressure waves that race back and forth across the cylinder at high speed, producing the familiar pinging or low-pitched sound associated with knock. Knock not only reduces engine performance but produces rapid wear and objectionable noise in the engine. Thus it is important for a SI engine fuel to have a high autoignition temperature. It is therefore important for SI engine fuel to have a high autoignition temperature. Thus the knock characteristics of commercially available fuels limit the maximum allowable design compression ratio for SI engines and hence limit their best efficiency.

The *octane number* is a measure of a gasoline's ability to avoid knock. Additives such as tetraethyl lead have been used in the past to suppress engine knock. However, the accumulation of lead in the environment and its penetration into the food cycle has resulted in the phaseout of lead additives. Instead refineries now use appropriate blends of hydrocarbons as a substitute for lead additives in unleaded fuels.

The octane number of a fuel is measured in a special variable-compression-ratio engine called a CFR (Cooperative Fuels Research) engine. The octane rating of a fuel is determined by comparison of its knocking characteristics with those of different mixtures of isooctane, C_8H_{18}, and *n*-heptane, C_7H_{16}. One hundred percent isooctane is defined as having an octane number of 100 because it had the highest resistance to knock at the time the rating system was devised. On the other hand, *n*-heptane is assigned a value of 0 on the octane number scale because of its very poor knock resistance. If a gasoline tested in the CFR engine has the same knock threshold as a blend of 90% isooctane and 10% *n*-heptane, the fuel is assigned an octane rating of 90.

In combustion chamber design, the designer attempts to balance many factors to achieve good performance. Design considerations include locating intake valves away from and exhaust valves near spark plugs, to keep end-gas in a relatively cool area of the combustion chamber and thereby suppress hot-surface-induced autoignition tendencies. Valves are, of course, designed as large as possible to reduce induction and exhaust flow restrictions. More than one intake and one exhaust valve per cylinder are now used in some engines to improve "engine breathing." In some engines, four valves in a single cylinder are employed for this purpose. The valves are also designed to induce swirl and turbulence to promote mixing of fuel and air and to improve combustion stability and burning rate.

Pollution and fuel economy considerations have in recent years profoundly influenced overall engine and combustion chamber design. Stratified-charge engines, for example, attempt to provide a locally rich combustion region to control peak temperatures and thus suppress NO_x formation. The resulting combustion gases containing unburned fuel then mix with surrounding lean mixture to complete the combustion process, thus eliminating CO and unburned hydrocarbons from the exhaust. These processes occur at lower temperatures than in conventional combustion chamber designs and therefore prevent significant nitrogen reactions.

6.6 Representing Reciprocating Engine Perfomance

In an earlier section, the theoretical work per unit mass of working fluid of the Otto engine was evaluated for a single cycle of the engine, using the cyclic integral of the First Law of Thermodynamics. The work done by pressure forces acting on a piston can also be evaluated as the integral of pdV. It is evident therefore that the work done during a single engine cycle is the area enclosed by the cycle process curves on the pressure-volume diagram. Thus, instead of using the cyclic integral or evaluating pdV for each process of the cycle, the work of a reciprocating engine can be found by drawing the theoretical process curves on the p–V diagram and graphically integrating them. Such a plot of pressure versus volume for any reciprocating engine, real or theoretical, is called an *indicator diagram*.

In the nineteenth and early twentieth centuries a mechanical device known as an engine indicator was used to produce indicator cards or diagrams to determine the work per cycle for slow-running steam and gas reciprocating engines. The indicator card was attached to a cylinder that rotated back and forth on its axis as the piston oscillated, thus generating a piston position (volume) coordinate. At the same time a pen driven by a pressure signal from the engine cylinder moved parallel to the cylinder axis, scribing the p-V diagram over and over on the card. The work of high speed engines is still evaluated from traces of pressure obtained with electronic sensors and displayed on electronic monitors and through digital techniques.

The work done per cycle (from an indicator card, for instance) can be represented as an average pressure times a volume. Because the displacement volumes of engines are usually known, an engine performance parameter known as the *mean effective pressure*, MEP, is defined in terms of the piston displacement. The mean effective pressure is defined as the value of the pressure obtained by dividing the net work per cylinder per cycle at a given operating condition by the piston displacement volume:

$$\mathrm{MEP} = W/\mathrm{disp} \qquad [\mathrm{lb_f/ft^2} \mid \mathrm{kPa}] \tag{6.8}$$

Thus the MEP is a measure of the effectiveness of a given displacement volume in producing net work.

The power output of an engine with identical cylinders may be represented as the product of the work per cycle and the number of cycles executed per unit time by the engine. Thus if the engine has n cylinders, each executing N identical thermodynamic cycles per unit time, and delivering W work units per cylinder, with a piston displacement, disp, the power output is given by

$$P = n \cdot N \cdot W = n \cdot N \cdot \mathrm{MEP} \cdot \mathrm{disp} \qquad [\mathrm{ft\text{-}lb_f/min} \mid \mathrm{kW}] \tag{6.9}$$

Expressed for the entire engine, the engine displacement is DISP = n·disp and the engine work is MEP ·DISP. Hence the engine power is:

$$P = N \cdot \mathrm{MEP} \cdot \mathrm{DISP} \qquad [\mathrm{ft\text{-}lb_f/min} \mid \mathrm{kW}] \tag{6.10}$$

where N, the number of thermodynamic cycles of a cylinder per unit time, is the number of crank-shaft revolutions per unit time for a *two-stroke-cycle engine* and one-half of the revolutions per unit time for a *four-stroke-cycle engine*. The factor of ½ for the four-stroke-cycle engine arises because one thermodynamic cycle is executed each time the crank rotates through two revolutions.

EXAMPLE 6.2

What is the displacement of an engine that develops 60 horsepower at 2500 rpm in a four-stroke-cycle engine having an MEP of 120 psi?

Solution

From Equation (6.10), the displacement of the engine is

$$\text{DISP} = P/(N \cdot \text{MEP}) = (60)(33{,}000)(12)/[(2500/2)(120)] = 158.4 \text{ in}^3$$

Checking units: $(\text{HP})(\text{ft-lb}_f/\text{HP-min})(\text{in/ft})/[(\text{cycles/min})(\text{lb}_f/\text{in}^2)] = \text{in}^3$

If the work is evaluated from an indicator diagram the work is called *indicated work*; the MEP is called the *indicated mean effective pressure*, IMEP; and the power is *indicated power, IP*. Note that the indicated work and power, being associated with the work done by the combustion chamber gases on the piston, do not account for frictional or mechanical losses in the engine, such as piston-cylinder friction or the drag of moving parts (like connecting rods) as they move through air or lubricating oil.

Brake Performance Parameters

Another way of evaluating engine performance is to attach the engine output shaft to a device known as a *dynamometer,* or *brake*. The dynamometer measures the torque, T, applied by the engine at a given rotational speed. The power is then calculated from the relation

$$P = 2\pi \cdot \text{rpm} \cdot T \qquad [\text{ft-lb}_f/\text{min} \mid \text{N-m/min}] \qquad (6.11)$$

A simple device called a prony brake, which was used in the past, demonstrates the concept for the measurement of the shaft torque of engines. Figure 6.5 shows the prony brake configuration in which a stationary metal band wrapped around the rotating flywheel of the engine resists the torque transmitted to it by friction. The product of the force measured by a spring scale, w, and the moment arm, d , gives the resisting torque. The power dissipated is then given by $2\pi(\text{rpm})w \cdot d$.

Modern devices such as water brakes and electrical dynamometers long ago replaced the prony brake. The *water brake* is like a centrifugal water pump with no outflow, mounted on low-friction bearings, and driven by the test engine. As with the prony brake, the force required to resist turning of the brake (pump) housing provides the torque data. This, together with speed measurement, yields the power output from Equation (6.11). The power dissipated appears as increased temperature of the water in the brake and heat transfer from the brake. Cool water is circulated slowly through the brake to maintain a steady operating condition. The torque measured in this way is called the *brake torque*, BT, and the resulting power is called the *brake power*, BP. To summarize: while *indicated* parameters relate to gas forces in the cylinder, *brake* parameters deal with output shaft forces.

Thus the brake power differs from the indicated power in that it accounts for the effect of all of the energy losses in the engine. The difference between the two is referred to as the *friction power,* FP. Thus FP = IP – BP.

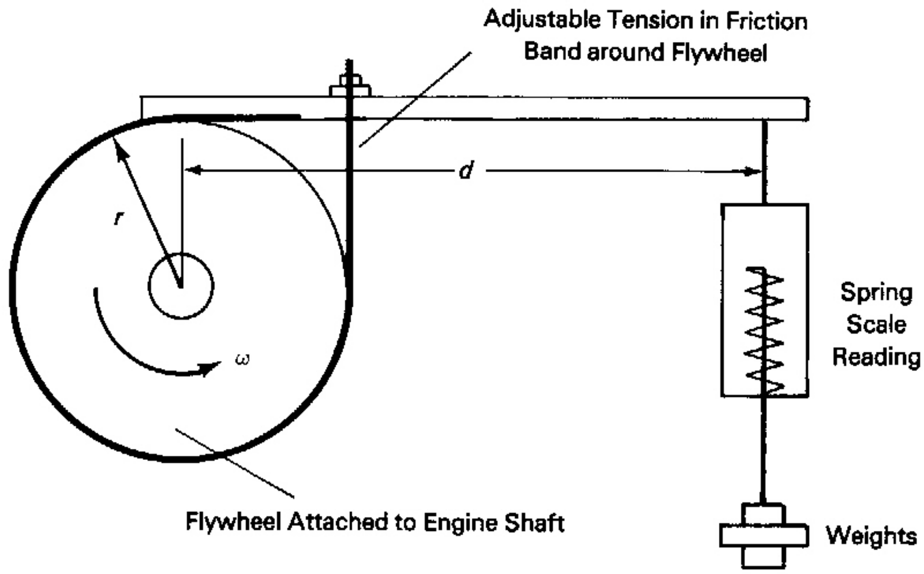

FIGURE 6.5 The prony brake.

Friction power varies with engine speed and is difficult to measure directly. An engine is sometimes driven without fuel by a motor-dynamometer to evaluate friction power. An alternative to using friction power to relate brake and indicated power is through the *engine mechanical efficiency*, η_m:

$$\eta_m = BP/IP \qquad [dl] \qquad (6.12)$$

Because of friction, the brake power of an engine is always less than the indicated power; hence the engine mechanical efficiency must be less than 1. Clearly, mechanical efficiencies as close to 1 as possible are desired.

The engine indicated power can also be expressed in terms of torque, through Equation (6.11). Thus an *indicated torque*, IT, can be defined. Similarly, a *brake mean effective pressure*, BMEP, may be defined that, when multiplied by the engine displacement and speed, yields the brake power, analogous to Equation (6.10). Table 6.1 summarizes these and other performance parameters and relations.

The thermal efficiency, as for other engines, is a measure of the fuel economy of a reciprocating engine. It tells the amount of power output that can be achieved for a given rate of heat release from the fuel. The rate of energy release is, in turn, the product of the rate of fuel flow and the fuel heating value. Thus, for a given thermal efficiency, power output can be increased by employing a high fuel flow rate and/or selecting a fuel with a high heat of combustion.

If the thermal efficiency is evaluated using the brake power, it is called the *brake thermal efficiency*, BTE. If the evaluation uses the indicated power, it is called the *indicated thermal efficiency*, ITE.

It is common practice in the reciprocating engine field to report engine fuel economy in terms of a parameter called the *specific fuel consumption*, SFC, analogous to the thrust specific fuel consumption used to describe jet engine performance. The specific fuel consumption is defined as the ratio of the fuel-mass flow rate to the power output. Typical units are pounds per horsepower-hour or kilograms per kilowatt-hour. Obviously, good fuel economy is indicated by low values of SFC. The SFC is called *brake specific fuel consumption*, BSFC, if it is defined using brake power or *indicated specific fuel consumption*, ISFC, when based on indicated power. The SFC for a reciprocating engine is analogous to the heat rate for a steam power plant in that both are measures of the rate of energy supplied per unit of power output, and in that low values of both are desirable.

Volumetric Efficiency

The theoretical energy released during the combustion process is the product of the mass of fuel contained in the combustion chamber and its heating value if the fuel is completely reacted. The more air that can be packed into the combustion chamber, the

Table 6.1 Engine Performance Parameters

	Indicated	Brake	Friction
Mean effective pressure	IMEP	BMEP	FMEP = IMEP – BMEP η_m = BMEP / IMEP
Power	IP	BP	FP = IP – BP η_m = BHP / IHP
Torque	IT	BT	FT = IT – BT η_m = BT / IT
Thermal efficiency	ITE	BTE	η_m = BTE / ITE
Specific fuel consumption	ISFC	BSFC	η_m = ISFC / BSFC

more fuel that can be burned with it. Thus a measure of the efficiency of the induction system is of great importance. The *volumetric efficiency*, η_v, is the ratio of the actual mass of mixture in the combustion chamber to the mass of mixture that the displacement volume could hold if the mixture were at ambient (free-air) density. Thus the average mass-flow rate of air through a cylinder is η_v (disp) $\rho_a N$. Pressure losses across intake and exhaust valves, combustion-chamber clearance volume, the influence of hot cylinder walls on mixture density, valve timing, and gas inertia effects all influence the volumetric efficiency.

EXAMPLE 6.3

A six-cylinder, four-stroke-cycle SI engine operates at 3000 rpm with an indicated mean effective pressure of five atmospheres using octane fuel with an equivalence ratio of 0.9. The brake torque at this condition is 250 lb_f–ft., and the volumetric efficiency is 85%. Each cylinder has a five inch bore and 6 inch stroke. Ambient conditions are 14.7 psia and 40°F. What is the indicated horsepower, brake horsepower, and friction horsepower; the mechanical efficiency; the fuel flow rate; and the BSFC?

Solution

The six cylinders have a total displacement of

$$DISP = 6\pi \times 5^2 \times 6/4 = 706.86 \text{ in}^3$$

Then the indicated horsepower is

$$IP = MEP \times DISP \times N/[12 \times 33{,}000] \quad [lb_f/in^2][in^3][cycles/min]/[in/ft][ft\text{-}lb_f/HP\text{-}min]$$

$$= (5)(14.7)(706.86)(3000/2)/[12 \times 33{,}000] = 196.8 \text{ horsepower}$$

The brake horsepower, from Equation (6.11), is:

$$BP = 2\pi \times 3000 \times 250 / 33{,}000 = 142.8 \text{ horsepower}$$

Then the friction power is the difference between the indicated and brake power:

$$FP = 196.8 - 142.8 = 54 \text{ horsepower}$$

and the mechanical efficiency is

$$\eta_m = 142.8/196.8 = 0.726$$

The ambient density is

$$\rho_a = 14.7 \times 144/ [53.3 \times 500] = 0.0794 \text{ lb}_m/\text{ft}^3$$

and the mass flow rate of air to the engine is

$$m_a = 0.85 \times 0.0794 \times 706.86 \times (3000/2)/1728 = 41.4 \text{ lb}_m/\text{min}$$

For octane the stoichiometric reaction equation is

$$C_8H_{18} + 12.5O_2 + (12.5 \times 3.76)N_2 \rightarrow 8CO_2 + 9H_2O + (12.5 \times 3.76)N_2$$

The fuel-air ratio is then

$$F/A = 0.9 \times [(8 \times 12) + (18 \times 1)]/[12.5(32 + 3.76 \times 28)] = 0.0598 \text{ lb}_m\text{-fuel}/\text{lb}_m\text{-air}$$

The fuel flow rate is

$m_f = m_a\,(F/A) = 41.4 \times 0.0598 = 2.474\ lb_m\ /min$

The brake specific fuel consumption is

$BSFC = 60\ m_f/BHP = 60{\times}2.474/142.8 = 1.04\ lb_m\ /BHP\text{-}hr$

6.7 Spark-Ignition Engine Performance

A typical indicator diagram showing intake and exhaust processes, valve actuation, and spark timing for a four-stroke-cycle SI engine is shown in Figure 6.6. It is assumed that an appropriate air-fuel mixture is supplied from a carburetor through an intake manifold to an intake valve, IV, and that the combustion gas is discharged through an exhaust valve, EV, into an exhaust manifold.

The induction of the air-fuel mixture starts with the opening of the intake valve at point A just before TC. As the piston sweeps to the right, the mixture is drawn into the cylinder through the IV. The pressure in the cylinder is somewhat below that in the intake manifold due to the pressure losses across the intake valve. In order to use the momentum of the mixture inflow through the valve at the end of the intake stroke to improve the volumetric efficiency, intake valve closure is delayed to shortly after BC at point B. Power supplied from inertia of a flywheel (and the other rotating masses in the engine) drives the piston to the left, compressing and raising the temperature of the trapped mixture.

The combustion process in a properly operating SI engine is progressive in that the reaction starts at the spark plug and progresses into the unburned mixture at a finite speed. Thus the combustion process takes time and cannot be executed instantaneously as implied by the theoretical cycle. In order for the process to take place as near to TC as possible, the spark plug is fired at point S. The number of degrees of crank rotation before TC at which the spark occurs is called the *ignition advance*. Advances of 10° to 30° are common, depending on speed and load. The spark advance may be controlled by devices that sense engine speed and intake manifold pressure. Microprocessors are now used to control spark advance and other functions, based on almost instantaneous engine performance measurements.

Recalling the slider-crank analysis, we observ that the piston velocity at top center is momentarily zero as the piston changes direction. Therefore no work can be done at this point, regardless of the magnitude of the pressure force. Thus, to maximize the work output, it is desired to have the maximum cylinder pressure occur at about 20° after TC. Adjustment of the spark advance (in degrees before TC) allows some control of the combustion process and the timing of peak pressure. For a fixed combustion duration, the combustion crank-angle interval must increase with engine speed. As a consequence, the ignition advance must increase with increasing engine speed to

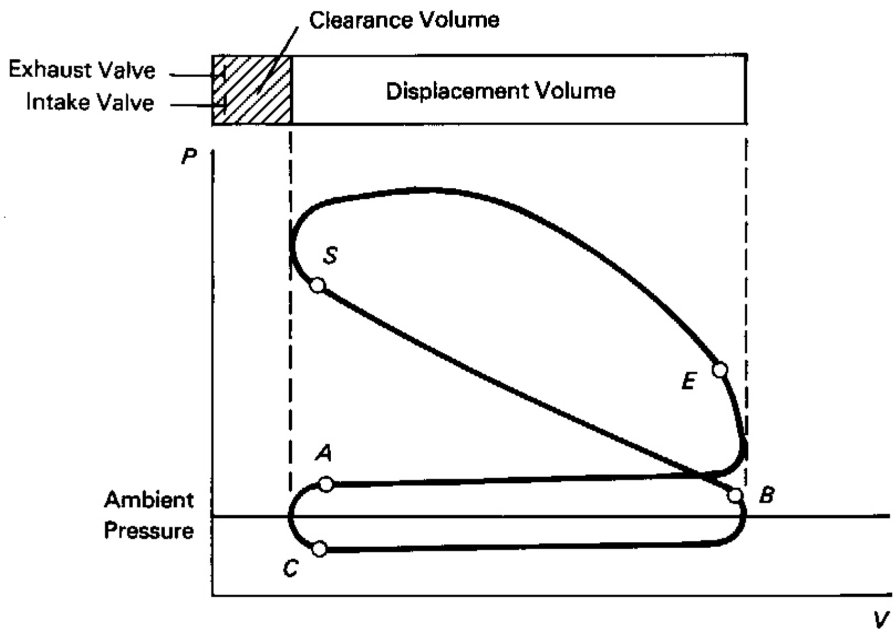

FIGURE 6.6 Indicator diagram for a four-stroke-cycle spark-ignition engine.

maintain optimum timing of the peak pressure.

Following combustion, the piston continues toward bottom center as the high pressure gases expand and do work on the piston during the power stroke. As the piston approaches BC, the gases do little work on the piston as its velocity again approaches zero. As a result, not much work is lost by early opening of the exhaust valve before BC (at point E) to start the blowdown portion of the exhaust process. It is expedient to sacrifice a little work during the end of the power stroke in order to reduce the work needed to overcome an otherwise-high exhaust stroke cylinder pressure. Inertia of the gas in the cylinder and resistance to flow through the exhaust valve opening slow the drop of gas pressure in the cylinder after the valve opens.

Thus the gases at point E are at a pressure above the exhaust manifold pressure and, during blowdown, rush out through the EV at high speed. Following blowdown, gases remaining in the cylinder are then expelled as the piston returns to TC. They remain above exhaust manifold pressure until reaching TC because of the flow resistance of the exhaust valve. The EV closes shortly after TC at point C, terminating the exhaust process. The period of overlap at TC between the intake valve opening at point A and exhaust valve closing at point C in Figure 6.6 allows more time for the intake and exhaust processes at high engine speeds, when about 10 milliseconds may be available for these processes. At low engine speed and at idling there may be some mixture loss through the exhaust valve and discharge into the intake manifold during this valve overlap period.

The combined exhaust and induction processes are seen to form a “pumping loop” that traverses the p-V diagram in a counterclockwise direction and therefore

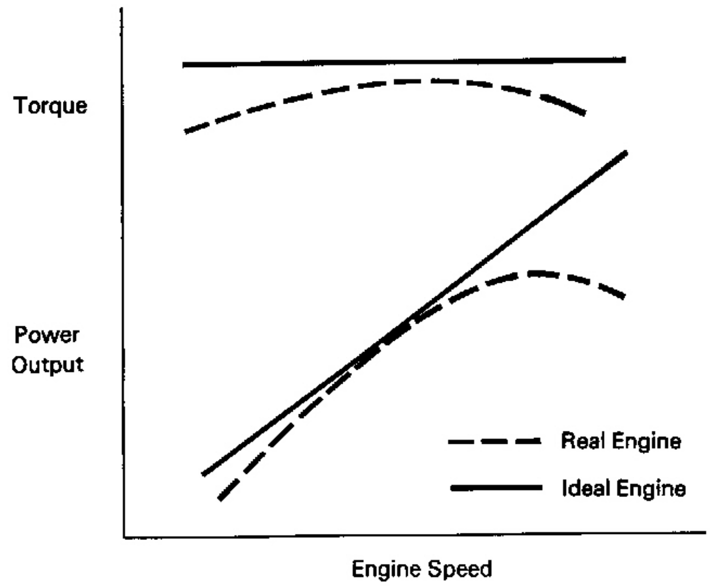

FIGURE 6.7 Torque and power characteristics of a reciprocating engine.

represents work input rather than work production. The higher the exhaust stroke pressure and the lower the intake stroke pressure, the greater the area of the pumping loop and hence the greater the work that must be supplied by the power loop (clockwise) to compensate. Great attention is therefore paid to valve design and other engine characteristics that influence the exhaust and induction processes. Volumetric efficiency is a major parameter that indicates the degree of success of these efforts.

Performance Characteristics

A given ideal Otto-cycle engine produces a certain amount of work per cycle. For such a cycle, MEP = W/disp is a constant. Equating the power equations (6.9) and (6.11) shows that the average torque is proportional to MEP and independent of engine engine speed. Therefore power output for the ideal engine is directly proportional to the number of cycles executed per unit time, or to engine speed. Thus an Otto engine has ideal torque and power characteristics, as shown by the solid lines in Figure 6.7.

The characteristics of real engines (represented by the dashed lines) tend to be similar in nature to the ideal characteristics but suffer from speed-sensitive effects, particularly at low or high speeds. Torque and power characteristics for a 3.1 liter V6 engine (ref. 9) are shown by the solid lines in Figure 6.8. Note the flatness of the torque-speed curve and the expected peaking of the power curve at higher speed than the torque curve. Rather than present graphical characteristics such as this in their

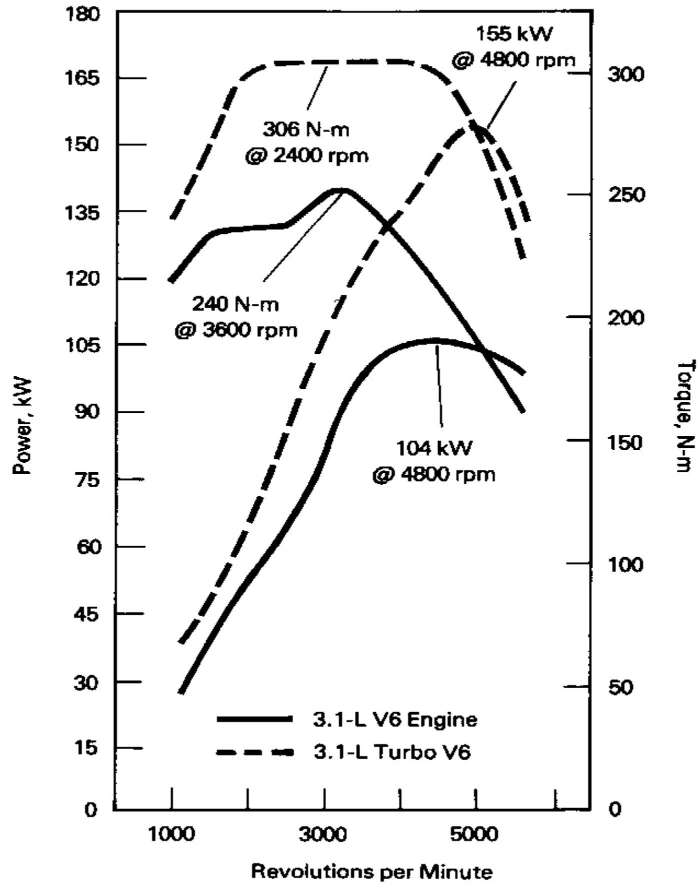

FIGURE 6.8 Comparison of normally aspirated and turbocharged 3.1-liter engines. (Reprinted with permission from *Automotive Engineering* magazine, October 1988. Society of Automotive Engineers.)

brochures, automobile manufacturers usually present only values for the maximum power and torque and the speeds at which they occur. Engine characteristics such as those shown in the figure are invaluable to application engineers seeking a suitable engine for use in a product.

6.8 The Compression-Ignition or Diesel Cycle

The ideal Diesel cycle differs from the Otto cycle in that combustion is at constant pressure rather than constant volume. The ideal cycle, shown in Figure 6.9, is commonly implemented in a reciprocating engine in which air is compressed without fuel from state 1 to state 2. With a typically high compression ratio, state 2 is at a temperature high enough that fuel will ignite spontaneously when sprayed directly into the air in the combustion chamber from a high-pressure fuel injection system.

By controlling the fuel injection rate and thus the rate of chemical energy release in relation to the rate of expansion of the combustion gases after state 2, a constant-

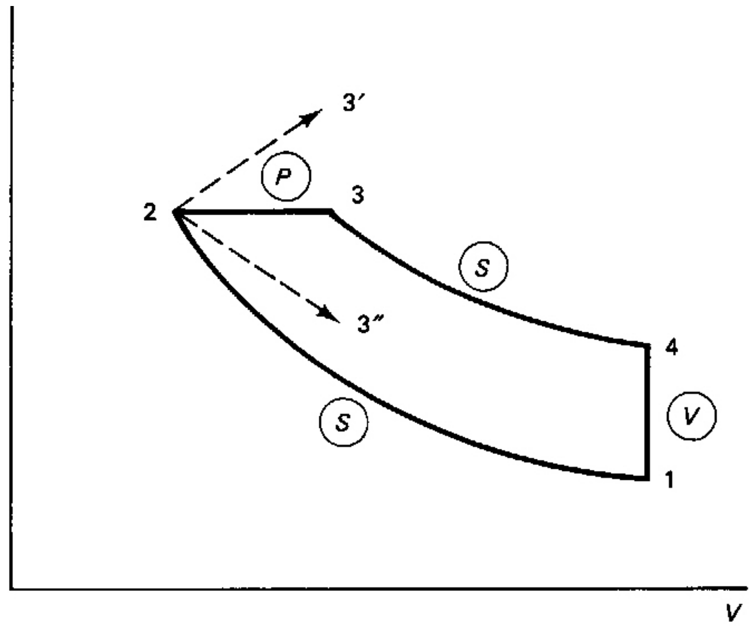

FIGURE 6.9 The diesel cycle.

pressure process or other energy release pattern may be achieved as in Figure 6.9. For example, if the energy release rate is high, then pressure may rise, as from 2 to 3', and if low may fall to 3". Thus constant-pressure combustion made possible by controlling the rate of fuel injection into the cyclinder implies the use of a precision fuel injection system.

Instead of injecting fuel into the high-temperature compressed air, the cycle might be executed by compression of an air-fuel mixture, with ignition occurring either spontaneously or at a hot spot in the cylinder near the end of the compression process. Inconsistency and unpredictability of the start of combustion in this approach, due to variations in fuel and operating conditions, and to lack of control of the rate of heat release with the possibility of severe knock, makes the operation of such an engine unreliable, at the least, and also limits the maximum compression ratio. The Diesel engine therefore usually employs fuel injection into compressed air rather than carbureted mixture formation.

In the Air Standard cycle analysis of the Diesel cycle, the heat addition process is at constant pressure:

$$q_{2\to3} = c_p(T_3 - T_2) \qquad [\text{Btu/lb}_m \mid \text{kJ/kg}] \qquad (6.13)$$

and, as with the Otto cycle, the closing process is at constant volume:

$$q_{4\to1} = c_v(T_1 - T_4) \qquad [\text{Btu/lb}_m \mid \text{kJ/kg}] \qquad (6.14)$$

The net work and thermal efficiency are then:

$$w = q_{2\to3} + q_{4\to1}$$

$$= c_p(T_3 - T_2) + c_v(T_1 - T_4)$$

$$= c_vT_1[k(T_3/T_1 - T_2/T_1) + 1 - T_4/T_1] \qquad [\text{Btu/lb}_m \mid \text{kJ/kg}] \qquad (6.15)$$

$$\eta_{\text{Diesel}} = w/q_{2\to3} = 1 + q_{4\to1}/q_{2\to3} = 1 + (c_v/c_p)(T_1 - T_4)/(T_3 - T_2)$$

$$= 1 - (1/k)(T_1/T_2)(T_4/T_1 - 1)/(T_3/T_2 - 1) \qquad [\text{dl}] \qquad (6.16)$$

The expressions for the net work and cycle efficiency may be expressed in terms two parameters, the compression ratio, CR = V_1/V_2 (as defined earlier in treating the Otto cycle) and the *cutoff ratio*, COR = V_3/V_2. The temperature ratios in Equations (6.15) and (6.16) may be replaced by these parameters using, for the constant-pressure process,

$$\text{COR} = V_3/V_2 = T_3/T_2$$

and by expanding the following identity:

$$T_4/T_1 = (T_4/T_3)(T_3/T_2)(T_2/T_1)$$

$$= (V_3/V_4)^{k-1}(V_3/V_2)(V_1/V_2)^{k-1}$$

$$= [(V_3/V_4)(V_1/V_2)]^{k-1}\text{COR} = (\text{COR})^{k-1}\text{COR}$$

$$= \text{COR}^k$$

where the product of the volume ratios was simplified by recognizing that $V_4 = V_1$. Thus the nondimensionalized net work and Diesel-cycle thermal efficiency are given by

$$w/c_vT_1 = k\text{CR}^{k-1}(\text{COR} - 1) + (1 - \text{COR}^k) \qquad [\text{dl}] \qquad (6.17)$$

and

$$\eta_{\text{Diesel}} = 1 - (1/k)[(\text{COR}^k - 1)/(\text{COR} - 1)]/\text{CR}^{k-1} \qquad [\text{dl}] \qquad (6.18)$$

where the cutoff ratio, COR, is the ratio of the volume at the end of combustion, V_3, to that at the start of combustion, V_2. Thus the cutoff ratio may be thought of as a measure of the duration of fuel injection, with higher cutoff ratios corresponding to longer combustion durations.

Diesel-cycle net work increases with both compression ratio and cutoff ratio. This is readily seen graphically from Figure 6.9 in terms of p-V diagram area. As with the Otto cycle, increasing compression ratio increases the Diesel-cycle thermal efficiency. Increasing cutoff ratio, however, decreases thermal efficiency. This may be rationalized by observing from the p-V diagram that much of the additional heat supplied when injection is continued is rejected at increasingly higher temperatures. Another view is that heat added late in the expansion process can produce work only over the remaining part of the stroke and thus adds less to net work than to heat rejection.

EXAMPLE 6.4

A Diesel engine has a compression ratio of 20 and a peak temperature of 3000K. Using an Air Standard cycle analysis, estimate the work per unit mass of air, the thermal efficiency, the combustion pressure, and the cutoff ratio.

Solution

Assuming an ambient temperature and pressure of 300K and 1 atmosphere, the temperature at the end of the compression stroke is

$$T_2 = (300)(20)^{1.4-1} = 994.3\text{K}$$

and the combustion pressure is

$$p_2 = (1)(20)^{1.4} = 66.3 \text{ atm}$$

Then the cutoff ratio is

$$V_3/V_2 = T_3/T_2 = 3000/994.3 = 3.02$$

The expansion ratio is calculated as follows:

$$V_4/V_3 = (V_1/V_2)/(V_3/V_2) = 20/3.02 = 6.62$$

$$T_4 = T_3(V_3/V_4)^{1.4-1} = 3000/6.62^{0.4} = 1409\text{K}$$

$$w = 1.005(3000 - 994.3) + (1.005/1.4)(300 - 1409) = 1219.6 \text{ kJ/kg}$$

$$q_a = 1.005(3000 - 994.3) = 2015.7 \text{ kJ/kg}$$

$$\eta_{th} = w/q_a = 1219.6/2015.6 = 0.605, \quad \text{or } 60.5\%$$

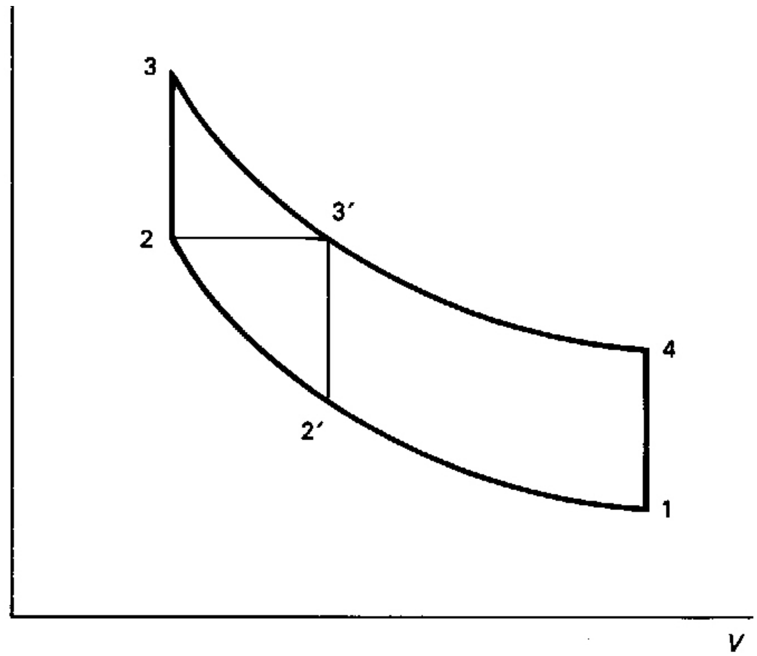

FIGURE 6.10 Comparison of Otto-cycle and diesel-cycle efficiencies.

6.9 Comparing Otto-Cycle and Diesel-Cycle Efficiencies

A reasonable question at this point is: Which cycle is more efficient, the Otto cycle or the Diesel cycle? Figure 6.10 assists in examining this question. In general notation, the cycle efficiency may be written as

$$\eta_{th} = w_{net}/q_{in} = w_{net}/(w_{net} + |q_{out}|)$$

$$= 1/(1 + |q_{out}|/w_{net}) \qquad [dl] \qquad (6.19)$$

Comparing the Otto cycle 1–2–3–4 and the Diesel cycle with the same compression ratio 1–2–3'–4, we see that both have the same heat rejection but that the Otto cycle has the higher net work. Equation (6.19) then shows that, for the same compression ratio, the Otto cycle has the higher efficiency.

It has been observed that Diesel-cycle efficiency decreases with increasing cutoff ratio for a given compression ratio. Let us examine the limit of the Diesel-cycle efficiency for constant CR as COR approaches its minimum value, 1. We may write Equation (6.18) as

$$\eta_{Diesel} = 1 - 1/(k\mathrm{CR}^{k-1})\, f(\mathrm{COR})$$

where $f(\mathrm{COR}) = (\mathrm{COR}^k - 1)/(\mathrm{COR} - 1)$. Applying L'Hospital's rule, with primes designating differentiation with respect to COR, to the limit of f(COR) as COR $\rightarrow 1$, yields

$$\lim_{COR \to 1} f(COR) = \lim_{COR \to 1} (COR^k - 1)' / \text{Lim } (COR - 1)' = \lim_{COR \to 1} kCOR^{k-1} = k$$

and

$$\lim_{COR \to 1} \eta_{Diesel} = 1 - 1/CR^{k-1} = \eta_{Otto}$$

Thus the limit of the Diesel-cycle efficiency as COR approaches 1 is the Otto cycle efficiency. Hence Equation (6.18) shows that the efficiency of the Diesel cycle must be less than or equal to the Otto-cycle efficiency if both engines have the same compression ratio, the same conclusion we reached by examination of the p-V diagram.

Suppose, however, that the compression ratios are not the same. Compare the Otto cycle 1–2'–3'–4 with the Diesel cycle 1–2–3'–4 having the same maximum temperature in Figure 6.10. The Otto cycle has a smaller area, and therefore less work, than the Diesel cycle, but the same heat rejection. Equation (6.19) demonstrates that the Otto cycle has a lower thermal efficiency than the Diesel cycle with the same maximum temperature.

The conclusion that must be drawn from the above comparisons is quite clear. As in most comparative engineering studies, the result depends on the ground rules which were adopted at the start of the study. The Otto cycle is more efficient if the compression ratio is the same or greater than that of the competing Diesel cycle. But knock in spark-ignition (Otto) engines limits their compression ratios to about 12, while Diesel-engine compression ratios may exceed 20. Thus, with these higher compression ratios, the Air Standard Diesel-cycle efficiency can exceed that of the Otto cycle. In practice, Diesel engines tend to have higher efficiencies than SI engines because of higher compression ratios.

6.10 Diesel-Engine Performance

In 1897, five years after Rudolph Diesel's first patents and twenty-one years after Otto's introduction of the spark-ignition engine, Diesel's compression-ignition engine was proven to develop 13.1 kilowatts of power with an unprecedented brake thermal efficiency of 26.2% (ref. 7). At that time, most steam engines operated at thermal efficiencies below 10 %; and the best gas engines did not perform much better than the steam machines.

Diesel claimed (and was widely believed) to have developed his engine from the principles expounded by Carnot. He had developed "the rational engine." Whether his claims were exaggerated or not, Diesel's acclaim was well deserved. He had developed an engine that operated at unprecedented temperatures and pressures, had proven his concept of ignition of fuel by injection into the compressed high-temperature air, and had overcome the formidable problems of injecting a variety of fuels in appropriate amounts with the precise timing required for satisfactory combustion. His is a fascinating story of a brilliant and dedicated engineer (refs. 7, 8).

In the Diesel engine, the high air temperatures and pressures prior to combustion are attributable to the compression of air alone rather than an air-fuel mixture. Compression of air alone eliminates the possibility of autiognition during compression and makes high compression ratios possible. However, because of the high pressures and temperatures, Diesel engines must be designed to be structurally more rugged. Therefore, they tend to be heavier than SI engines with the same brake power.

The energy release process in the Diesel engine is controlled by the rate of injection of fuel. After a brief ignition lag, the first fuel injected into the combustion chamber autoignites and the resulting high gas temperature sustains the combustion of the remainder of the fuel stream as it enters the combustion chamber. Thus it is evident that the favorable fuel characteristic of high autoignition temperature for an SI engine is an unfavorable characteristic for a Diesel engine.

In the Diesel engine, a low autoignition temperature and a short ignition delay are desirable. Knock is possible in the Diesel engine, but it is due to an entirely different cause than knock in a spark-ignition engine. If fuel is ignited and burns as rapidly as it is injected, then smooth, knock-free combustion occurs. If, on the other hand, fuel accumulates in the cylinder before ignition due to a long ignition lag, an explosion or detonation occurs, producing a loud Diesel knock. The *cetane number* is the parameter that identifies the ignition lag characteristic of a fuel.

The cetane number, like the octane number, is determined by testing in a CFR engine. The ignition lag of the test fuel is compared with that of a mixture of *n*-cetane, $C_{16}H_{34}$, and heptamethylnonane, HMN (ref. 10). Cetane, which has good ignition qualities, is assigned a value of 100; and HMN, which has poor knock behavior, a value of 15. The cetane number is then given by the sum of the percentage of *n*-cetane and 0.15 times the percentage of HMN in the knock-comparison mixture. A cetane number of 40 is the minimum allowed for a Diesel fuel.

6.11 Superchargers and Turbochargers

The importance of the volumetric efficiency, representing the efficiency of induction of the air-fuel mixture into the reciprocating-engine cylinders, was discussed earlier. Clearly, the more mixture mass in the displacement volume, the more chemical energy can be released and the more power will be delivered from that volume.

During the Second World War, the mechanical supercharger was sometimes used with SI aircraft engines to increase the power and operational ceiling of American airplanes. Today supercharging is used with both Diesel engines and SI engines. The *supercharger* is a compressor that supplies air to the cylinder at high pressure so that the gas density in the cylinder at the start of compression is well above the free-air density. The piston exhaust gases are allowed to expand freely to the atmosphere through the exhaust manifold and tailpipe. The supercharger is usually driven by a belt or gear train from the engine crank shaft.

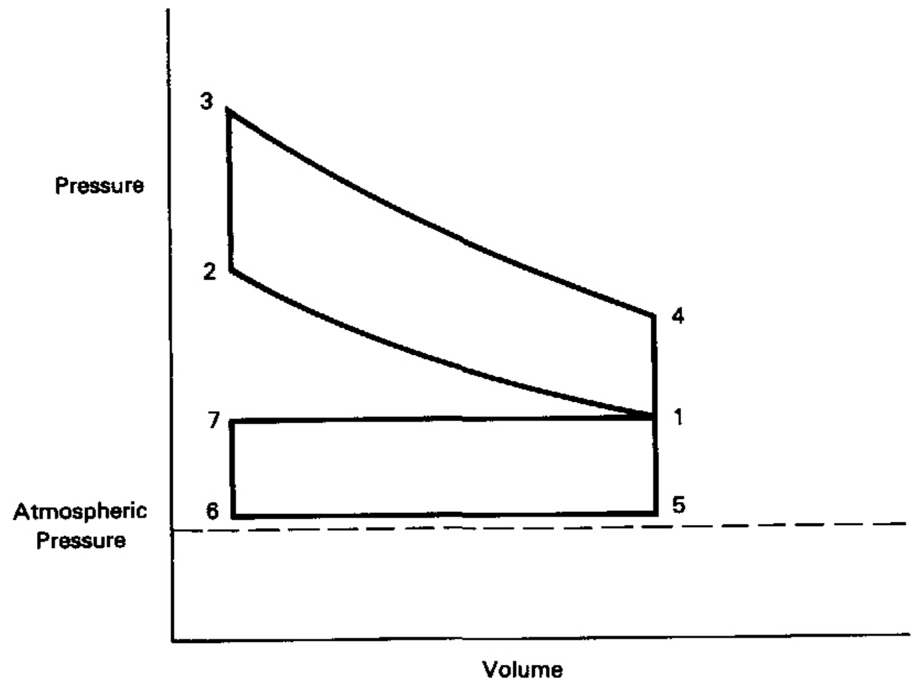

FIGURE 6.11 Supercharged Otto cycle.

Figure 6.11 shows a modification of the theoretical Otto cycle to accommodate mechanical supercharging. The supercharger supplies air to the engine cyclinders at pressure p_7 in the intake process $7 \rightarrow 1$. The processes $4 \rightarrow 5 \rightarrow 6$ purge most of the combustion gas from the cylinder. The most striking change in the cycle is that the induction-exhaust loop is now traversed counterclockwise, indicating that the cylinder is delivering net work during these processes as well as during the compression-expansion loop. It should be remembered, however, that part of the cycle indicated power must be used to drive the external supercharger.

The turbosupercharger or *turbocharger,* for short, is a supercharger driven by a turbine using the exhaust gas of the reciprocating engine, as shown schematically in Figure 6.12. A cutaway view of a turbocharger is shown in Figure 6.13(a). Figure 6.13(b) presents a diagram for the turbocharger. Compact turbochargers commonly increase the brake power of an engine by 30% or more, as shown in Figure 6.8, where the performance of an engine with and without turbocharging is compared. There, a substantial increase in peak torque and flattening of the torque-speed curve due to turbocharging is evident.

For a supercharged engine, the brake power, BP, is the indicated power (as in Figure 6.11) less the engine friction power and the supercharger shaft power:

$$\mathrm{BP} = \mathrm{DISP} \cdot \mathrm{IMEP} \cdot N - P_m - \mathrm{FP} \qquad [\text{ft-lb}_f/\text{min} \mid \text{kJ/s}] \qquad (6.15)$$

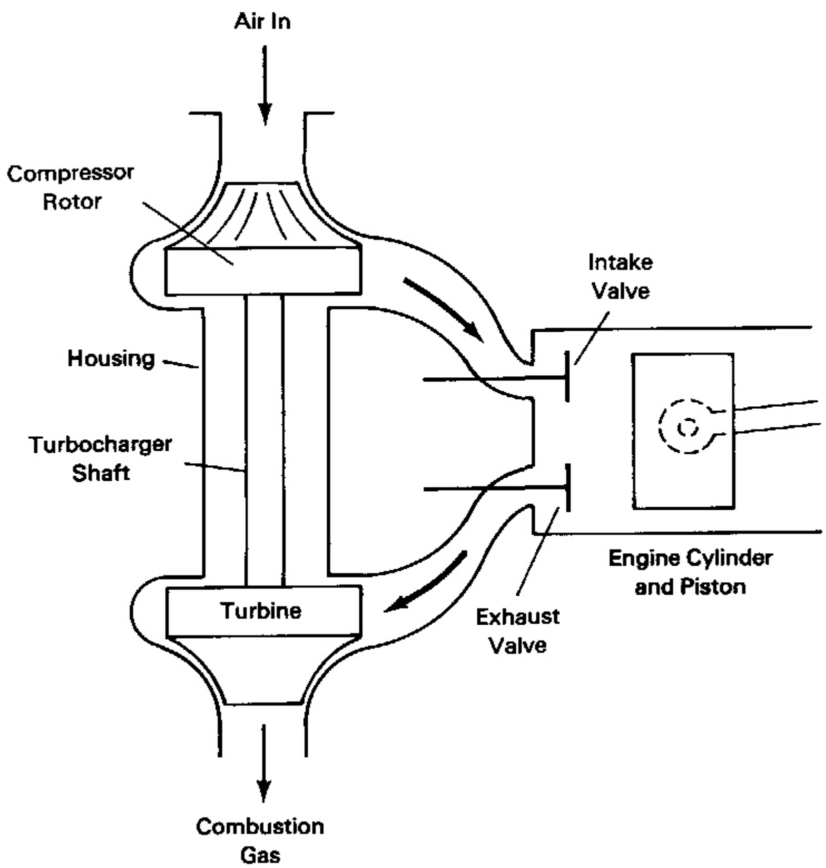

FIGURE 6.12 Schematic of a turbocharged engine cylinder.

where P_m is the supercharger-shaft mechanical power supplied by the engine (0 for a turbocharger). The IMEP includes the positive work contribution of the exhaust loop. The exhaust back pressure of the reciprocating engine is higher with a turbocharger than for a naturally aspirated or mechanically supercharged engine because of the drop in exhaust gas pressure through the turbine. The engine brake power increases primarily because of a higher IMEP due to the added mass of fuel and air in the cylinder during combustion. Intercooling between the compressor and the intake manifold may be used to further increase the cylinder charge density. Turbocharging may increase engine efficiency, but its primary benefit is a substantial increase in brake power.

In a turbocharged engine, a wastegate may be required to bypass engine exhaust gas around the turbine at high engine speeds. This becomes necessary when the compressor raises the intake manifold pressure to excessively high levels, causing engine knock or threatening component damage. Thirty to forty percent of the exhaust flow may be bypassed around the turbine at maximum speed and load (ref. 1).

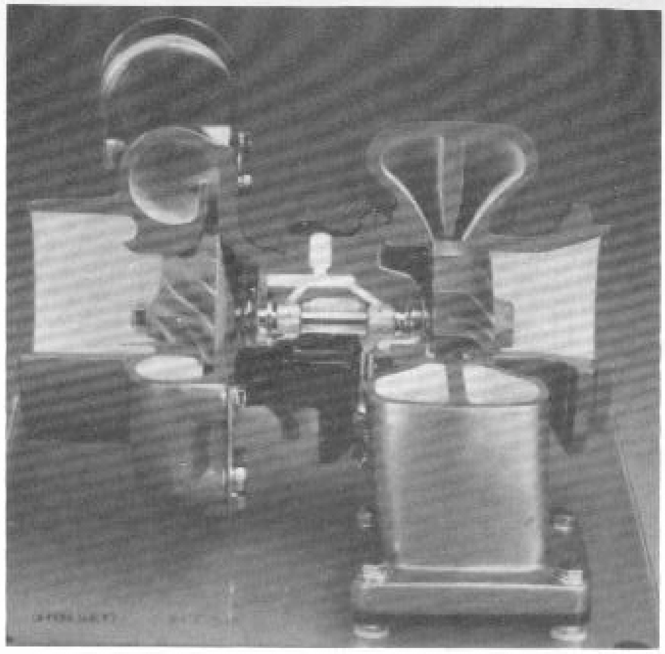

FIGURE 6.13a Cutaway of a diesel-engine turbocharger. The centrifugal compressor is on the left, and the turbine is on the right. (Photo courtesy of Holset Engineering Co.

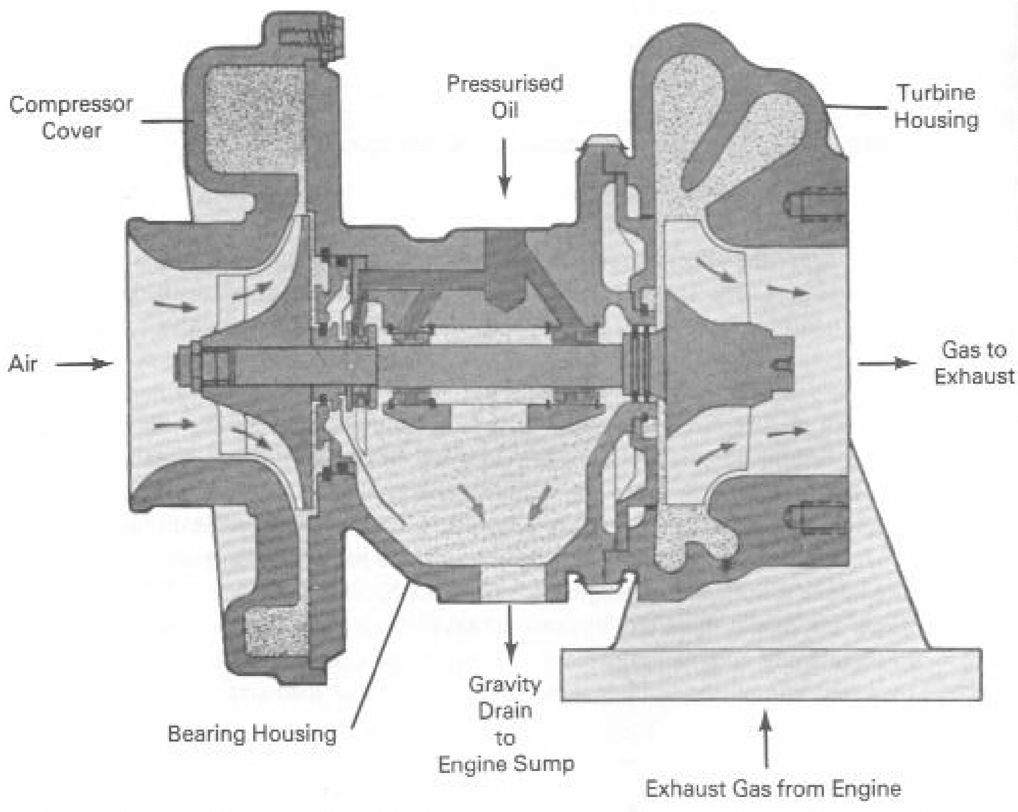

FIGURE 6.13b Cross-section of a diesel-engine turbocharger. (Courtesy of Holset Engineering Co.), Inc.)

6.12 The Automobile Engine and Air Pollution

Since the Second World War, concern for environmental pollution has grown from acceptance of the status quo to recognition and militance of national and international scope. Among other sources, causes of the well-known Los Angeles smog problem were identified as hydrocarbons (HC) and oxides of nitrogen (NO_x) in exhaust emissions from motor vehicle reciprocating engines. As a result, national and California automobile air pollution limits for automobiles have been established and toughened. Prior to the Clean Air Act of 1990, the U.S. federal exhaust-gas emissions standards limited unburned hydrocarbons, carbon monoxide, and oxides of nitrogen to 0.41, 3.4, and 1.0 g/mile, respectively. According to reference 12, today it takes 25 autos to emit as much CO and unburned hydrocarbons and 4 to emit as much NO_x as a single car in 1960. The reference anticipated that, led by existing California law and other factors, future engine designs should be targeted toward satisfying a tailpipe standard of 0.25, 3.4, 0.4 g/mile. Indeed, the 1990 Clean Air Act (refs. 15,16) specified these limits for the first 50,000 miles or five years of operation for all passenger cars manufactured after 1995. In addition to the regulations on gaseous emissions, the Clean Air Act of 1990 adopted the California standard for particulate matter of 0.08 g/mile for passenger cars. The standards on particulates are particularly difficult for the Diesel engine, because of its of soot-producing tendency.

The automobile air pollution problem arises in part because the reactions in the exhaust system are not in chemical equilibrium as the gas temperature drops. Oxides of nitrogen, once formed in the cylinder at high temperature, do not return to equilibrium concentrations of nitrogen and oxygen in the cooling exhaust products. Likewise, CO formed with rich mixtures or by dissociation of CO_2 in the cylinder at high temperature does not respond rapidly to an infusion of air as its temperature drops in the exhaust system. Their concentrations may be thought of as constant or *frozen*. Unburned hydrocarbons are produced not only by rich combustion but also by unburned mixture lurking in crevices (such as between piston and cylinder above the top piston ring), by lubricating oil on cylinder walls and the cylinder head that absorbs and desorbs hydrocarbons before and after combustion, and by transient operating conditions.

Starting in 1963, positive crankcase ventilation was used in all new cars to duct fuel-rich crankcase gas previously vented to the atmosphere back into the engine intake system. Later in the ‘60s, various fixes were adopted to comply with regulation of tailpipe unburned hydrocarbons and CO, including lowering compression ratios.

In 1973, NO_x became federally regulated, and exhaust gas recirculation (EGR) was employed to reduce NO_x formation through reduced combustion temperatures. At the same time, HC and CO standards were reduced further, leading to the use of the oxidizing catalytic converter. Introduction of air pumped into the tailpipe provided additional oxygen to assist in completion of the oxidation reactions. In 1981, a reducing catalytic converter came into use to reduce NO_x further. This device does not perform well in an oxidizing atmosphere. As a result, two-stage catalytic converters were applied, with the first stage reducing NO_x in a near-stoichiometric mixture and the second oxidizing the combustibles remaining in the exhaust with the help of air introduced between the stages. This fresh air does not the increase NO_x significantly, because of the relatively low temperature of the exhaust. The three-way catalytic converter using several exotic metal catalysts to reduce all three of the gaseous pollutants was also introduced.

The use of catalytic converters to deal with all three pollutants brought about significant simultaneous reductions in the three major gaseous pollutants from automobiles. This allowed fuel-economy-reducing modifications that had been introduced earlier to satisfy emission reduction demands to be eliminated or relaxed, leading to further improvements in fuel economy.

Catalytic converters, however, require precise control of exhaust gas oxygen to near-stoichiometric mixtures. The on-board computer has made possible control of mixture ratio and spark timing in response to censor outputs of intake manifold pressure, exhaust gas oxygen, engine speed, air flow, and incipient knock. The oxygen, or lambda, censor located in the exhaust pipe upstream of the three-way converter or between the two-stage converters is very sensitive to transition from rich to lean exhaust and allows close computer control of the mixture ratio to ensure proper operation of the catalytic converter. Computer control of carburetors or fuel injection as well as other engine functions has allowed simultaneous improvement in fuel economy and emissions in recent years. Thus, while emissions have been drastically reduced since 1974, according to reference 11 the EPA composite fuel economy of the average U.S. passenger car has nearly doubled; although this improvement has not come from the engine alone. Despite the hard-won gains in emissions control and fuel economy, further progress may be expected.

EXAMPLE 6.5

The 1990 NO_x emissions standard is 0.4 grams per mile. For an automobile burning stoichiometric octane with a fuel mileage of 30 mpg, what is the maximum tailpipe concentration of NO_x in parts per million? Assume that NO_x is represented by NO_2 and that the fuel density is 692 kilograms per cubic meter.

Solution

For the stoichiometric combustion of octane, C_8H_{18}, the air-fuel ratio is 15.05 and the molecular weight of combustion products is 28.6. The consumption of octane is

$$m_f = (692)(1000)(3.79\times10^{-3})/\ 30 = 87.4 \text{ g/mile}$$

[Note: $(kg/m^3)(g/kg)(m^3/gal)/(mile/gal) = g/mile$.] The concentration of NO_x is the ratio of the number of moles of NO_x to moles of combustion gas products:

$$\text{mole}_{\text{Nox}}/\text{mole cg} = (m_{NOx}/m_f)(m_f/m_{cg})(M_{cg}/M_{NOx})$$

$$= (0.4/87.4)(28.6/46)/\ (15.05 + 1) = 0.0001773$$

or 177.3 parts per million (ppm).

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EXERCISES

6.1 Plot dimensionless piston position against crank angle for $S/2L$ = 0.5, 0.4, 0.3, and 0.2.

6.2* Obtain expressions for the piston velocity and acceleration as a function of the crank angle, constant angular velocity, and $S/2L$ ratio. Use a spreadsheet to calculate and plot velocity and acceleration against crank angle for $S/2L$ = 0.5, 0.4, 0.3, and 0.2.

6.3 Determine the equation for the piston motion for a scotch yoke mechanism in terms of crank angle. Obtain an equation for the piston velocity for a crank that turns with a given angular velocity, ω.

6.4 Derive an equation for the Otto-engine net work by integration of pdV for the Air Standard cycle. Compare with Equation (6.6).

6.5* Use a spreadsheet to calculate and plot cycle efficiency as a function of compression ratio for the Diesel cycle for cutoff ratios of 1, 2, and 3. Indentify the Otto-cycle efficiency on the plot. Explain and show graphically from the plot how a Diesel engine can be more efficient than an Otto engine.

6.6 A single-cylinder Air Standard Otto engine has a compression ratio of 8.5 and a peak temperature of 3500°F at ambient conditions of 80°F and one atmosphere. Determine the cycle efficiency, maximum cylinder pressure, and mean effective pressure.

6.7 A six-cylinder engine with a compression ratio of 11 runs at 2800 rpm at 80°F and 14.7 psia. Each cylinder has a bore and stroke of three inches and a volumetric efficiency of 0.82. Assume an Air Standard, four-stroke Otto cycle

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* Exercise numbers with an asterisk indicate that they involve computer usage.

with stoichiometric octane as fuel. Assume that the energy release from the fuel is equally divided between internal energy increase in cylinder gases and cylinder wall heat loss. What are the cylinder mean effective pressures and the engine horsepower and specific fuel consumption?

6.8 A single-cylinder four-stroke-cycle spark-ignition engine has a BSFC of 0.4 kg/kW-hr and a volumetric efficiency of 78% at a speed of 45 rps. The bore is 6 cm and the stroke is 8.5 cm. What is the fuel flow rate, fuel-air ratio, and brake torque if the brake power output is 6 kW with ambient conditions of 100 kPa and 22°C?

6.9 A single-cylinder four-stroke-cycle spark-ignition engine operating at 3500 rpm has a brake mean effective pressure of 1800 kPa and a displacement of 400 cm^3. Atmospheric conditions are 101kPa and 27°C.

(a) If the stroke is 6 cm, what is the bore?
(b) What is the brake power?
(c) If the mass air-fuel ratio is 16 and the fuel flow rate is 0.00065 kg/s, what is the volumetric efficiency?
(d) Compare your results with the performance of a two-cylinder engine with the same overall geometric characteristics.

6.10 A four-cylinder four-stroke-cycle spark-ignition engine operating at 3500 rpm has a brake mean effective pressure of 80 psi and a displacement of 400 cm^3. Atmospheric conditions are one bar and 80°F.

(a) If the stroke is 3 inches, what is the bore?
(b) What is the brake power?
(c) If the mass air-fuel ratio is 16 and the fuel flow rate is 0.00065 kg/s, what is the volumetric efficiency?

6.11 A four-cylinder four-stroke-cycle spark-ignition engine with 200 cm^3 displacement and operating in air at 27°C and 110 kPa has a friction power of 27 kW and a brake power output of 136 kW at 3600 rpm.

(a) What is the mechanical efficiency?
(b) If it has a volumetric efficiency of 74% and burns liquid methanol with 15% excess air, what is the brake specific fuel consumption?

6.12 An eight-cylinder four-stroke-cycle engine has a bore of three inches and a stroke of 4 inches. At a shaft speed of 3000 rpm, the brake horsepower is 325 and the mechanical efficiency is 88%. Fuel with a heating value of 19,000 Btu/lb_m is supplied at a rate of 80 lb_m/hr. What are the engine displacement, BMEP, brake torque, and indicated specific fuel consumption in lb_m/HP-hr?

6.13 An eight-cylinder four-stroke-cycle engine has a bore of 10 cm and a stroke of 12 cm. At a shaft speed of 53 rps, the brake power is 300kW and the mechanical efficiency is 85%. Fuel with a heating value of 40,000 kJ/kg is supplied at a rate of 40 kg/hr. What are the engine displacement, BMEP, brake torque, and indicated specific fuel consumption in kg/kW-hr?

6.14 Consider a naturally aspirated eight-cylinder four-stroke-cycle Diesel engine with a compression ratio of 20 and cutoff ratio of 2.5. Air is inducted into the cylinder from the atmosphere at 14.5 psia and 80°F. Assume an Air Standard cycle.

(a) Determine the temperatures and pressures immediately before and after combustion.
(b) What is the heat added in the combustion process, in Btu/lb_m?
(c) What is the net work, in Btu/lb_m, and the thermal efficiency?
(d) If the volumetric efficiency is 85%, the engine displacement is 300 in^3, and the engine speed is 2000 rpm, what is the mass flow rate of air through the engine in lb_m/min?
(e) What is the engine horsepower?
(f) Assuming that losses through the valves cause a 20-psi pressure differential between the pressures during the exhaust and intake strokes, estimate the actual and fractional losses, in horsepower, due to these processes. Sketch the appropriate p-V diagram.

6.15 A two-cylinder four-stroke-cycle engine produces 30 brake horsepower at a brake thermal efficiency of 20% at 2600 rpm. The fuel is methane burning in air with an equivalence ratio of 0.8 and a heating value of 21,560 Btu/lb_m. Ambient conditions are 520°R and 14.7 psia. The engine mechanical efficiency is 88%, and the volumetric efficiency is 92%. What are the fuel flow rate, the displacement volume per cylinder, and the brake specific fuel consumption? What is the bore if the bore and stroke or equal?

6.16 Sketch carefully a single p–V diagram showing Otto and Diesel cycles having the same minimum and maximum temperatures. Shade the area representing the difference in net work between the cycles. Repeat for cycles having the same compression ratio. Discuss the implications of these sketches.

6.17 An eight-cylinder reciprocating engine has a 3-in. bore and a 4-in. stroke and runs at 1000 cycles per minute. If the brake horsepower is 120 and the mechanical efficiency is 80%, estimate the indicated mean effective pressure.

6.18 Consider a naturally aspirated eight-cylinder two-stroke-cycle Diesel engine with a compression ratio of 20 and a cutoff ratio of 2.5. Air is inducted into the cylinder at 1 atm and 23°C. Assume an Air Standard cycle.

(a) Determine the temperatures and pressures immediately before and after combustion.
(b) What is the heat added in the combustion process, in kJ/kg?
(c) What is the net work, in kJ/kg, and the thermal efficiency?
(d) If the volumetric efficiency is 85%, the engine displacement is 2500 cc, and the engine speed is 2200 rpm, what is the mass flow rate of air through the engine, in lb_m per minute?
(e) What is the engine horsepower?
(f) If losses through the valves cause a 120-kPa pressure differential between the pressures during the exhaust and intake strokes, estimate the actual and fractional losses, in horsepower, due to these processes. Sketch the appropriate p–V diagram.

6.19 A twelve-cylinder four-stroke-cycle Diesel engine has a 4-in. bore, a 4.5-in. stroke, and a compression ratio of 20. The mechanical efficiency is 89%, the cutoff ratios is 2, and the engine speed is 1200 rpm. The air entering the cylinder is at 14.5 psia and 60° F. Assuming Air Standard cycle performance, determine the cycle temperatures, indicated power, IMEP, and engine brake horsepower.

6.20 A hypothetical engine cycle consists of an isentropic compression, a constant-pressure heat addition, and a constant-volume blowdown, consecutively.

(a) Draw and label a p–V diagram for the cycle.
(b) Use the cyclic integral of the First Law to derive an equation for the cycle net work in terms of the temperature.
(c) Use the definition of mechanical work to derive an equation for the cycle net work also. Show that your equation is equivalent to the result obtained in part (b) using the cyclic integral.
(d) Express an equation for the cycle thermal efficiency in terms of cycle temperature ratios and k.
(e) If T_1 = 60°F and the volume ratio is 10, determine the other cycle temperatures; and compare the cycle efficiency with the efficiency the Otto cycle having the same compression ratio.

6.21 A slightly more complex model of a reciprocating engine cycle than those discussed combines constant-pressure and constant-volume heat additions in a single Air Standard cycle.

(a) Sketch and label a p–V diagram for this cycle that consists of the following

consecutive processes: isentropic compression, constant-volume heat addition, constant-pressure heat addition, isentropic expansion, and constant-volume blowdown.
(b)The engine may be characterized by three parameters: the compression ratio; the Diesel engine cutoff ratio; and a third parameter, the ratio of the pressure after to that before the constant-volume combustion. Define these parameters in terms of the symbols in your sketch and derive an equation for the thermal efficiency of the cycle.
(c) Show how varying the parameters appropriately reduces your efficiency equation to the equations for the Otto and Diesel cycles.

6.22 As a plant engineer you must recommend whether electric power for a plant expansion (2.5 MW continuous generation requirement) will be purchased from a public utility or generated using a fully attended Diesel-engine-driven generator or an automatic remotely controlled gas turbine generator set. The price of electricity is 4.8 cents per kW-hr, and the price of natural gas is 60 cents per thousand cubic feet. Both Diesel engine and gas turbine are to be natural-gas fired. The gas turbine has a heat rate of 11,500 Btu/kW-hr, and the Diesel engine 13,200 Btu/kW-hr. The initial costs of Diesel engine and gas turbine are $750,000 and $850,000, respectively. Control equipment for the gas turbine costs an additional $150,000. The engines and control equipment are estimated to have a useful life of 20 years. The annual wages and benefits for a Diesel engine operating engineer working eight-hour daily shifts is $36,000. Assume a 10% per annum interest-rate. Evaluate the alternatives for a natural gas heating value of 1000 Btu/ft^3, and present a table of their costs, in cents per kW-hr. Discuss your recommendation.

6.23 Evaluate the alternatives in Exercise 6.22 based on the present-worth method.

6.24 In terms of the notation of Figure 6.3, what are the piston displacement, compression ratio, and expansion ratio for the Lenoir cycle?

6.25 What are the fuel and air flow rates and brake specific fuel consumption for an eight-cylinder engine with a 3.75-in. bore and 3.5-in. stroke delivering 212 horsepower at 3600 rpm with a brake thermal efficiency of 25%? The fuel is C_8H_{18}, and the equivalence ratio is 1.2. What is the power per cubic inch of engine displacement?

6.26* Construct a spreadsheet to perform an Air Standard cycle analysis for a Diesel engine with a compression ratio of 20 and a range of peak temperatures from 1000K to 3000K, in 500° increments. Use it to tabulate and plot both the net work per unit mass of air and the thermal efficiency against the cutoff ratio.

6.27 Determine the maximum tailpipe concentrations of the three federally regulated gaseous pollutants based on the existing standards for an automobile that achieves 28 mile/gal of iso-octane. Assume that the engine mixture equivalence ratio is 0.9, that NO_x is represented by NO_2 and unburned hydrocarbons by monatomic carbon, and that the fuel density is 700 kg/m^3.

6.28 A single-cylinder Air Standard Otto engine has a compression ratio of 9.0 and a peak temperature of 3000°F at 80°F and one atmosphere ambient conditions. Determine the net work, cycle efficiency, maximum cylinder pressure, and mean effective pressure.

6.29 A six-cylinder engine with a compression ratio of 11 runs at 3200 rpm and 80°F and 14.7 psia. Each cylinder has a bore of 3 inches, a stroke of 3.25 inches, and a volumetric efficiency of 0.85. Assume an Air Standard four-stroke Otto cycle with stoichiometric octane as fuel. Assume that the energy release from the fuel is equally divided between internal energy increase in cylinder gases and cylinder wall heat loss. What are the cylinder mean effective pressures and the engine horsepower and specific fuel consumption? Assume a heating value of 20,600 Btu/lb_m.

6.30 An eight-cylinder four-stroke-cycle compression-ignition engine operates with a fuel-air ratio of 0.03 at 2400 rpm. It has a turbocharger and intercooler, as diagrammed nearby, with compressor pressure ratio of 1.7 and intercooler exit temperature of 320K. The engine bore and stroke are 10 cm and 12 cm, respectively. The compressor efficiency, turbine efficiency, and volumetric efficiency are 70%, 75%, and 87%, respectively. The entrance temperature of the turbine gases is 1000K. What are the compressor power, the turbine pressure ratio, and the engine power, in kW and in horsepower? Assume that the engine is constructed of ceramic components that minimize engine heat losses so that they may be neglected—and ideal "adiabatic" engine.