CHAPTER 2

ENGINE DESIGN AND OPERATING PARAMETERS

2.1 IMPORTANT ENGINE CHARACTERISTICS

In this chapter, some basic geometrical relationships and the parameters commonly used to characterize engine operation are developed. The factors important to an engine user are:

1. The engine's performance over its operating range
2. The engine's fuel consumption within this operating range and the cost of the required fuel
3. The engine's noise and air pollutant emissions within this operating range
4. The initial cost of the engine and its installation
5. The reliability and durability of the engine, its maintenance requirements, and how these affect engine availability and operating costs

These factors control total engine operating costs—usually the primary consideration of the user—and whether the engine in operation can satisfy environmental regulations. This book is concerned primarily with the performance, efficiency, and emissions characteristics of engines; the omission of the other factors listed above does not, in any way, reduce their great importance.

Engine performance is more precisely defined by:

1. The maximum power (or the maximum torque) available at each speed within the useful engine operating range
2. The range of speed and power over which engine operation is satisfactory

The following performance definitions are commonly used:

Maximum rated power. The highest power an engine is allowed to develop for short periods of operation.

Normal rated power. The highest power an engine is allowed to develop in continuous operation.

Rated speed. The crankshaft rotational speed at which rated power is developed.

2.2 GEOMETRICAL PROPERTIES OF RECIPROCATING ENGINES

The following parameters define the basic geometry of a reciprocating engine (see Fig. 2-1):

Compression ratio r_c:

$$r_c = \frac{\text{maximum cylinder volume}}{\text{minimum cylinder volume}} = \frac{V_d + V_c}{V_c} \tag{2.1}$$

where V_d is the displaced or swept volume and V_c is the clearance volume. Ratio of cylinder bore to piston stroke:

$$R_{bs} = \frac{B}{L} \tag{2.2}$$

Ratio of connecting rod length to crank radius:

$$R = \frac{l}{a} \tag{2.3}$$

In addition, the stroke and crank radius are related by

$$L = 2a$$

Typical values of these parameters are: $r_c = 8$ to 12 for SI engines and $r_c = 12$ to 24 for CI engines; $B/L = 0.8$ to 1.2 for small- and medium-size engines, decreasing to about 0.5 for large slow-speed CI engines; $R = 3$ to 4 for small- and medium-size engines, increasing to 5 to 9 for large slow-speed CI engines.

The cylinder volume V at any crank position θ is

$$V = V_c + \frac{\pi B^2}{4}(l + a - s) \tag{2.4}$$

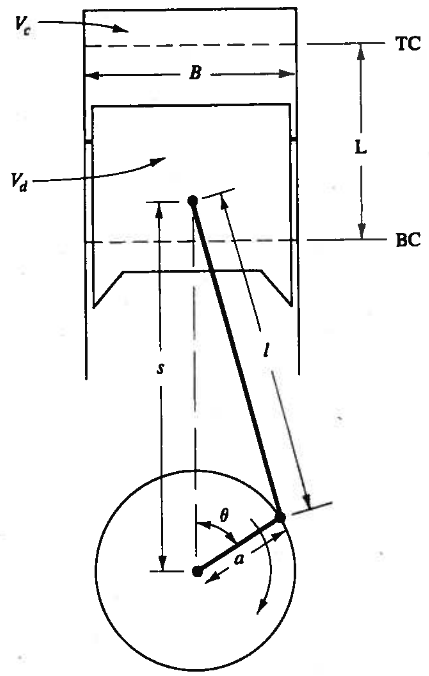

FIGURE 2-1
Geometry of cylinder, piston, connecting rod, and crankshaft where B = bore, L = stroke, l = connecting road length, a = crank radius, θ = crank angle.

where s is the distance between the crank axis and the piston pin axis (Fig. 2-1), and is given by

$$s = a\cos\theta + (l^2 - a^2\sin^2\theta)^{1/2} \tag{2.5}$$

The angle θ, defined as shown in Fig. 2-1, is called the *crank angle*. Equation (2.4) with the above definitions can be rearranged:

$$\frac{V}{V_c} = 1 + \tfrac{1}{2}(r_c - 1)[R + 1 - \cos\theta - (R^2 - \sin^2\theta)^{1/2}] \tag{2.6}$$

The combustion chamber surface area A at any crank position θ is given by

$$A = A_{ch} + A_p + \pi B(l + a - s) \tag{2.7}$$

where A_{ch} is the cylinder head surface area and A_p is the piston crown surface area. For flat-topped pistons, $A_p = \pi B^2/4$. Using Eq. (2.5), Eq. (2-7) can be rearranged:

$$A = A_{ch} + A_p + \frac{\pi BL}{2}[R + 1 - \cos\theta - (R^2 - \sin^2\theta)^{1/2}] \tag{2.8}$$

An important characteristic speed is the *mean piston speed* $\bar{S}_p$:

$$\bar{S}_p = 2LN \tag{2.9}$$

where N is the rotational speed of the crankshaft. Mean piston speed is often a

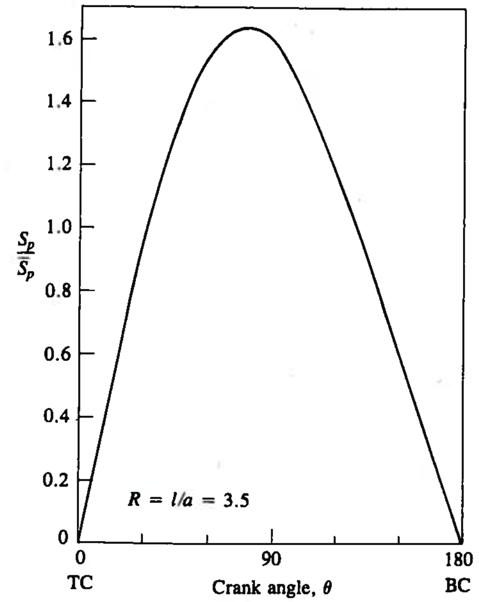

FIGURE 2-2
Instantaneous piston speed/mean piston speed as a function of crank angle for $R = 3.5$.

more appropriate parameter than crank rotational speed for correlating engine behavior as a function of speed. For example, gas-flow velocities in the intake and the cylinder all scale with $\bar{S}_p$. The *instantaneous* piston velocity S_p is obtained from

$$S_p = \frac{ds}{dt} \tag{2.10}$$

The piston velocity is zero at the beginning of the stroke, reaches a maximum near the middle of the stroke, and decreases to zero at the end of the stroke. Differentiation of Eq. (2.5) and substitution gives

$$\frac{S_p}{\bar{S}_p} = \frac{\pi}{2}\sin\theta\left[1 + \frac{\cos\theta}{(R^2 - \sin^2\theta)^{1/2}}\right] \tag{2.11}$$

Figure 2-2 shows how S_p varies over each stroke for $R = 3.5$.

Resistance to gas flow into the engine or stresses due to the inertia of the moving parts limit the maximum mean piston speed to within the range 8 to 15 m/s (1500 to 3000 ft/min). Automobile engines operate at the higher end of this range; the lower end is typical of large marine diesel engines.

2.3 BRAKE TORQUE AND POWER

Engine torque is normally measured with a dynamometer.[1] The engine is clamped on a test bed and the shaft is connected to the dynamometer rotor. Figure 2-3 illustrates the operating principle of a dynamometer. The rotor is

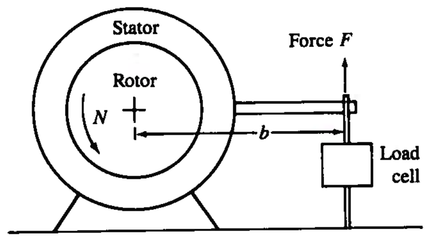

FIGURE 2-3
Schematic of principle of operation of dynamometer.

coupled electromagnetically, hydraulically, or by mechanical friction to a stator, which is supported in low friction bearings. The stator is balanced with the rotor stationary. The torque exerted on the stator with the rotor turning is measured by balancing the stator with weights, springs, or pneumatic means.

Using the notation in Fig. 2-3, if the torque exerted by the engine is T:

$$T = Fb \tag{2.12}$$

The power P delivered by the engine and absorbed by the dynamometer is the product of torque and angular speed:

$$P = 2\pi NT \tag{2.13a}$$

where N is the crankshaft rotational speed. In SI units:

$$P(\text{kW}) = 2\pi N(\text{rev/s})T(\text{N}\cdot\text{m}) \times 10^{-3} \tag{2.13b}$$

or in U.S. units:

$$P(\text{hp}) = \frac{N(\text{rev/min})\ T(\text{lbf}\cdot\text{ft})}{5252} \tag{2.13c}$$

Note that torque is a measure of an engine's ability to do work; power is the rate at which work is done.

The value of engine power measured as described above is called *brake power* P_b. This power is the usable power delivered by the engine to the load—in this case, a "brake."

2.4 INDICATED WORK PER CYCLE

Pressure data for the gas in the cylinder over the operating cycle of the engine can be used to calculate the work transfer from the gas to the piston. The cylinder pressure and corresponding cylinder volume throughout the engine cycle can be plotted on a p-V diagram as shown in Fig. 2-4. The *indicated work per cycle* $W_{c,i}$† (per cylinder) is obtained by integrating around the curve to obtain the

† The term indicated is used because such p-V diagrams used to be generated directly with a device called an engine indicator.

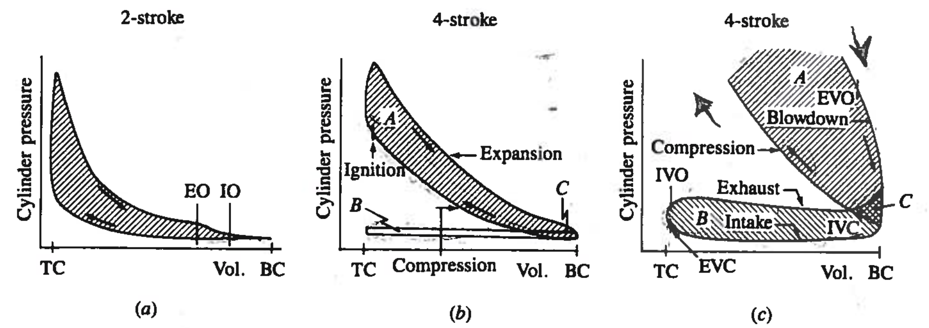

FIGURE 2-4
Examples of p-V diagrams for (*a*) a two-stroke cycle engine, (*b*) a four-stroke cycle engine; (*c*) a four-stroke cycle spark-ignition engine exhaust and intake strokes (pumping loop) at part load.

area enclosed on the diagram:

$$W_{c,i} = \oint p\, dV \tag{2.14}$$

With two-stroke cycles (Fig. 2-4*a*), the application of Eq. (2.14) is straightforward. With the addition of inlet and exhaust strokes for the four-stroke cycle, some ambiguity is introduced as two definitions of indicated output are in common use. These will be defined as:

Gross indicated work per cycle $W_{c,ig}$. Work delivered to the piston over the compression and expansion strokes only.

Net indicated work per cycle $W_{c,in}$. Work delivered to the piston over the entire four-stroke cycle.

In Fig. 2-4*b* and *c*, $W_{c,ig}$ is (area A + area C) and $W_{c,in}$ is (area A + area C) − (area B + area C), which equals (area A − area B), where each of these areas is regarded as a positive quantity. Area B + area C is the work transfer between the piston and the cylinder gases during the inlet and exhaust strokes and is called the *pumping work* W_p (see Chaps. 5 and 13). The pumping work transfer will be *to* the cylinder gases if the pressure during the intake stroke is less than the pressure during the exhaust stroke. This is the situation with naturally aspirated engines. The pumping work transfer will be *from* the cylinder gases to the piston if the exhaust stroke pressure is lower than the intake pressure, which is normally the case with highly loaded turbocharged engines.†

† With some two-stroke engine concepts there is a piston pumping work term associated with compressing the scavenging air in the crankcase.

The power per cylinder is related to the indicated work per cycle by

$$P_i = \frac{W_{c,i} N}{n_R} \tag{2.15}$$

where n_R is the number of crank revolutions for each power stroke per cylinder. For four-stroke cycles, n_R equals 2; for two-stroke cycles, n_R equals 1. This power is the indicated power; i.e., the rate of work transfer from the gas within the cylinder to the piston. It differs from the brake power by the power absorbed in overcoming engine friction, driving engine accessories, and (in the case of gross indicated power) the pumping power.

In discussing indicated quantities of the four-stroke cycle engine, such as work per cycle or power, the definition used for "indicated" (i.e., gross or net) *should always be explicitly stated.* The gross indicated output, the definition most commonly used, will be chosen where possible in this book for the following reasons. Indicated quantities are used primarily to identify the impact of the compression, combustion, and expansion processes on engine performance, etc. The gross indicated output is, therefore, the most appropriate definition. It represents the sum of the useful work available at the shaft and the work required to overcome all the engine losses. Furthermore, the standard engine test codes[2] define procedures for measuring brake power and friction power (the friction power test provides a close approximation to the total lost power in the engine). The sum of brake power and friction power provides an alternative way of estimating indicated power; the value obtained is a close approximation to the gross indicated power.

The terms brake and indicated are used to describe other parameters such as mean effective pressure, specific fuel consumption, and specific emissions (see the following sections) in a manner similar to that used for work per cycle and power.

2.5 MECHANICAL EFFICIENCY

We have seen that part of the gross indicated work per cycle or power is used to expel exhaust gases and induct fresh charge. An additional portion is used to overcome the friction of the bearings, pistons, and other mechanical components of the engine, and to drive the engine accessories. All of these power requirements are grouped together and called *friction power* P_f.† Thus:

$$P_{ig} = P_b + P_f \tag{2.16}$$

Friction power is difficult to determine accurately. One common approach for high-speed engines is to drive or motor the engine with a dynamometer (i.e., operate the engine without firing it) and measure the power which has to be

† The various components of friction power are examined in detail in Chap. 13.

supplied by the dynamometer to overcome *all* these frictional losses. The engine speed, throttle setting, oil and water temperatures, and ambient conditions are kept the same in the motored test as under firing conditions. The major sources of inaccuracy with this method are that gas pressure forces on the piston and rings are lower in the motored test than when the engine is firing and that the oil temperatures on the cylinder wall are also lower under motoring conditions.

The ratio of the brake (or useful) power delivered by the engine to the indicated power is called the *mechanical efficiency* η_m:

$$\eta_m = \frac{P_b}{P_{ig}} = 1 - \frac{P_f}{P_{ig}} \tag{2.17}$$

Since the friction power includes the power required to pump gas into and out of the engine, mechanical efficiency depends on throttle position as well as engine design and engine speed. Typical values for a modern automotive engine at wide-open or full throttle are 90 percent at speeds below about 30 to 40 rev/s (1800 to 2400 rev/min), decreasing to 75 percent at maximum rated speed. As the engine is throttled, mechanical efficiency decreases, eventually to zero at idle operation.

2.6 ROAD-LOAD POWER

A part-load power level useful as a reference point for testing automobile engines is the power required to drive a vehicle on a level road at a steady speed. Called *road-load power*, this power overcomes the rolling resistance which arises from the friction of the tires and the aerodynamic drag of the vehicle. Rolling resistance and drag coefficients, C_R and C_D, respectively, are determined empirically. An approximate formula for road-load power P_r is

$$P_r = (C_R M_v g + \tfrac{1}{2}\rho_a C_D A_v S_v^2) S_v \tag{2.18a}$$

where C_R = coefficient of rolling resistance ($0.012 < C_R < 0.015$)[3]
M_v = mass of vehicle [for passenger cars: curb mass plus passenger load of 68 kg (150 lbm); in U.S. units W_v = vehicle weight in lbf]
g = acceleration due to gravity
ρ_a = ambient air density
C_D = drag coefficient (for cars: $0.3 < C_D \lesssim 0.5$)[3]
A_v = frontal area of vehicle
S_v = vehicle speed

With the quantities in the units indicated:

$$P_r(\text{kW}) = [2.73 C_R M_v(\text{kg}) + 0.0126 C_D A_v(\text{m}^2) S_v(\text{km/h})^2] S_v(\text{km/h}) \times 10^{-3} \tag{2.18b}$$

or

$$P_r(\text{hp}) = \frac{[C_R W_v(\text{lbf}) + 0.0025 C_D A_v(\text{ft}^2) S_v(\text{mi/h})^2] S_v(\text{mi/h})}{375} \tag{2.18c}$$

2.7 MEAN EFFECTIVE PRESSURE

While torque is a valuable measure of a particular engine's ability to do work, it depends on engine size. A more useful relative engine performance measure is obtained by dividing the work per cycle by the cylinder volume displaced per cycle. The parameter so obtained has units of force per unit area and is called the *mean effective pressure* (mep). Since, from Eq. (2.15),

$$\text{Work per cycle} = \frac{P n_R}{N}$$

where n_R is the number of crank revolutions for each power stroke per cylinder (two for four-stroke cycles; one for two-stroke cycles), then

$$\text{mep} = \frac{P n_R}{V_d N} \tag{2.19a}$$

For SI and U.S. units, respectively,

$$\text{mep(kPa)} = \frac{P(\text{kW}) n_R \times 10^3}{V_d(\text{dm}^3) N(\text{rev/s})} \tag{2.19b}$$

$$\text{mep(lb/in}^2) = \frac{P(\text{hp}) n_R \times 396{,}000}{V_d(\text{in}^3) N(\text{rev/min})} \tag{2.19c}$$

Mean effective pressure can also be expressed in terms of torque by using Eq. (2.13):

$$\text{mep(kPa)} = \frac{6.28 n_R T(\text{N}\cdot\text{m})}{V_d(\text{dm}^3)} \tag{2.20a}$$

or

$$\text{mep(lb/in}^2) = \frac{75.4 n_R T(\text{lbf}\cdot\text{ft})}{V_d(\text{in}^3)} \tag{2.20b}$$

The maximum brake mean effective pressure of good engine designs is well established, and is essentially constant over a wide range of engine sizes. Thus, the actual bmep that a particular engine develops can be compared with this norm, and the effectiveness with which the engine designer has used the engine's displaced volume can be assessed. Also, for design calculations, the engine displacement required to provide a given torque or power, at a specified speed, can be estimated by assuming appropriate values for bmep for that particular application.

Typical values for bmep are as follows. For naturally aspirated spark-ignition engines, maximum values are in the range 850 to 1050 kPa (~ 125 to 150 lb/in^2) at the engine speed where maximum torque is obtained (about 3000 rev/min). At the maximum rated power, bmep values are 10 to 15 percent lower. For turbocharged automotive spark-ignition engines the maximum bmep is in the 1250 to 1700 kPa (180 to 250 lb/in^2) range. At the maximum rated power, bmep is in the 900 to 1400 kPa (130 to 200 lb/in^2) range. For naturally aspirated four-stroke diesels, the maximum bmep is in the 700 to 900 kPa (100 to 130 lb/in^2) range, with the bmep at the maximum rated power of about 700 kPa (100 lb/in^2). Turbocharged four-stroke diesel maximum bmep values are typically in the range 1000 to 1200 kPa (145 to 175 lb/in^2); for turbocharged aftercooled engines this can rise to 1400 kPa. At maximum rated power, bmep is about 850 to 950 kPa (125 to 140 lb/in^2). Two-stroke cycle diesels have comparable performance to four-stroke cycle engines. Large low-speed two-stroke cycle engines can achieve bmep values of about 1600 kPa.

An example of how the above engine performance parameters can be used to initiate an engine design is given below.

Example. A four-cylinder automotive spark-ignition engine is being designed to provide a maximum brake torque of 150 N·m (110 lbf·ft) in the mid-speed range (~ 3000 rev/min). Estimate the required engine displacement, bore and stroke, and the maximum brake power the engine will deliver.

Equation (2.20*a*) relates torque and mep. Assume that 925 kPa is an appropriate value for bmep at the maximum engine torque point. Equation (2.20*a*) gives

$$V(\text{dm}^3) = \frac{6.28 n_R T_{\max}(\text{N}\cdot\text{m})}{\text{bmep}_{\max}(\text{kPa})} = \frac{6.28 \times 2 \times 150}{925} = 2\ \text{dm}^3$$

For a four-cylinder engine, the displaced volume, bore, and stroke are related by

$$V_d = 4 \times \frac{\pi}{4} B^2 L$$

Assume $B = L$; this gives $B = L = 86$ mm.

The maximum rated engine speed can be estimated from an appropriate value for the maximum mean piston speed, 15 m/s (see Sec. 2.2):

$$\bar{S}_{p\max} = 2LN_{\max} \rightarrow N_{\max} = 87\ \text{rev/s}\ (5200\ \text{rev/min})$$

The maximum brake power can be estimated from the typical bmep value at maximum power, 800 kPa (116 lb/in^2), using Eq. (2.19*b*):

$$P_{b\max}(\text{kW}) = \frac{\text{bmep(kPa)} V(\text{dm}^3) N_{\max}(\text{rev/s})}{n_R \times 10^3} = \frac{800 \times 2 \times 87}{2 \times 10^3} = 70\ \text{kW}$$

2.8 SPECIFIC FUEL CONSUMPTION AND EFFICIENCY

In engine tests, the fuel consumption is measured as a flow rate—mass flow per unit time $\dot{m}_f$. A more useful parameter is the *specific fuel consumption* (sfc)—the fuel flow rate per unit power output. It measures how efficiently an engine is using the fuel supplied to produce work:

$$\text{sfc} = \frac{\dot{m}_f}{P} \tag{2.21}$$

With units,

$$\text{sfc(mg/J)} = \frac{\dot{m}_f(\text{g/s})}{P(\text{kW})} \tag{2.22a}$$

or

$$\text{sfc(g/kW}\cdot\text{h)} = \frac{\dot{m}_f(\text{g/h})}{P(\text{kW})} = 608.3\ \text{sfc(lbm/hp}\cdot\text{h)} \tag{2.22b}$$

or

$$\text{sfc(lbm/hp}\cdot\text{h)} = \frac{\dot{m}_f(\text{lbm/h})}{P(\text{hp})} = 1.644 \times 10^3\ \text{sfc(g/kW}\cdot\text{h)} \tag{2.22c}$$

Low values of sfc are obviously desirable. For SI engines typical best values of brake specific fuel consumption are about 75 μg/J = 270 g/kW·h = 0.47 lbm/hp·h. For CI engines, best values are lower and in large engines can go below 55 μg/J = 200 g/kW·h = 0.32 lbm/hp·h.

The specific fuel consumption has units. A dimensionless parameter that relates the desired engine output (work per cycle or power) to the necessary input (fuel flow) would have more fundamental value. The ratio of the work produced per cycle to the amount of fuel energy supplied per cycle that can be released in the combustion process is commonly used for this purpose. It is a measure of the engine's efficiency. The fuel energy supplied which can be released by combustion is given by the mass of fuel supplied to the engine per cycle times the heating value of the fuel. The heating value of a fuel, Q_{HV}, defines its energy content. It is determined in a standardized test procedure in which a known mass of fuel is fully burned with air, and the thermal energy released by the combustion process is absorbed by a calorimeter as the combustion products cool down to their original temperature.

This measure of an engine's "efficiency," which will be called the *fuel conversion efficiency* η_f,† is given by

$$\eta_f = \frac{W_c}{m_f Q_{HV}} = \frac{(P n_R/N)}{(\dot{m}_f n_R/N) Q_{HV}} = \frac{P}{\dot{m}_f Q_{HV}} \tag{2.23}$$

where m_f is the mass of fuel inducted per cycle. Substitution for $P/\dot{m}_f$ from Eq. (2.21) gives

$$\eta_f = \frac{1}{\text{sfc}\ Q_{HV}} \tag{2.24a}$$

† This empirically defined engine efficiency has previously been called thermal efficiency or enthalpy efficiency. The term fuel conversion efficiency is preferred because it describes this quantity more precisely, and distinguishes it clearly from other definitions of engine efficiency which will be developed in Sec. 3.6. Note that there are several different definitions of heating value (see Sec. 3.5). The numerical values do not normally differ by more than a few percent, however. In this text, the lower heating value at constant pressure is used in evaluating the fuel conversion efficiency.

or with units:

$$\eta_f = \frac{1}{\text{sfc(mg/J)}Q_{HV}(\text{MJ/kg})} \tag{2.24b}$$

$$\eta_f = \frac{3600}{\text{sfc(g/kW}\cdot\text{h)}Q_{HV}(\text{MJ/kg})} \tag{2.24c}$$

$$\eta_f = \frac{2545}{\text{sfc(lbm/hp}\cdot\text{h)}Q_{HV}(\text{Btu/lbm})} \tag{2.24d}$$

Typical heating values for the commercial hydrocarbon fuels used in engines are in the range 42 to 44 MJ/kg (18,000 to 19,000 Btu/lbm). Thus, specific fuel consumption is inversely proportional to fuel conversion efficiency for normal hydrocarbon fuels.

Note that the fuel energy supplied to the engine per cycle is not fully released as thermal energy in the combustion process because the actual combustion process is incomplete. When enough air is present in the cylinder to oxidize the fuel completely, almost all (more than about 96 percent) of this fuel energy supplied is transferred as thermal energy to the working fluid. When insufficient air is present to oxidize the fuel completely, lack of oxygen prevents this fuel energy supplied from being fully released. This topic is discussed in more detail in Secs. 3.5 and 4.9.4.

2.9 AIR/FUEL AND FUEL/AIR RATIOS

In engine testing, both the air mass flow rate $\dot{m}_a$ and the fuel mass flow rate $\dot{m}_f$ are normally measured. The ratio of these flow rates is useful in defining engine operating conditions:

$$\text{Air/fuel ratio } (A/F) = \frac{\dot{m}_a}{\dot{m}_f} \tag{2.25}$$

$$\text{Fuel/air ratio } (F/A) = \frac{\dot{m}_f}{\dot{m}_a} \tag{2.26}$$

The normal operating range for a conventional SI engine using gasoline fuel is $12 \le A/F \le 18$ $(0.056 \le F/A \le 0.083)$; for CI engines with diesel fuel, it is $18 \le A/F \le 70$ $(0.014 \le F/A \le 0.056)$.

2.10 VOLUMETRIC EFFICIENCY

The intake system—the air filter, carburetor, and throttle plate (in a spark-ignition engine), intake manifold, intake port, intake valve—restricts the amount of air which an engine of given displacement can induct. The parameter used to measure the effectiveness of an engine's induction process is the *volumetric efficiency* η_v. Volumetric efficiency is only used with four-stroke cycle engines which have a distinct induction process. It is defined as the volume flow rate of air into the intake system divided by the rate at which volume is displaced by the piston:

$$\eta_v = \frac{2\dot{m}_a}{\rho_{a,i} V_d N} \tag{2.27a}$$

where $\rho_{a,i}$ is the inlet air density. An alternative equivalent definition for volumetric efficiency is

$$\eta_v = \frac{m_a}{\rho_{a,i} V_d} \tag{2.27b}$$

where m_a is the mass of air inducted into the cylinder per cycle.

The inlet density may either be taken as atmosphere air density (in which case η_v measures the pumping performance of the entire inlet system) or may be taken as the air density in the inlet manifold (in which case η_v measures the pumping performance of the inlet port and valve only). Typical maximum values of η_v for naturally aspirated engines are in the range 80 to 90 percent. The volumetric efficiency for diesels is somewhat higher than for SI engines. Volumetric efficiency is discussed more fully in Sec. 6.2.

2.11 ENGINE SPECIFIC WEIGHT AND SPECIFIC VOLUME

Engine weight and bulk volume for a given rated power are important in many applications. Two parameters useful for comparing these attributes from one engine to another are:

$$\text{Specific weight} = \frac{\text{engine weight}}{\text{rated power}} \tag{2.28}$$

$$\text{Specific volume} = \frac{\text{engine volume}}{\text{rated power}} \tag{2.29}$$

For these parameters to be useful in engine comparisons, a consistent definition of what components and auxiliaries are included in the term "engine" must be adhered to. These parameters indicate the effectiveness with which the engine designer has used the engine materials and packaged the engine components.[4]

2.12 CORRECTION FACTORS FOR POWER AND VOLUMETRIC EFFICIENCY

The pressure, humidity, and temperature of the ambient air inducted into an engine, at a given engine speed, affect the air mass flow rate and the power output. Correction factors are used to adjust measured wide-open-throttle power and volumetric efficiency values to standard atmospheric conditions to provide a more accurate basis for comparisons between engines. Typical standard ambient

conditions used are:

Dry air pressure	Water vapour pressure	Temperature
736.6 mmHg 29.00 inHg	9.65 mmHg 0.38 inHg	29.4°C 85°F

The basis for the correction factor is the equation for one-dimensional steady compressible flow through an orifice or flow restriction of effective area A_E (see App. C):

$$\dot{m} = \frac{A_E p_0}{\sqrt{RT_0}} \left\{ \frac{2\gamma}{\gamma - 1} \left[\left(\frac{p}{p_0} \right)^{2/\gamma} - \left(\frac{p}{p_0} \right)^{(\gamma+1)/\gamma} \right] \right\}^{1/2} \tag{2.30}$$

In deriving this equation, it has been assumed that the fluid is an ideal gas with gas constant R and that the ratio of specific heats ($c_p/c_v = \gamma$) is a constant; p_0 and T_0 are the total pressure and temperature upstream of the restriction and p is the pressure at the throat of the restriction.

If, in the engine, p/p_0 is assumed constant at wide-open throttle, then for a given intake system and engine, the mass flow rate of dry air $\dot{m}_a$ varies as

$$\dot{m}_a \propto \frac{p_0}{\sqrt{T_0}} \tag{2.31}$$

For mixtures containing the proper amount of fuel to use all the air available (and thus provide maximum power), the indicated power at full throttle P_i will be proportional to $\dot{m}_a$, the *dry* air flow rate. Thus if

$$P_{i,s} = C_F P_{i,m} \tag{2.32}$$

where the subscripts s and m denote values at the standard and measured conditions, respectively, the correction factor C_F is given by

$$C_F = \frac{p_{s,d}}{p_m - p_{v,m}} \left(\frac{T_m}{T_s} \right)^{1/2} \tag{2.33}$$

where $p_{s,d}$ = standard dry-air absolute pressure
p_m = measured ambient-air absolute pressure
$p_{v,m}$ = measured ambient–water vapour partial pressure
T_m = measured ambient temperature, K
T_s = standard ambient temperature, K

The rated *brake* power is corrected by using Eq. (2.33) to correct the *indicated* power and making the assumption that friction power is unchanged. Thus

$$P_{b,s} = C_F P_{i,m} - P_{f,m} \tag{2.34}$$

Volumetric efficiency is proportional to $\dot{m}_a/\rho_a$ [see Eq. (2.27)]. Since ρ_a is proportional to p/T, the correction factor for volumetric efficiency, C'_F, is

$$C'_F = \frac{\eta_{v,s}}{\eta_{v,m}} = \left(\frac{T_s}{T_m} \right)^{1/2} \tag{2.35}$$

2.13 SPECIFIC EMISSIONS AND EMISSIONS INDEX

Levels of emissions of oxides of nitrogen (nitric oxide, NO, and nitrogen dioxide, NO_2, usually grouped together as NO_x), carbon monoxide (CO), unburned hydrocarbons (HC), and particulates are important engine operating characteristics.

The concentrations of gaseous emissions in the engine exhaust gases are usually measured in parts per million or percent by volume (which corresponds to the mole fraction multiplied by 10^6 or by 10^2, respectively). Normalized indicators of emissions levels are more useful, however, and two of these are in common use. *Specific emissions* are the mass flow rate of pollutant per unit power output:

$$sNO_x = \frac{\dot{m}_{NO_x}}{P} \tag{2.36a}$$

$$sCO = \frac{\dot{m}_{CO}}{P} \tag{2.36b}$$

$$sHC = \frac{\dot{m}_{HC}}{P} \tag{2.36c}$$

$$sPart = \frac{\dot{m}_{part}}{P} \tag{2.36d}$$

Indicated and brake specific emissions can be defined. Units in common use are μg/J, g/kW · h, and g/hp · h.

Alternatively, emission rates can be normalized by the fuel flow rate. An *emission index* (EI) is commonly used: e.g.,

$$EI_{NO_x} = \frac{\dot{m}_{NO_x}(\mathrm{g/s})}{\dot{m}_f(\mathrm{kg/s})} \tag{2.37}$$

with similar expressions for CO, HC, and particulates.

2.14 RELATIONSHIPS BETWEEN PERFORMANCE PARAMETERS

The importance of the parameters defined in Secs. 2.8 to 2.10 to engine performance becomes evident when power, torque, and mean effective pressure are expressed in terms of these parameters. From the definitions of engine power [Eq. (2.13)], mean effective pressure [Eq. (2.19)], fuel conversion efficiency [Eq. (2.23)], fuel/air ratio [Eq. (2.26)], and volumetric efficiency [Eq. (2.27)], the following relationships between engine performance parameters can be developed. For power P:

$$P = \frac{\eta_f m_a N Q_{HV}(F/A)}{n_R} \tag{2.38}$$

For four-stroke cycle engines, volumetric efficiency can be introduced:

$$P = \frac{\eta_f \eta_v N V_d Q_{HV} \rho_{a,i}(F/A)}{2} \tag{2.39}$$

For torque T:

$$T = \frac{\eta_f \eta_v V_d Q_{HV} \rho_{a,i}(F/A)}{4\pi} \tag{2.40}$$

For mean effective pressure:

$$\mathrm{mep} = \eta_f \eta_v Q_{HV} \rho_{a,i}(F/A) \tag{2.41}$$

The power per unit piston area, often called the *specific power*, is a measure of the engine designer's success in using the available piston area regardless of cylinder size. From Eq. (2.39), the specific power is

$$\frac{P}{A_p} = \frac{\eta_f \eta_v N L Q_{HV} \rho_{a,i}(F/A)}{2} \tag{2.42}$$

Mean piston speed can be introduced with Eq. (2.9) to give

$$\frac{P}{A_p} = \frac{\eta_f \eta_v \bar{S}_p Q_{HV} \rho_{a,i}(F/A)}{4} \tag{2.43}$$

Specific power is thus proportional to the product of mean effective pressure and mean piston speed.

These relationships illustrate the direct importance to engine performance of:

1. High fuel conversion efficiency
2. High volumetric efficiency
3. Increasing the output of a given displacement engine by increasing the inlet air density
4. Maximum fuel/air ratio that can be usefully burned in the engine
5. High mean piston speed

2.15 ENGINE DESIGN AND PERFORMANCE DATA

Engine ratings usually indicate the highest power at which manufacturers expect their products to give satisfactory economy, reliability, and durability under service conditions. Maximum torque, and the speed at which it is achieved, is usually given also. Since both of these quantities depend on displaced volume, for comparative analyses between engines of different displacements in a given engine category normalized performance parameters are more useful. The following measures, at the operating points indicated, have most significance:[4]

TABLE 2.1
Typical design and operating data for internal combustion engines

	Operating cycle	Compression ratio	Bore, m	Stroke/bore	Rated maximum: Speed, rev/min	Rated maximum: bmep, atm	Rated maximum: Power per unit volume kW/dm^3	Weight/power ratio, kg/kW	Approx. best bsfc, g/kW·h
Spark-ignition engines:									
Small (e.g., motorcycles)	2S,4S	6–11	0.05–0.085	1.2–0.9	4500–7500	4–10	20–60	5.5–2.5	350
Passenger cars	4S	8–10	0.07–0.1	1.1–0.9	4500–6500	7–10	20–50	4–2	270
Trucks	4S	7–9	0.09–0.13	1.2–0.7	3600–5000	6.5–7	25–30	6.5–2.5	300
Large gas engines	2S,4S	8–12	0.22–0.45	1.1–1.4	300–900	6.8–12	3–7	23–35	200
Wankel engines	4S	≈ 9	0.57 dm^3 per chamber		6000–8000	9.5–10.5	35–45	1.6–0.9	300
Diesel engines:									
Passenger cars	4S	17–23	0.075–0.1	1.2–0.9	4000–5000	5–7.5	18–22	5–2.5	250
Trucks (NA)	4S	16–22	0.1–0.15	1.3–0.8	2100–4000	6–9	15–22	7–4	210
Trucks (TC)	4S	14–20	0.1–0.15	1.3–0.8	2100–4000	12–18	18–26	7–3.5	200
Locomotive, industrial, marine	4S,2S	12–18	0.15–0.4	1.1–1.3	425–1800	7–23	5–20	6–18	190
Large engines, marine and stationary	2S	10–12	0.4–1	1.2–3	110–400	9–17	2–8	12–50	180

1. At maximum or normal rated point:

   Mean piston speed. Measures comparative success in handling loads due to inertia of the parts, resistance to air flow, and/or engine friction.

   Brake mean effective pressure. In naturally aspirated engines bmep is not stress limited. It then reflects the product of volumetric efficiency (ability to induct air), fuel/air ratio (effectiveness of air utilization in combustion), and fuel conversion efficiency. In supercharged engines bmep indicates the degree of success in handling higher gas pressures and thermal loading.

   Power per unit piston area. Measures the effectiveness with which the piston area is used, regardless of cylinder size.

   Specific weight. Indicates relative economy with which materials are used.

   Specific volume. Indicates relative effectiveness with which engine space has been utilized.

2. At all speeds at which the engine will be used with full throttle or with maximum fuel-pump setting:

   Brake mean effective pressure. Measures ability to obtain/provide high air flow and use it effectively over the full range.

3. At all useful regimes of operation and particularly in those regimes where the engine is run for long periods of time:

   Brake specific fuel consumption or *fuel conversion efficiency.*
   Brake specific emissions.

Typical performance data for spark-ignition and diesel engines over the normal production size range are summarized in Table 2.1.[4] The four-stroke cycle dominates except in the smallest and largest engine sizes. The larger engines are turbocharged or supercharged. The maximum rated engine speed decreases as engine size increases, maintaining the maximum mean piston speed in the range of about 8 to 15 m/s. The maximum brake mean effective pressure for turbocharged and supercharged engines is higher than for naturally aspirated engines. Because the maximum fuel/air ratio for spark-ignition engines is higher than for diesels, their naturally aspirated maximum bmep levels are higher. As engine size increases, brake specific fuel consumption decreases and fuel conversion efficiency increases, due to reduced importance of heat losses and friction. For the largest diesel engines, brake fuel conversion efficiencies of about 50 percent and indicated fuel conversion efficiencies of over 55 percent can be obtained.

PROBLEMS

2.1. Explain why the brake mean effective pressure of a naturally aspirated diesel engine is lower than that of a naturally aspirated spark-ignition engine. Explain why the bmep is lower at the maximum rated power for a given engine than the bmep at the maximum torque.