CHAPTER 59
INTERNAL COMBUSTION ENGINES

Ronald Douglas Matthews
General Motors Foundation Combustion Sciences and Automotive Research Laboratories
The University of Texas at Austin
Austin, Texas

An internal combustion engine is a device that operates on an open thermodynamic cycle and is used to convert the chemical energy of a fuel to rotational mechanical energy. This rotational mechanical energy is most often used directly to provide motive power through an appropriate drive train, such as for an automotive application. The rotational mechanical energy may also be used directly to drive a propeller for marine or aircraft applications. Alternatively, the internal combustion engine may be coupled to a generator to provide electric power or may be coupled to hydraulic pump or a gas compressor. It may be noted that the favorable power-to-weight ratio of the internal combustion engine makes it ideally suited to mobile applications and therefore most internal combustion engines are manufactured for the motor vehicle, rail, marine, and aircraft industries. The high power-to-weight ratio of the internal combustion engine is also responsible for its use in other applications where a lightweight power source is needed, such as for chain saws and lawn mowers.

This chapter is devoted to discussion of the internal combustion engine, including types, principles of operation, fuels, theory, performance, efficiency, and emissions.

59.1 TYPES AND PRINCIPLES OF OPERATION

This chapter discusses internal combustion engines that have an intermittent combustion process. Gas turbines, which are internal combustion engines that incorporate a continuous combustion system, are discussed in a separate chapter.

Internal combustion (IC) engines may be most generally classified by the method used to initiate combustion as either spark ignition (SI) or compression ignition (CI or diesel) engines. Another general classification scheme involves whether the rotational mechanical energy is obtained via reciprocating piston motion, as is more common, or directly via the rotational motion of a rotor in a rotary (Wankel) engine (see Fig. 59.1). The physical principles of a rotary engine are equivalent to those of a piston engine if the geometric considerations are properly accounted for, so that the following discussion will focus on the piston engine and the rotary engine will be discussed only briefly. All of these IC engines include five general processes:

Mechanical Engineers' Handbook, 2nd ed., Edited by Myer Kutz.
ISBN 0-471-13007-9

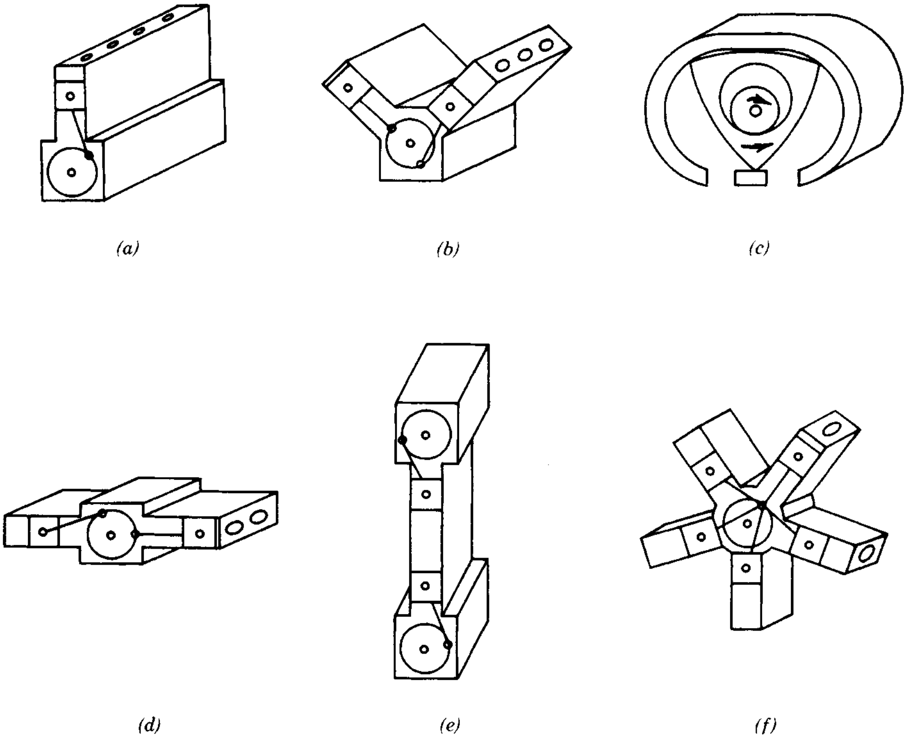

Fig. 59.1 IC engine configurations: (*a*) inline 4; (*b*) V6; (*c*) rotary (Wankel); (*d*) horizontal, flat, or opposed cylinder; (e) opposed piston; (*f*) radial.

1. An intake process, during which air or a fuel–air mixture is inducted into the combustion chamber
2. A compression process, during which the air or fuel–air mixture is compressed to higher temperature, pressure, and density
3. A combustion process, during which the chemical energy of the fuel is converted to thermal energy of the products of combustion
4. An expansion process, during which a portion of the thermal energy of the working fluid is converted to mechanical energy
5. An exhaust process, during which most of the products of combustion are expelled from the combustion chamber

The mechanics of how these five general processes are incorporated in an engine may be used to more specifically classify different types of internal combustion engines.

59.1.1 Spark Ignition Engines

In SI engines, the combustion process is initiated by a precisely timed discharge of a spark across an electrode gap in the combustion chamber. Before ignition, the combustible mixture may be either homogeneous (i.e., the fuel–air mixture ratio may be approximately uniform throughout the combustion chamber) or stratified (i.e., the fuel–air mixture ratio may be more fuel-lean in some regions of the combustion chamber than in other portions). In all SI engines, except the direct injection stratified charge (DISC) SI engine, the power output is controlled by controlling the air flow rate (and thus the volumetric efficiency) through the engine and the fuel–air ratio is approximately constant (and approximately stoichiometric) for almost all operating conditions. The power output of the DISC engine is controlled by varying the fuel flow rate, and thus the fuel–air ratio is variable while the volumetric efficiency is approximately constant. The fuel and air are premixed before entering the combustion chamber in all SI engines except the direct injection SI engine. These various categories of SI engines are discussed below.

Homogeneous Charge SI Engines

In the homogeneous charge SI engine, a mixture of fuel and air is inducted during the intake process. Traditionally, the fuel was mixed with the air in the venturi section of a carburetor. More recently, as more precise control of the fuel–air ratio became desirable, throttle body fuel injection took the place of carburetors for most automotive applications. Even more recently, intake port fuel injection has almost entirely replaced throttle body injection. The five processes mentioned above may be combined in the homogeneous charge SI engine to produce an engine that operates on either a 4-stroke cycle or on a 2-stroke cycle.

4-Stroke Homogeneous Charge SI Engines. In the more common 4-stroke cycle (see Fig. 59.2), the first stroke is the movement of the piston from top dead center (TDC—the closest approach of the piston to the cylinder head, yielding the minimum combustion chamber volume) to bottom dead center (BDC—when the piston is farthest from the cylinder head, yielding the maximum combustion chamber volume), during which the intake valve is open and the fresh fuel–air charge is inducted into the combustion chamber. The second stroke is the compression process, during which the intake and exhaust valves are both in the closed position and the piston moves from BDC back to TDC. The compression process is followed by combustion of the fuel–air mixture. Combustion is a rapid hydrocarbon oxidation process (not an explosion) of finite duration. Because the combustion process requires a finite, though very short, period of time, the spark is timed to initiate combustion slightly before the piston reaches TDC to allow the maximum pressure to occur slightly after TDC (peak pressure should, optimally, occur after TDC to provide a torque arm for the force caused by the high cylinder pressure). The combustion process is essentially complete shortly after the piston has receded away from TDC. However, for the purposes of a simple analysis and because combustion is very rapid, to aid explanation it may be approximated as being instantaneous and occurring while the piston is motionless at TDC. The third stroke is the expansion process or power stroke, during which the piston returns to BDC. The fourth stroke is the exhaust process, during which the exhaust valve is open and the piston proceeds from BDC to TDC and expels the products of combustion. The exhaust process for a 4-stroke engine is actually composed of two parts, the first of which is blowdown. When the exhaust valve opens, the cylinder pressure is much higher than the pressure in the exhaust manifold and this large pressure difference forces much of the exhaust out during what is called "blowdown" while the piston is almost motionless. Most of the remaining products of combustion are forced out during the exhaust stroke, but an "exhaust residual" is always left in the combustion chamber and mixes with the fresh charge that is inducted during the subsequent intake stroke. Once the piston reaches TDC, the intake valve opens and the exhaust valve closes and the cycle repeats, starting with a new intake stroke.

This explanation of the 4-stroke SI engine processes implied that the valves open or close instantaneously when the piston is either at TDC or BDC, when in fact the valves open and close relatively slowly. To afford the maximum open area at the appropriate time in each process, the exhaust valve opens before BDC during expansion, the intake valve closes after BDC during the compression stroke, and both the intake and exhaust valves are open during the valve overlap period since the intake valve opens before TDC during the exhaust stroke while the exhaust valve closes after TDC during the intake stroke. Considerations of valve timing are not necessary for this simple explanation of the 4-stroke cycle but do have significant effects on performance and efficiency. Similarly, spark timing will not be discussed in detail but does have significant effects on performance, fuel economy, and emissions.

The rotary (Wankel) engine is sometimes perceived to operate on the 2-stroke cycle because it shares several features with 2-stroke SI engines: a complete thermodynamic cycle within a single revolution of the output shaft (which is called an eccentric shaft rather than a crank shaft) and lack of intake and exhaust valves and associated valve train. However, unlike a 2-stroke, the rotary has a true exhaust "stroke" and a true intake "stroke" and operates quite well without boosting the pressure of the fresh charge above that of the exhaust manifold. That is, the rotary operates on the 4-stroke cycle.

2-Stroke Homogeneous Charge SI Engines. Alternatively, these five processes may be incorporated into a homogeneous charge SI engine that requires only two strokes per cycle (see Fig. 59.3). All commercially available 2-stroke SI engines are of the homogeneous charge type. That is, any nonuniformity of the fuel–air ratio within the combustion chamber is unintentional in current 2-stroke SI engines. The 2-stroke SI engine does not have valves, but rather has intake "transfer" and exhaust ports that are normally located across from each other near the position of the crown of the piston when the piston is at BDC. When the piston moves toward TDC, it covers the ports and the compression process begins. As previously discussed, for the ideal SI cycle, combustion may be perceived to occur instantaneously while the piston is motionless at TDC. The expansion process then occurs as the high pressure resulting from combustion pushes the piston back toward BDC. As the piston approaches BDC, the exhaust port is generally uncovered first, followed shortly thereafter by uncovering of the intake transfer port. The high pressure in the combustion chamber relative to that of the

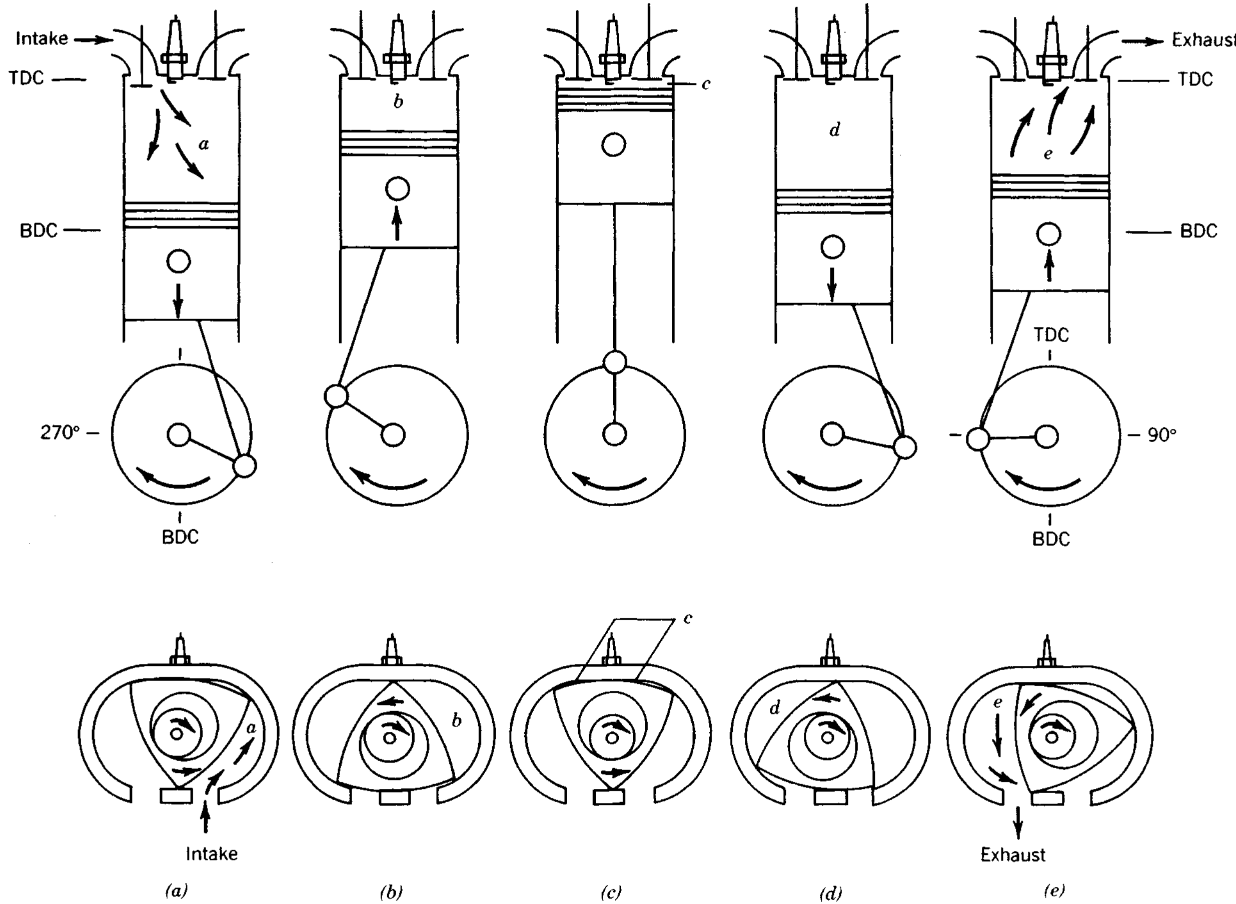

Fig. 59.2 Schematic of processes for 4-stroke SI piston and rotary engines (for 4-stroke CI, replace spark plug with fuel injector): (*a*) intake, (*b*) compression, (*c*) spark ignition and combustion (for CI, fuel injection, and autoignition), (*d*) expansion or power stroke, (*e*) exhaust.

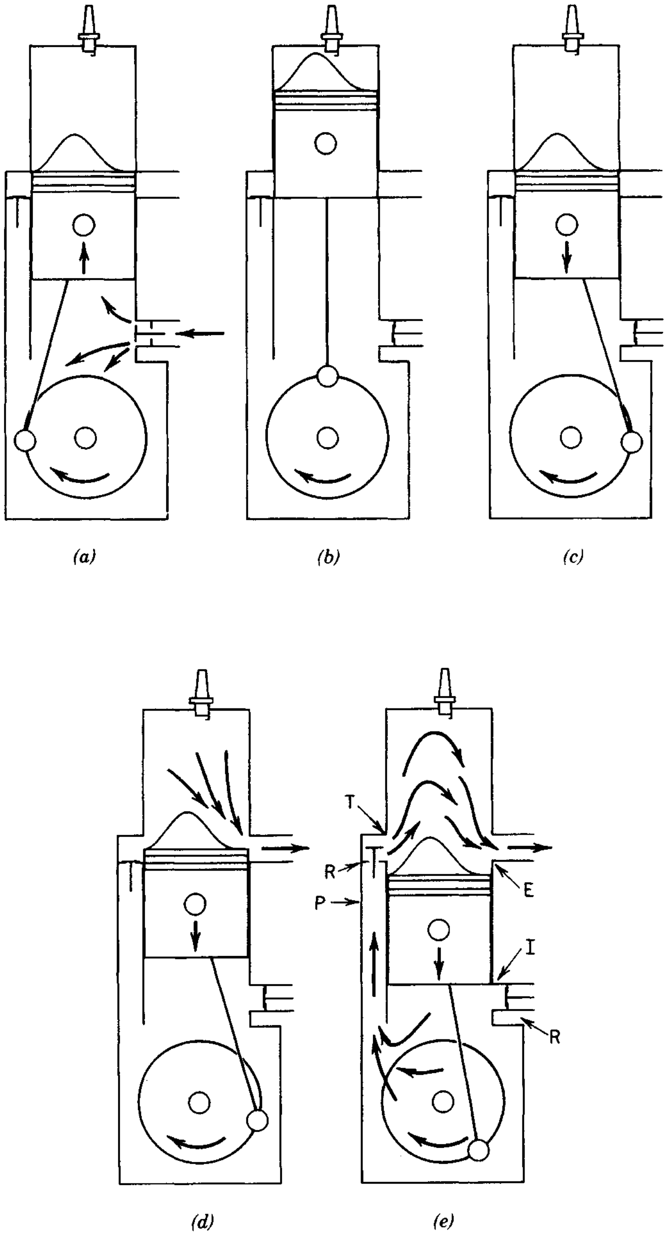

Fig. 59.3 Processes for 2-stroke crankcase compression SI engine (for CI engine, replace spark plug with fuel injector): (*a*) compression of trapped working fluid and simultaneous intake to crankcase, (*b*) spark ignition and combustion (for CI, fuel injection, and autoignition), (*c*) expansion or power, (*d*) beginning of exhaust, (*e*) intake and "loop" scavenging. E, exhaust port; I, intake port; P, transfer passage; R, read valve; T, transfer port.

exhaust manifold results in "blowdown" of much of the exhaust before the intake transfer port is uncovered. However, as soon as the intake transfer port is uncovered, the exhaust and intake processes can occur simultaneously. However, if the chamber pressure is high with respect to the pressure in the transfer passage, the combustion products can flow into the transfer passage. To prevent this, a reed valve can be located within the intake transfer passage, as illustrated in Fig. 59.3. Alternatively, a disc valve that is attached to the crankshaft can be used to control timing of the intake transfer process. Independent of when and how the intake transfer process is initiated, the momentum of the exhaust flowing out the exhaust port will entrain some fresh charge, resulting in short-circuiting of fuel out the exhaust. This results in relatively high emissions of unburned hydrocarbons and a fuel economy penalty. This problem is minimized but not eliminated by designing the port shapes and/ or piston crown to direct the intake flow toward the top of the combustion chamber so that the fresh charge must travel a longer path before reaching the exhaust port. After the piston reaches BDC and moves back up to cover the exhaust port again, the exhaust process is over. Thus, one of the strokes that is required for the 4-stroke cycle has been eliminated by not having an exhaust stroke. The penalty is that the 2-stroke has a relatively high exhaust residual fraction (the mass fraction of the remaining combustion products relative to the total mass trapped upon port closing).

As the piston proceeds from TDC to BDC on the expansion stroke, it compresses the fuel–air mixture which is routed through the crankcase on many modern 2-stroke SI engines. To prevent backflow of the fuel–air mixture back out of the crankcase through the carburetor, a reed valve may be located between the carburetor exit and the crankcase, as illustrated in Fig. 59.3. This crankcase compression process of the fuel–air mixture results in the fuel–air mixture being at relatively high pressure when the intake transfer port is uncovered. When the pressure in the combustion chamber becomes less than the pressure of the fuel–air mixture in the crankcase, the reed valve in the transfer passage opens and the intake charge flows into the combustion chamber. Thus, the 4-stroke's intake stroke is eliminated in the 2-stroke design by having both sides of the piston do work.

Because it is important to fill the combustion chamber as completely as possible with fresh fuel–air charge and thus important to purge the combustion chamber as completely as possible of combustion products, 2-stroke SI engines are designed to promote scavenging of the exhaust products via fluid dynamics (see Figs. 59.3 and 59.6). Scavenging results in the flow of some unburned fuel through the exhaust port during the period when the transfer passage reed valve and the exhaust port are both open. This results in poor combustion efficiency, a fuel economy penalty, and high emissions of hydrocarbons. However, since the 2-stroke SI engine has one power stroke per crankshaft revolution, it develops as much as 80% more power per unit weight than a comparable 4-stroke SI engine, which has only one power stroke per every two crankshaft revolutions. Therefore, the 2-stroke SI engine is best suited for applications for which a very high power per unit weight is needed and fuel economy and pollutant emissions are not significant considerations.

Stratified Charge SI Engines

All commercially available stratified charge SI engines in the United States operate on the 4-stroke cycle, although there has been a significant effort to develop a direct injection (stratified) 2-stroke SI engine. They may be subclassified as being either divided chamber or direct injection SI engines.

Divided Chamber. The divided chamber SI engine, as shown in Fig. 59.4, generally has two intake systems: one providing a stoichiometric or slightly fuel-rich mixture to a small prechamber and the other providing a fuel-lean mixture to the main combustion chamber. A spark plug initiates combustion in the prechamber. A jet of hot reactive species then flows through the orifice separating the two chambers and ignites the fuel-lean mixture in the main chamber. In this manner, the stoichiometric or fuel-rich combustion process stabilizes the fuel-lean combustion process that would otherwise be prone to misfire. This same stratified charge concept can be attained solely via fluid mechanics, thereby eliminating the complexity of the prechamber, but the motivation is the same as for the divided chamber engine. This overall fuel-lean system is desired since it can result in decreased emissions of the regulated pollutants in comparison to the usual, approximately stoichiometric, combustion process. Furthermore, lean operation produces a thermal efficiency benefit. For these reasons, there have been many attempts to develop a lean-burn homogeneous charge SI engine. However, the emissions of the oxides of nitrogen (NO_x) peak for a slightly lean mixture before decreasing to very low values when the mixture is extremely lean. Unfortunately, most lean-burn homogeneous charge SI engines cannot operate sufficiently lean—before encountering ignition problems—that they produce a significant NO_x benefit. The overall lean-burn stratified charge SI engine avoids these ignition limits by producing an ignitable mixture in the vicinity of the spark plug but a very lean mixture far from the spark plug. Unfortunately, the flame zone itself is nearly stoichiometric, resulting in much higher emissions of NO_x than would be expected from the overall extremely lean fuel–air ratio. For this reason, divided chamber SI engines are becoming rare. However, the direct injection process offers promise of overcoming this obstacle, as discussed below.

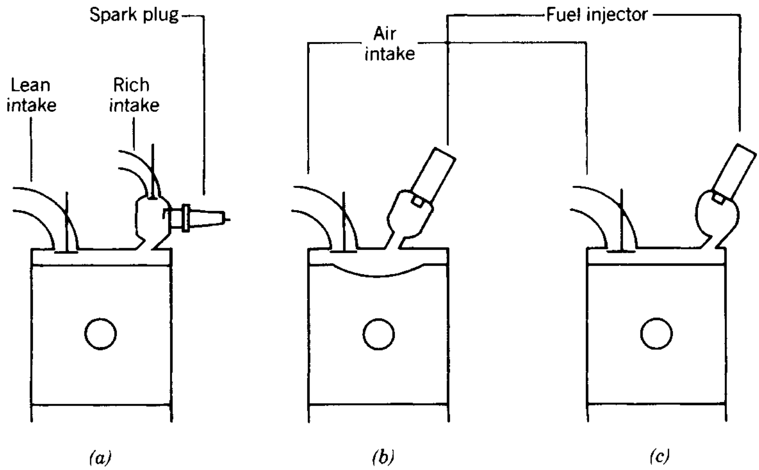

Fig. 59.4 Schematic cross sections of divided chamber engines: (*a*) prechamber SI, (*b*) prechamber IDI diesel, (*c*) swirl chamber IDI diesel.

Direct Injection. In the direct injection engine, only air is inducted during the intake stroke. The direct injection engine can be divided into two categories: early and late injection.

The first 40 years of development of the direct injection SI engine focussed upon late injection. This version is commonly known as the *direct injection stratified charge* (DISC) engine. As shown in Fig. 59.5, fuel is injected late in the compression stroke near the center of the combustion chamber and ignited by a spark plug. The DISC engine has three primary advantages:

1. A wide fuel tolerance, that is, the ability to burn fuels with a relatively low octane rating without knock (see Section 59.2).
2. This decreased tendency to knock allows use of a higher compression ratio, which in turn results in higher power per unit displacement and higher efficiency (see Section 59.3).
3. Since the power output is controlled by the amount of fuel injected instead of the amount of air inducted, the DISC engine is not throttled (except at idle), resulting in higher volumetric efficiency and higher power per unit displacement for part load conditions (see Section 59.3).

Unfortunately, the DISC engine is also prone to high emissions of unburned hydrocarbons.

However, more recent developments in the DISC engine aim the fuel spray at the top of the piston to avoid wetting the cylinder liner with liquid fuel to minimize emissions of unburned hydro-

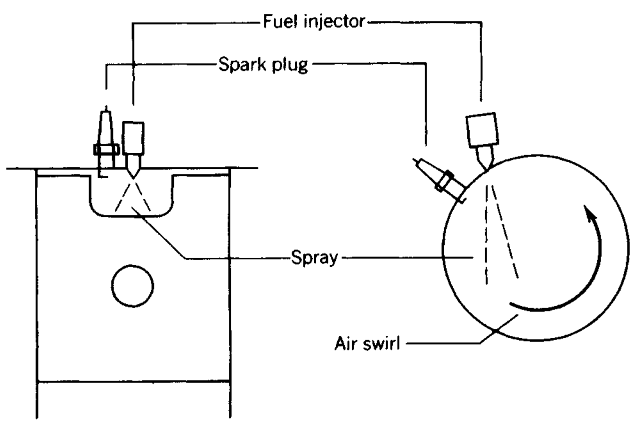

Fig. 59.5 Schematic of DISC SI engine combustion chambers.

carbons. The shape of the piston, together with the air motion and ignition location, ensure that there is still an ignitable mixture in the vicinity of the spark plug even though the overall mixture is extremely lean. However, the extremely lean operation results in a low power capability. Thus, at high loads this version of the direct injection engine uses early injection timing, as discussed below.

Early injection results in sufficient time available for the mixture to become essentially completely mixed before ignition, given sufficient turbulence to aid the mixing process. A stoichiometric or slightly rich mixture is used to provide maximum power output and also ensures that ignition is not a difficulty.

59.1.2 Compression Ignition (Diesel) Engines

CI engines induct only air during the intake process. Late in the compression process, fuel is injected directly into the combustion chamber and mixes with the air that has been compressed to a relatively high temperature. The high temperature of the air serves to ignite the fuel. Like the DISC SI engine, the power output of the diesel is controlled by controlling the fuel flow rate while the volumetric efficiency is approximately constant. Although the fuel–air ratio is variable, the diesel always operates overall fuel-lean, with a maximum allowable fuel–air ratio limited by the production of unacceptable levels of smoke (also called soot or particulates). Diesel engines are inherently stratified because of the nature of the fuel-injection process. The fuel–air mixture is fuel-rich near the center of the fuel-injection cone and fuel-lean in areas of the combustion chamber that are farther from the fuel injection cone. Unlike the combustion process in the SI engine, which occurs at almost constant volume, the combustion process in the diesel engine ideally occurs at constant pressure. That is, the combustion process in the CI engine is relatively slow, and the fuel–air mixture continues to burn during a significant portion of the expansion stroke (fuel continues to be injected during this portion of the expansion stroke) and the high pressure that would normally result from combustion is relieved as the piston recedes. After the combustion process is completed, the expansion process continues until the piston reaches BDC. The diesel may complete the five general engine processes through either a 2-stroke cycle or a 4-stroke cycle. Furthermore, the diesel may be subclassified as either an indirect injection diesel or a direct injection diesel.

Indirect injection (IDI) or divided chamber diesels are geometrically similar to divided chamber stratified charge SI engines. All IDI diesels operate on a 4-stroke cycle. Fuel is injected into the prechamber and combustion is initiated by autoignition. A glow plug is also located in the prechamber, but is only used to alleviate cold start difficulties. As shown in Fig. 59.4, the IDI may be designed so that the jet of hot gases issuing into the main chamber promotes swirl of the reactants in the main chamber. This configuration is called the *swirl chamber IDI diesel.* If the system is not designed to promote swirl, it is called the *prechamber IDI diesel.* The divided chamber design allows a relatively inexpensive pintle-type fuel injector to be used on the IDI diesel.

Direct injection (DI) or "open" chamber diesels are similar to DISC SI engines. There is no prechamber, and fuel is injected directly into the main chamber. Therefore, the characteristics of the fuel-injection cone have to be tailored carefully for proper combustion, avoidance of knock, and minimum smoke emissions. This requires the use of a high-pressure close-tolerance fuel injection system that is relatively expensive. The DI diesel may operate on either a 4-stroke or a 2-stroke cycle. Unlike the 2-stroke SI engine, the 2-stroke diesel often uses a mechanically driven blower for supercharging rather than crankcase compression and also may use multiple inlet ports in each cylinder, as shown in Fig. 59.6. Also, one or more exhaust valves in the top of the cylinder may be used instead of exhaust ports near the bottom of the cylinder, resulting in "through" or "uniflow" scavenging rather than "loop" or "cross" scavenging.

59.2 FUELS AND KNOCK

Knock is the primary factor that limits the design of most IC engines. Knock is the result of engine design characteristics, engine operating conditions, and fuel properties. The causes of knock are discussed in this section. Fuel characteristics, especially those that affect either knock or performance, are also discussed in this section.

59.2.1 Knock in Spark Ignition Engines

Knock occurs in the SI engine if the fuel–air mixture autoignites too easily. At the end of the compression stroke, the fuel–air mixture exists at a relatively high temperature and pressure, the specific values of which depend primarily on the compression ratio and the intake manifold pressure (which is a function of the load). The spark plug then ignites a flame that travels toward the periphery of the combustion chamber. The increase in temperature and number of moles of the burned gases behind the flame front causes the pressure to rise throughout the combustion chamber. The "end gases" located in the peripheral regions of the combustion chamber (in the "unburned zone") are compressed to even higher temperatures by this increase in pressure. The high temperature of the end gases can lead to a sequence of chemical reactions that are called *autoignition.* If the autoignition

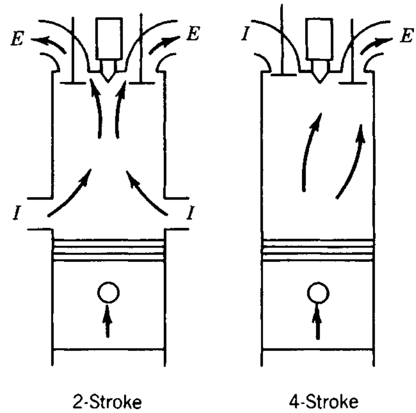

Fig. 59.6 Schematic of 2-stroke and 4-stroke DI diesels. The 2-stroke incorporates "uniflow" scavenging.

reactions have sufficient time at a sufficiently high temperature, the reaction sequence can produce strongly exothermic reactions such that the temperature in the unburned zone may increase at a rate of several million K/sec, which results in knock. That is, if the reactive end gases remain at a high temperature for a sufficient period of time (i.e., longer than the "ignition delay time"), then the autoignition reactions will produce knock. Normal combustion occurs if the flame front passes through the end gases before the autoignition reactions reach a strongly exothermic stage.

For most fuels, autoignition is characterized by three stages that are dictated by the unburned mixture (or end gas) temperature. Here, it is important to note that the temperature varies with crank angle due to compression by the piston motion and, after ignition, due to compression by the expanding flame front, and the entire temperature history shifts up or down due to the effects of load, ambient air temperature, etc. At "low" temperatures, the reactivity of the end gases increases with increasing temperature. As the temperature increases further, the rate of increase of the reactivity either slows markedly or even decreases (the so-called "negative temperature coefficient" regime). When the temperature increases to even higher values (typically, above ~900 K), the reactivity begins to increase extremely strongly, the autoignition reactions reach an energy liberating stage, enough energy may be released during this stage to initiate a "high" temperature (>1000 K) chemical mechanism,[1,2] and a runaway reaction occurs. If the rate of energy release is greater than the rate of expansion, then a strong pressure gradient will result. The steep pressure wave thus established will travel throughout the combustion chamber, reflect off the walls, and oscillate at the natural frequency characteristic of the combustion chamber geometry. This acoustic vibration results in an audible sound called *knock.* It should be noted that the flame speeds associated with knock are generally considered to be lower than the flame speeds associated with detonation (or explosion).[3,4] Nevertheless, the terms *knock* and *detonation* are often used interchangeably in reference to end gas autoignition.

The tendency of the SI engine to knock will be affected by any factors that affect the temperature and pressure of the end gases, the ignition delay time, the end gas residence time (before the normal flame passes through the end gases), and the reactivity of the mixture. The flame speed is a function of the turbulence intensity in the combustion chamber, and the turbulence intensity increases with increasing engine speed. Thus, the end gases will have a shorter residence time at high engine speed and there will be a decreased tendency to knock. As the load on the engine increases, the throttle plate is opened wider and the pressure in the intake manifold increases, thereby increasing the end gas pressure (and, thereby, temperature), resulting in a greater tendency to knock. Thus, knock is most likely to be observed for SI engines used in motor vehicles at conditions of high load and low engine speed, such as acceleration from a standing start.

Other factors that increase the knock tendency of an SI engine[1–5] include increased compression ratio, increased inlet air temperature, increased distance between the spark plug and the end gases, location of the hot exhaust valve near the region of the end gases that is farthest from the spark plug, and increased intake manifold temperature and pressure due to pressure boosting (supercharging or turbocharging). Factors that decrease the knock tendency of an SI engine[1–5] include retarding the spark timing, operation with either rich or lean mixtures (and thus the ability to operate the DISC SI engine at a higher compression ratio, since the end gases for this engine are extremely lean and therefore not very reactive), and increased inert levels in the mixture (via exhaust gas recirculation, water injection, etc.). The fuel characteristics that affect knock are quantified using octane rating tests, which are discussed in more detail in Section 59.2.3. A fuel with higher octane number has a decreased tendency to knock.

59.2.2 Knock in the Diesel Engine

Knock occurs in the diesel engine if the fuel–air mixture does not autoignite easily enough. Knock occurs at the beginning of the combustion process in a diesel engine, whereas it occurs near the end of the combustion process in an SI engine. After the fuel injection process begins, there is an ignition delay time before the combustion process is initiated. This ignition delay time is not caused solely by the chemical delay that is critical to autoignition in the SI engine, but is also due to a physical delay. The physical delay results from the need to vaporize and mix the fuel with the air to form a combustible mixture. If the overall ignition delay time is high, then too much fuel may be injected prior to autoignition. This oversupply of fuel will result in an energy release rate that is too high immediately after ignition occurs. In turn, this will result in an unacceptably high rate of pressure rise and cause the audible sound called *knock*.

The factors that will increase the knock tendency of a diesel engine[1,3,5] are those that decrease the rates of atomization, vaporization, mixing, and reaction, and those that increase the rate of fuel injection. The diesel engine is most prone to knock under cold start conditions because

1. The fuel, air, and combustion chamber walls are initially cold, resulting in high fuel viscosity (poor mixing and therefore a longer physical delay), poor vaporization (longer physical delay), and low initial reaction rates (longer chemical delay).
2. The low engine speed results in low turbulence intensity (poor mixing, yielding a longer physical delay) and may result in low fuel-injection pressures (poor atomization and longer physical delay).
3. The low starting load will lead to low combustion temperatures and thus low reaction rates (longer chemical delay).

After a diesel engine has attained normal operating temperatures, knock will be most liable to occur at high speed and low load (exactly the opposite of the SI engine). The low load results in low combustion temperatures and thus low reaction rates and a longer chemical delay. Since most diesel engines have a gear-driven fuel-injection pump, the increased rate of injection at high speed will more than offset the improved atomization and mixing (shorter physical delay).

Because the diesel knocks for essentially the opposite reasons than the SI engine, the factors that increase the knock tendency of an SI engine will decrease the knock tendency of a diesel engine: increased compression ratio, increased inlet air temperature, increased intake manifold temperature and pressure due to supercharging or turbocharging, and decreased concentrations of inert species. The knock tendency of the diesel engine will be increased if the injection timing is advanced or retarded from the optimum value and if the fuel has a low volatility, a high viscosity, and/or a low "cetane number." The cetane rating test and other fuel characteristics are discussed in more detail in the following section.

59.2.3 Characteristics of Fuels

Several properties are of interest for both SI engine fuels and diesel fuels. Many of these properties are presented in Table 59.1 for the primary reference fuels, for various types of gasolines and diesel fuels, and for the alternative fuels that are of current interest.

The stoichiometry, or relative amount of air and fuel, in the combustion chamber is usually specified by the air–fuel mass ratio (AF), the fuel–air mass ratio (FA = 1/AF), the equivalence ratio (ϕ), or the excess air ratio (λ). Measuring instruments may be used to determine the mass flow rates of air and fuel into an engine so that AF and FA may be easily determined. Alternatively, AF and FA may be calculated if the exhaust product composition is known, using any of several available techniques.[5] The equivalence ratio normalizes the actual fuel–air ratio by the stoichiometric fuel–air ratio (FA_s), where "stoichiometric" refers to the chemically correct mixture with no excess air and no excess fuel. Recognizing that the stoichiometric mixture contains 100% "theoretical air" allows the equivalence ratio to be related to the actual percentage of theoretical air (TA, percentage by volume or mole):

$$\phi = FA/FA_s = AF_s/AF = 100/TA = 1/\lambda \tag{59.1}$$

The equivalence ratio is a convenient parameter because $\phi < 1$ refers to a fuel-lean mixture, $\phi > 1$ to a fuel-rich mixture, and $\phi = 1$ to a stoichiometric mixture.

The stoichiometric fuel–air and air–fuel ratios can be easily calculated from a reaction balance by assuming "complete combustion" [only water vapor (H_2O) and carbon dioxide (CO_2) are formed

Table 59.1 Properties for Various Fuels

Name	Formula	MW	AF_s	LHV_p	h_f[a]	h_v*	sg[b]	RON	MON	Ref.
Primary Reference Fuels										
Iso-octane	C_8H_{18}	114	15.1	44.6	−224.3[c]	35.1[c]	0.69	100	100	5
Normal heptane	C_7H_{16}	100	15.1	44.9	−187.9[c]	36.6[c]	0.68	0	0	5
Normal hexadecane	$C_{16}H_{34}$	226	14.9	44.1	−418.3[d]	50.9[e]	0.77	—	0[f]	5
Alternative Fuels										
Average CNG	$CH_{3.88}$	17.4	16.3	47.9	−79.6	—	0.60[b]	> 120	> 120	26, 27
LPG as propane[(1)]	C_3H_8	44	15.6	46.3	−103.9	15.1	0.50	112	97	5, 22
Methanol	CH_3OH	32	6.4	21.2	−201.3[g]	37.5	0.79	112	91	5, 21
Ethanol	C_2H_5OH	46	9.0	27.8	−235.5[g]	42.4	0.78	111	92	5, 21
*Gasolines***										
1988 U.S. avg.	$C_8H_{14.53}$	111	14.5	42.6	−189.9[h]	NA[j]	0.75	92.0	82.6	28
Certification[(2)]	$C_8H_{14.69}$	111	14.5	42.6	−202.4[h]	NA	0.74	96.7	87.5	28
Cal. Phase 2 RFG[(3)]	$C_8H_{16.28}O_{0.24}$	116	14.2	41.6	−273.3[h]	NA	0.74	NA	NA	29
Aviation	C_8H_{17}	113	14.9	41.9–43.1	−390.3[i]	NA	0.72	NA	NA	3
Diesel Fuels										
Automotive	$C_{12}H_{23.7}$	168	14.7	40.6–44.4	−445.0[i]	90.1–131.7	0.81–0.85	—	—	3
No. 1D	$C_{12}H_{26}$	170	15.0	42.4	−596.0	45.4	0.88	—	—	1
No. 2D	$C_{13}H_{28}$	184	15.0	41.8	−747.6	44.9	0.92	—	—	1
No. 4D	$C_{14}H_{30}$	198	15.0	41.3	−894.6	46.0	0.96	—	—	1

[a] Of vapor phase fuel at 298 K in MJ/kmole, except when noted otherwise.

[b] sg is ρ_F at 20°C/ρ_w at 4°C (1000 kg/m^3), except values from Ref. 1 (reference temp. is 15°C) and CNG (referenced to air).

[c] From Ref. 23.

[d] Calculated.

[e] At 1 atm and boiling temperature.

[f] Estimate from Ref. 1, p. 147.

[g] Reference 22.

[h] Enthalpy of formation is for the liquid fuel (rather than the gaseous fuel), as calculated from fuel properties.

[i] Enthalpy of formation is for the liquid fuel as calculated from the average heating value.

[j] Typically 35–40.

*At 298 K and corresponding saturation pressure, except when noted otherwise.

**As C8.

(1) Liquefied petroleum gas has a variable composition, normally dominated by propane.

(2) The properties of emissions certification gasoline vary somewhat.

(3) Properties of California Phase 2 Reformulated Gasoline from a sample of Arco EC-X.

during the combustion process], even though the actual combustion process will almost never be complete. The reaction balance for the complete combustion of a stoichiometric mixture of air with a fuel of the atomic composition C_xH_y is

$$C_xH_y + (x + .25y)O_2 + 3.764(x + .25y)N_2 = xCO_2 + 0.5yH_2O + 3.764(x + .25y)N_2 \quad (59.2)$$

where air is taken to be 79% by volume "effective nitrogen" (N_2 plus the minor components in air) and 21% by volume oxygen (O_2) and thus the nitrogen-to-oxygen ratio of air is 0.79/0.21 = 3.764. Given that the molecular weight (MW) of air is 28.967, the MW of carbon (C) is 12.011, and the MW of hydrogen (H) is 1.008, then AF_s and FA_s for any hydrocarbon fuel may be calculated from

$$AF_s = 1/FA_s = (x + 0.25y) \times 4.764 \times 28.967/(12.011x + 1.008y) \quad (59.3)$$

The stoichiometric air–fuel ratios for a number of fuels of interest are presented in Table 59.1.

The energy content of the fuel is most often specified using the constant-pressure lower heating value (LHV_p). The lower heating value is the maximum energy that can be released during combustion of the fuel if (1) the water in the products remains in the vapor phase, (2) the products are returned to the initial reference temperature of the reactants (298 K), and (3) the combustion process is carried out such that essentially complete combustion is attained. If the water in the products is condensed, then the higher heating value (HHV) is obtained. If the combustion system is a flow calorimeter, then the constant-pressure heating value is measured (and, most usually, this is HHV_p). If the combustion system is a bomb calorimeter, then the constant-volume heating value is measured (usually HHV_v). The constant-pressure heating value is the negative of the standard enthalpy of reaction (ΔH_R^{298}, also known as the heat of combustion) and ΔH_R^{298} is a function of the standard enthalpies of formation h_f^{298} of the reactant and product species. For a fuel of composition C_xH_y, Eq. (59.4) may be used to calculate the constant-pressure heating value (HV_p), given the enthalpy of formation of the fuel, or may be used to calculate h_f^{298} of the fuel, given HV_p:

$$HV_p = -\Delta H_R^{298} = \frac{(h_{f,C_xH_y}^{298} - \beta h_{v,C_xH_y}) + 393.418x + 0.5y(241.763 + 43.998\alpha)}{12.011x + 1.008y} \quad (59.4)$$

In Eq. (59.4): (1) $\alpha = 0$ if the water in the products is not condensed (yielding LHV_p) and $\alpha = 1$ if the water is condensed (yielding HHV_p); (2) $\beta = 0$ if the fuel is initially a vapor and $\beta = 1$ if the fuel is initially a liquid; (3) h_{v,C_xH_y} is the enthalpy of vaporization per kmole* of fuel at 298 K; (4) the standard enthalpies of formation are CO_2: −393.418 MJ/kmole, H_2O: −241.763 MJ/kmole, O_2: 0 MJ/kmole, N_2: 0 MJ/kmole; (5) the enthalpy of vaporization of H_2O at 298 K is 43.998 MJ/kmole; and (6) the denominator is simply the molecular weight of the fuel yielding the heating value in MJ/kg of fuel. Also, the relationship between the constant volume heating value (HV_v) and HV_p for a fuel C_xH_y is

$$HV_v = HV_p + \frac{2.478(0.25y - 1)}{12.011x + 1.008y} \quad (59.5)$$

Of the several heating values that may be defined, LHV_p is preferred for engine calculations since condensation of water in the combustion chamber is definitely to be avoided and since an engine is essentially a steady-flow device and thus the enthalpy is the relevant thermodynamic property (rather than the internal energy). For diesel fuels, HHV_v may be estimated from nomographs, given the density and the "mid-boiling point temperature" or given the "aniline point," the density, and the sulfur content of the fuel.[6]

Values for LHV_p, h_f^{298}, and h_v^{298} for various fuels of interest are presented in Table 59.1.

The specific gravity of a liquid fuel (sg_F) is the ratio of its density (ρ_F, usually at either 20°C or 60°F) to the density of water (ρ_w, usually at 4°C):

$$sg_F = \rho_F/\rho_w \qquad \rho_F = sg_F \cdot \rho_w \quad (59.6)$$

For gaseous fuels, such as natural gas, the specific gravity is referenced to air at standard conditions rather than to water. The specific gravity of a liquid fuel can be easily calculated from a simple measurement of the American Petroleum Institute gravity (API):

$$sg_F = 141.5/(API + 131.5) \quad (59.7)$$

*A kmole is a mole based on a kg, also referred to as a kg-mole.

Tables are available (SAE Standard J1082 SEP80) to correct for the effects of temperature if the fuel is not at the prescribed temperature when the measurement is performed. Values of sg_F for various fuels are presented in Table 59.1.

The knock tendency of SI engine fuels is rated using an octane number (ON) scale. A higher octane number indicates a higher resistance to knock. Two different octane-rating tests are currently used. Both use a single-cylinder variable-compression-ratio SI engine for which all operating conditions are specified (see Table 59.2). The fuel to be tested is run in the engine and the compression ratio is increased until knock of a specified intensity (standard knock) is obtained. Blends of two primary reference fuels are then tested at the same compression ratio until the mixture is found that produces standard knock. The two primary reference fuels are 2,2,4-trimethyl pentane (also called iso-octane), which is arbitrarily assigned an ON of 100, and n-heptane, which is arbitrarily assigned an ON of 0. The ON of the test fuel is then simply equal to the percentage of iso-octane in the blend that produced the same knock intensity at the same compression ratio. However, if the test fuel has an ON above 100, then iso-octane is blended with tetraethyl lead instead of n-heptane. After the knock tests are completed, the ON is then computed from

$$\mathrm{ON} = 100 + \frac{28.28T}{1 + 0.736T + (1 + 1.472T - 0.035216T^2)^{1/2}} \tag{59.8}$$

where T is the number of milliliters of tetraethyl lead per U.S. gallon of iso-octane. The two different octane rating tests are called the Motor method (American Society of Testing and Materials, ASTM Standard D2700-82) and the Research method (ASTM D2600-82), and thus a given fuel (except these two primary reference fuels) will have two different octane numbers: a Motor octane number (MON) and a Research octane number (RON). The Motor method produces the lowest octane numbers, primarily because of the high intake manifold temperature for this technique, and thus the Motor method is said to be a more severe test for knock. The "sensitivity" of a fuel is defined as the RON minus the MON of that fuel. The "antiknock index" is the octane rating posted on gasoline pumps at service stations in the United States and is simply the average of RON and MON. Octane numbers for various fuels of interest are presented in Table 59.1.

The standard rating test for the knock tendency of diesel fuels (ASTM D613-82) produces the cetane number (CN). Because SI and diesel engines knock for essentially opposite reasons, a fuel with a high ON will have a low CN and therefore would be a poor diesel fuel. A single-cylinder variable-compression-ratio CI engine is used to measure the CN, and all engine operating conditions are specified. The compression ratio is increased until the test fuel exhibits an ignition delay of 13°. Here, it should be noted that ignition delay rather than knock intensity is measured for the CN technique. A blend of two primary reference fuels (n-hexadecane, which is also called n-cetane: CN = 100; and heptamethyl nonane, or i-cetane: CN = 15) are then run in the engine and the compression ratio is varied until a 13° ignition delay is obtained. The CN of this blend is given by

$$\mathrm{CN} = \%\ n\text{-cetane} + 0.15 \times (\%\ \text{heptamethyl nonane}) \tag{59.9}$$

Various blends are tried until compression ratios are found that bracket the compression ratio of the test fuel. The CN is then obtained from a standard chart. General specifications for diesel fuels are presented in Table 59.3 along with characteristics of "average" diesel fuels for light duty vehicles.

Many other thermochemical properties of fuels may be of interest, such as vapor pressure, volatility, viscosity, cloud point, aniline point, mid-boiling-point temperature, and additives. Discussion of these characteristics is beyond the scope of this chapter but is available in the literature.[1,4–7]

Table 59.2 Test Specifications That Differ for Research and Motor Method Octane Tests—ASTM D2699-82 and D2700-82

Operating Condition	RON	MON
Engine speed (rpm)	600	900
Inlet air temperature (°C)	^{a}	38°C
FA mixture temperature (°C)	^{b}	149°C
Spark advance	13°BTDC	^{c}

^{a}Varies with barometric pressure.
^{b}No control of fuel–air mixture temperature.
^{c}Varies with compression ratio.

Table 59.3 Diesel Fuel Oil Specifications—ASTM D975-81

Property	Units	Fuel Type 1D[b]	2D[c]	4D[d]
Minimum flash point	°C	38	52	55
Maximum H_2O and sediment	Vol. %	0.05	0.05	0.50
Maximum carbon residue	%	0.15	0.35	—
Maximum ash	Wt. %	0.01	0.01	0.10
90% distillation temperature, min/max	°C	—/288	282/338	—/—
Kinematic viscosity,[a] min/max	mm^2/sec	1.3/2.4	1.9/4.1	5.5/24.0
Maximum sulfur	Wt. %	0.5	0.5	2.0
Maximum Cu strip corrosion	—	No. 3	No. 3	—
Minimum cetane number	—	40	40	30

[a] At 40°C.

[b] Preferred for high-speed diesels, especially for winter use, rarely available. 1976 U.S. average properties[17,24]: API, 42.2; 220°C midboiling point; 0.081 wt. % sulfur; and cetane index[e] of 49.5.

[c] For high-speed diesels (passenger cars and trucks) 1972 U.S. average properties[17,24]: API = 35.7; MBP = 261°C; 0.253 wt. % sulfur; cetane index[e] = 48.4.

[d] Low- and medium-speed diesels.

[e] The cetane index is an approximation of the CN, calculated from ASTM D976-80 given the API and the midpoint temperature, and accurate within ± 2 CN for 30 ≤ CN ≤ 60 for 75% of distillate fuels tested.

59.3 PERFORMANCE AND EFFICIENCY

The performance of an engine is generally specified through the brake power (bp), the torque (τ), or the brake mean effective pressure (bmep), while the efficiency of an engine is usually specified through the brake specific fuel consumption (bsfc) or the overall efficiency (η_e). Experimental and theoretical determination of important engine parameters is discussed in the following sections.

59.3.1 Experimental Measurements

Engine dynamometer (dyno) measurements can be used to obtain the various engine parameters using the relationships[5,8,9]

$$\text{bp} = LRN/9549.3 = LN/K \tag{59.10}$$

$$\tau = LR = 9549.3\ \text{bp}/N \tag{59.11}$$

$$\text{bmep} = 60{,}000\ \text{bp}\ X/DN \tag{59.12}$$

$$\text{bsfc} = \dot{m}_F/\text{bp} \tag{59.13}$$

$$\eta_e = \frac{3600\ \text{bp}}{\dot{m}_F \text{LHV}_p} = \frac{3600}{\text{bsfc LHV}_p} \tag{59.14}$$

Definitions and standard units* for the variables in the above equations are presented in the symbols list. The constants in the above equations are simply unit conversion factors. The brake power is the useful power measured at the engine output shaft. Some power is used to overcome frictional losses in the engine and this power (the friction power, fp) is not available at the output shaft. The total rate of energy production within the engine is called the *indicated power* (ip)

$$\text{ip} = \text{bp} + \text{fp} \tag{59.15}$$

where the friction power can be determined from dyno measurements using:

$$\text{fp} = FRN/9549.3 = FN/K \tag{59.16}$$

*Standard units are not in strict compliance with the International System of units, in order to produce numbers of convenient magnitude.

The efficiency of overcoming frictional losses in the engine is called the *mechanical efficiency* (η_M), which is defined as

$$\eta_M = \text{bp}/\text{ip} = 1 - \text{fp}/\text{ip} \tag{59.17}$$

The definitions of ip and η_M allow determination of the indicated mean effective pressure (imep) and the indicated specific fuel consumption (isfc):

$$\text{imep} = \text{bmep}/\eta_M = 60{,}000 \text{ ip } X/DN \tag{59.18}$$

$$\text{isfc} = \eta_M \text{bsfc} = \dot{m}_F/\text{ip} \tag{59.19}$$

Three additional efficiencies of interest are the volumetric efficiency (η_v), the combustion efficiency (η_c), and the indicated thermal efficiency (η_{ti}).

The volumetric efficiency is the effectiveness of inducting air into the engine[5,10–13] and is defined as the actual mass flow rate of air ($\dot{m}_A$) divided by the theoretical maximum air mass flow rate ($\rho_A DN/X$):

$$\eta_v = \frac{\dot{m}_A}{\rho_A DN/X} \tag{59.20}$$

The combustion efficiency is the efficiency of converting the chemical energy of the fuel to thermal energy (enthalpy) of the products of combustion.[12–14] Thus,

$$\eta_c = -\Delta H^{298}_{R,\text{act}}/\text{LHV}_p \tag{59.21}$$

The actual enthalpy of reaction ($\Delta H^{298}_{R,\text{act}}$) may be determined by measuring the mole fractions in the exhaust of CO_2, CO, O_2, and unburned hydrocarbons (expressed as "equivalent propane" in the following) and calculating the mole fractions of H_2O and H_2 from atom balances. For a fuel of composition C_xH_y,

$$-\Delta H^{298}_{R,\text{act}} = \frac{h^{298}_{f,C_xH_y} + x(Y_{CO_2}393.418 + Y_{H_2O}241.763 + Y_{CO}110.600 + Y_{C_3H_8}103.900)}{(Y_{CO_2} + Y_{CO} + 3Y_{C_3H_8})(12.011x + 1.008y)} \tag{59.22}$$

where Y_i is the mole fraction of species i in the "wet" exhaust. In Eq. (59.22), a carbon balance was used to convert moles of species i per mole of product mixture to moles of species i per mole of fuel burned and the molecular weight of the fuel appears in the denominator to produce the enthalpy of reaction in units of MJ per kg of fuel burned. If a significant amount of soot is present in the exhaust (e.g., a diesel under high load), then the carbon balance becomes inaccurate and an oxygen balance would have to be substituted.

The indicated thermal efficiency is the efficiency of the actual thermodynamic cycle. This parameter is difficult to measure directly, but may be calculated from

$$\eta_{ti} = \frac{3600 \text{ ip}}{\eta_c \dot{m}_F \text{LHV}_p} = \frac{3600}{\text{isfc } \eta_c \text{LHV}_p} \tag{59.23}$$

Because the engine performance depends on the air flow rate through it, the ambient temperature, barometric pressure, and relative humidity can affect the performance parameters and efficiencies. It is often desirable to correct the measured values to standard atmospheric conditions. The use of correction factors is discussed in the literature,[5,8,9] but is beyond the scope of this chapter.

The four fundamental efficiencies η_{ti}, η_c, η_v, and η_M are related to the global performance and efficiency parameters in the following section. Methods for modeling these efficiencies are also discussed in the following section.

59.3.2 Theoretical Considerations and Modeling

A set of *exact* equations relating the fundamental efficiencies to the global engine parameters is[12,13]

$$\text{bp} = \eta_{ti}\eta_c\eta_v\eta_M\, \rho_A\, D\, N\, \text{LHV}_p\text{FA}/(60X) \tag{59.24}$$

$$\text{ip} = \eta_{ti}\eta_c\eta_v\, \rho_A\, D\, N\, \text{LHV}_p\text{FA}/(60X) \tag{59.25}$$

$$\tau = 1000\, \eta_{ti}\eta_c\eta_v\eta_M\, \rho_A\, \text{D LHV}_p\text{FA}/(2\pi X) \tag{59.26}$$

$$\text{bmep} = 1000\, \eta_{ti}\eta_c\eta_v\eta_M\, \rho_A\, \text{LHV}_p\text{FA} \tag{59.27}$$

$$\text{imep} = 1000\, \eta_{ti}\eta_c\eta_v\, \rho_A\, \text{LHV}_p\text{FA} \tag{59.28}$$

$$\text{bsfc} = 3600/(\eta_{ti}\eta_c\eta_M\, \text{LHV}_p) \tag{59.29}$$

$$\text{isfc} = 3600/(\eta_{ti}\eta_c\, \text{LHV}_p) \tag{59.30}$$

$$\eta_e = \eta_{ti}\eta_c\eta_M \tag{59.31}$$

where, again, the constants are simply units conversion factors. Equations (59.24)–(59.31) are of interest because they (1) can be derived solely from physical and thermodynamic considerations,[12] (2) can be used to explain observed engine characteristics,[12] and (3) can be used as a base for modeling engine performance.[13] For example, Eqs. (59.27) and (59.28) demonstrate that the mean effective pressure is useful for comparing different engines because it is a measure of performance that is essentially independent of displacement (D), engine speed (N), and whether the engine is a 2-stroke ($X = 1$) or a 4-stroke ($X = 2$). Similarly, Eqs. (59.24), (59.27), and (59.29) show that a diesel should have less power, lower bmep, and better bsfc than a comparable SI engine because the diesel generally has about the same LHV_p and η_c, higher η_{ti} and η_v, but lower η_M and much lower FA.

The performance of an engine may be theoretically predicted by modeling each of the fundamental efficiencies (η_{ti}, η_c, η_v, and η_M) and then combining these models using Eqs. (59.2)–(59.31) to yield the performance parameters. Simplified models for each of these fundamental efficiencies are discussed below. More detailed engine models are available with varying degrees of sophistication and accuracy,[2,4,11,13,15,16,25] but, because of their length and complexity, are beyond the scope of this chapter.

The combustion efficiency may be most simply modeled by assuming complete combustion. It can be shown[13] that for complete combustion

$$\eta_c = 1.0 \qquad \phi \le 1 \tag{59.32a}$$

$$\eta_c = 1.0/\phi \qquad \phi \ge 1 \tag{59.32b}$$

Equation (59.32) implies that η_c is only dependent on FA. It has been shown that for homogeneous charge 4-stroke SI engines using fuels with a carbon-to-hydrogen ratio similar to that of gasoline, η_c is approximately independent of compression ratio, engine speed, ignition timing, and load. Although no data are available, it is expected that this is also true of 4-stroke stratified charge SI engines and diesel engines (at least up to the point of production of appreciable smoke). Such relationships will be less accurate for 2-stroke SI engines due to fuel short-circuiting. As shown in Fig. 59.7, Eq. (59.32) is accurate within 5–10% for fuel-lean combustion with accuracy decreasing to about 20% for the very fuel-rich equivalence ratio of 1.5 for the 4-stroke SI engine. Also shown in Fig. 59.7 is a quasi equilibrium equation that is slightly more accurate for fuel-lean systems and much more accurate for fuel-rich combustion:

$$\eta_c = 0.959 + 0.129\phi - 0.121\phi^2 \qquad 0.5 \le \phi \le 1.0 \tag{59.33a}$$

$$\eta_c = 2.594 - 2.173\phi + 0.546\phi^2 \qquad 1.0 \le \phi \le 1.5 \tag{59.33b}$$

The indicated thermal efficiency may be most easily modeled using air standard cycles. The values of η_{ti} predicted in this manner will be too high by a factor of about 2 or more, but the trends predicted will be qualitatively correct.

The SI engine ideally operates on the air standard Otto cycle, for which[2,4,5,10,11,14]

$$\eta_{ti} = 1 - (1/\text{CR})^{k-1} \tag{59.34}$$

where CR is the compression ratio and k is the ratio of specific heats of the working fluid. The assumptions of the air standard Otto cycle are (1) no intake or exhaust processes and thus no exhaust residual and no pumping loss, (2) isentropic compression and expansion, (3) constant-volume heat addition and therefore instantaneous combustion, (4) constant-volume heat rejection replacing the exhaust blowdown process, and (5) air is the sole working fluid and is assumed to have a constant value of k. The errors in this model primarily result from failure to account for (1) a working fluid with variable composition and variable specific heats, (2) the finite duration of combustion, (3) heat

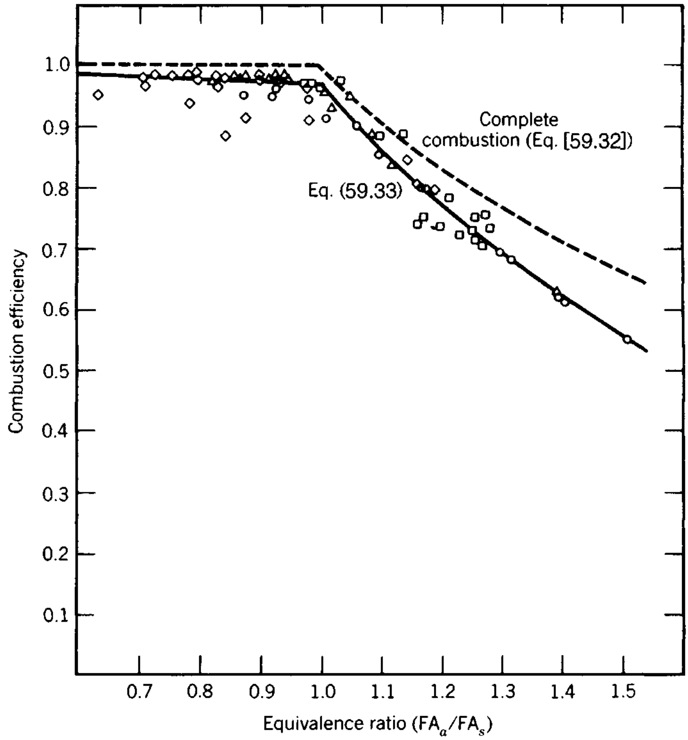

Fig. 59.7 Effect of equivalence ratio (normalized fuel–air ratio) on combustion efficiency for several different 4-stroke SI engines operating on indolene or iso-octane.[13] Model predictions of Eqs. (59.32) and (59.33) also shown.

losses, (4) fluid mechanics (especially as affecting flow past the intake and exhaust valves and the effects of these on the exhaust residual fraction), and (5) pumping losses. The air standard Otto cycle P–v and T–s diagrams are shown in Fig. 59.8. Equation (59.34) indicates that η_{ti} for the SI engine is only a function of CR. While this is not strictly correct, it has been shown[13] that, for the homogeneous charge 4-stroke SI engine, η_{ti} is not as strongly dependent on equivalence ratio, speed, and load as on CR. The predicted effect of CR on η_{ti} is compared with engine data in Fig. 59.9, showing that the theoretical trend is qualitatively correct. Thus, dividing the result of Eq. (59.34) by 2 will yield a reasonable estimate of the indicated thermal efficiency but will not reflect the effects of speed, load, spark timing, valve timing, or other engine design and operating conditions on η_{ti}.

The traditional simplified model for η_{ti} of the CI engine is the air standard Diesel cycle, but the air standard dual cycle is more representative of most modern diesel engines. Figure 59.8 compares the P–v and T–s diagrams for the air standard diesel, dual, and Otto cycles. For the air standard diesel cycle, it can be shown that [2,4,5,11,14]

$$\eta_{ti} = 1 - (1/\mathrm{CR})^{k-1} \frac{r_T^k - 1}{k(r_T - 1)} \tag{59.35}$$

where r_T is the ratio of the temperature at the end of combustion to the temperature at the beginning of combustion, and is thus a measure of the load. Assumptions for this cycle are (1) no intake or exhaust processes, (2) isentropic compression and expansion, (3) constant-pressure heat addition (combustion), (4) constant-volume heat rejection, and (5) air is the sole working fluid and has constant k. For the air standard dual cycle, it can be shown that[2,4,5,10,14]

$$\eta_{ti} = 1 - (1/\mathrm{CR})^{k-1} \frac{r_P r_v^k - 1}{(r_P - 1) + k r_P (r_v - 1)} \tag{59.36}$$

where r_P is the ratio of the maximum pressure to the pressure at the beginning of the combustion process and r_v is the ratio of the volume at the end of the combustion process to the volume at the

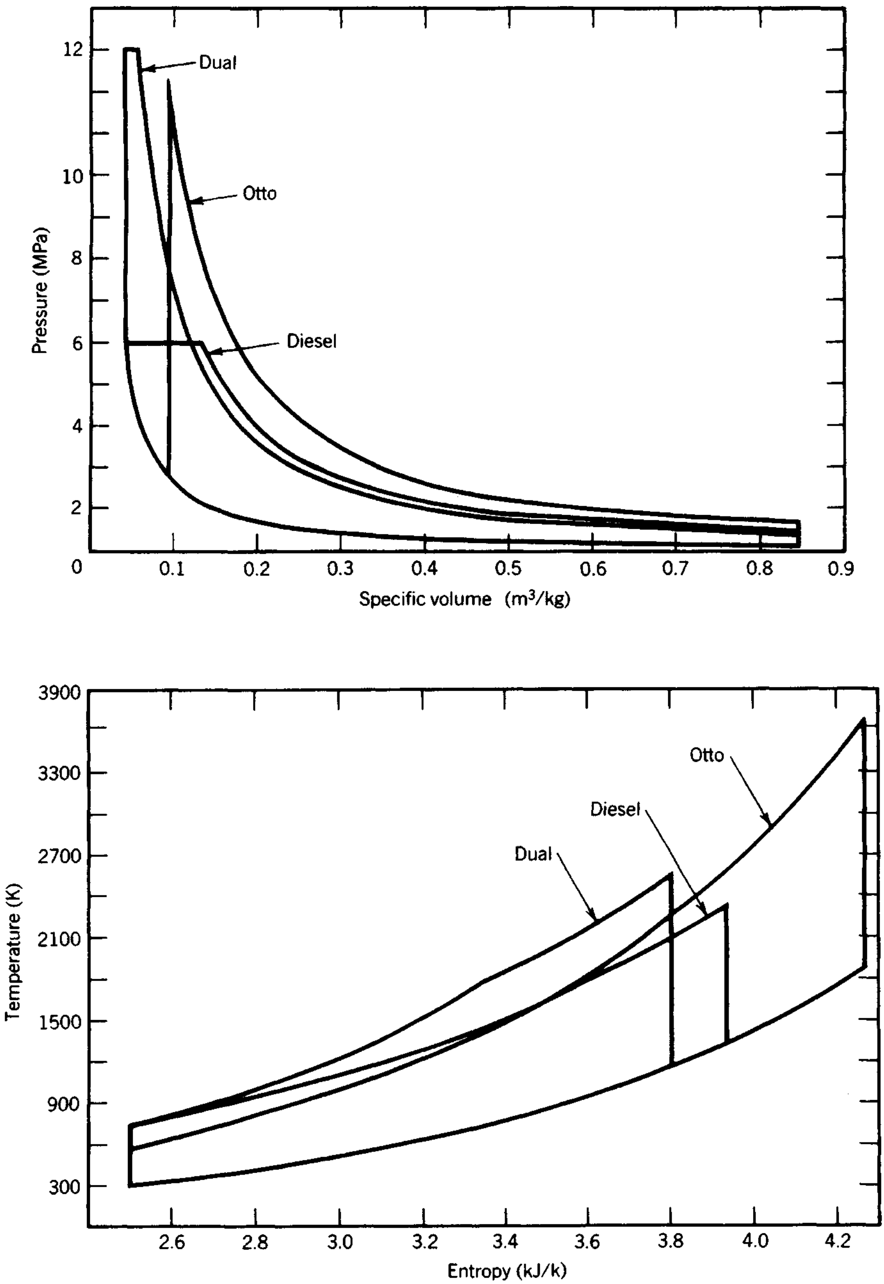

Fig. 59.8 Comparison of air standard Otto, diesel, and dual cycle P–v and T–s diagrams with $k = 1.3$. Otto with CR = 9:1, $\phi = 1.0$, C_8H_{18}. Diesel with CR = 20:1, $\phi = 0.7$, $C_{12}H_{26}$. Dual at same conditions as diesel, but with 50% of heat added at constant volume.

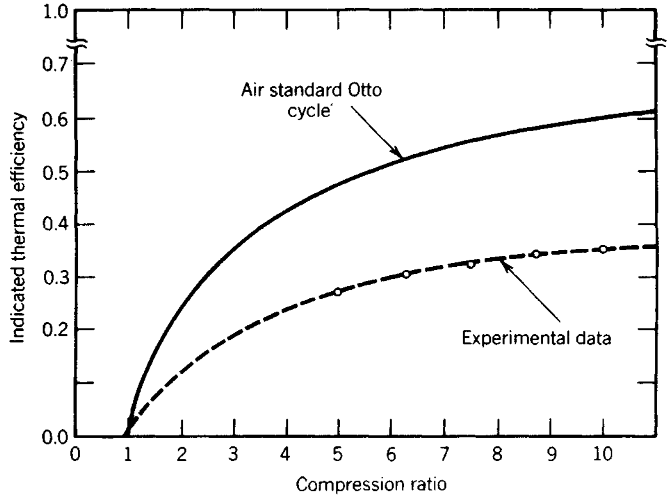

Fig. 59.9 Effect of compression ratio on indicated thermal efficiency of SI engine. Dashed line—experimental data for single cylinder 4-stroke SI engine.[13] Solid line—prediction of air standard Otto cycle [Eq. (59.34)].

beginning of the combustion process (TDC). The assumptions are the same as those for the air standard diesel cycle except that combustion is assumed to occur initially at constant volume with the remainder of the combustion process occurring at constant pressure. Equations (59.35) and (59.36) indicate that the η_{ti} of the diesel is a function of both compression ratio and load, predictions that are qualitatively correct.

The volumetric efficiency is the sole efficiency for which a simplified model cannot be developed from thermophysical principles without the need for iterative calculations. Factors affecting η_v include heat transfer, fluid mechanics (including intake and/or exhaust tuning and valve timing), and exhaust residual fraction. In fact, a 4-stroke engine with tuned intake (exhaust tuning is more important for the 2-stroke) and/or pressure boosting may have greater than 100% volumetric efficiency since Eq. (59.20) is referenced to the air inlet density rather than to the density of the air in the intake manifold. However, for unboosted engines, observed engine characteristics allow values to be estimated for η_v (for subsequent use in Eqs. (59.24)–(59.31)). For the unthrottled DISC SI engine, the diesel, and the SI engine at wide open throttle (full load), η_v is approximately independent of operating conditions other than engine speed (which is important due to valve timing and tuning effects). A peak value for η_v of 0.7–1.0 may be assumed for 4-stroke engines with untuned intake systems, recognizing that there is no justification for choosing any particular number unless engine data for that specific engine are available. For 4-stroke engines with tuned intake systems, a peak value of ~1.15–1.2 might be assumed. Because most SI engines control power output by varying η_v, the effect of load on η_v must be taken into account for this type of engine. Fortunately, other engine operating conditions have much less effect on part load η_v than does the load,[13] so that only load need be considered for this simplified approach. As shown in Fig. 59.10 for the 4-stroke SI engine, η_v is linearly related to load (imep). Because choked flow is attained at no load, a value for η_v at zero load at roughly one-half that assumed at full load may be used and a linear relationship between η_v and load (or intake manifold pressure) may then be used (the intake manifold pressure may drop well below that for choked flow during idle and deceleration, but the flow is still choked).

The mechanical efficiency may be most simply modeled by first determining η_{ti}, η_c, and η_v and then calculating the indicated power using Eq. (59.25). The friction power may then be calculated from an empirical relationship,[4] which has been shown to be reasonably accurate for a variety of production multicylinder SI engines (and it may apply reasonably well to diesels):

$$\text{fp} = 1.975 \times 10^{-9}\, D\, S\, \text{CR}^{1/2} N^2 \tag{59.37}$$

where S is the stroke in mm. It should be noted that oil viscosity (and therefore oil and coolant temperatures) can significantly affect fp, and that Eq. (59.37) applies for normal operating temperatures. Given fp and ip, η_M may be calculated using Eq. (59.17). Since load strongly affects ip and speed strongly affects fp, then η_M is more strongly dependent on speed and load than on other

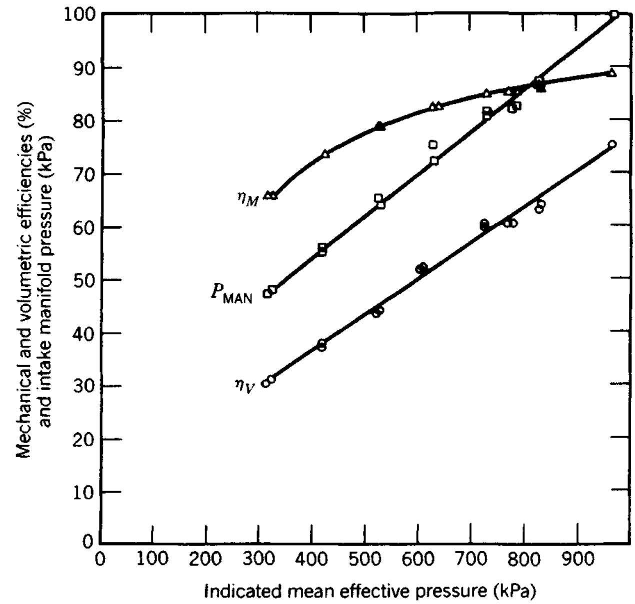

Fig. 59.10 Effect of load (imep) on volumetric efficiency, mechanical efficiency, and intake manifold absolute pressure for 4-stroke V6 SI engine at 2000 rpm.[13]

operating conditions. Figures 59.10 and 59.11 demonstrate the effects of speed and load on η_M for a 4-stroke SI engine. Similar trends would be observed for other types of engines.

The accuracy of the performance predictions discussed above depends on the accuracies of the prediction of η_{ti}, η_c, η_v, and fp. Of these, the models for η_c and fp are generally acceptable. Thus, the primary sources for error are the models of η_{ti} and η_v. As stated earlier, highly accurate models are available if more than qualitative performance predictions are needed.

59.3.3 Engine Comparisons

Table 59.4 presents comparative data for various types of engines.

One of the primary advantages of the SI engine is the wide speed range attainable owing to the short ignition delay, high flame speed, and low rotational mass. This results in high specific weight (bp/W) and high power per unit displacement (bp/D). Additionally, the energy content (LHV$_p$) of gasoline is about 3% greater than that of diesel fuel (on a mass basis; diesel fuel has ~13% more energy per gallon than gasoline). This aids both bmep and bp. The low CR results in relatively high η_M at full load and low engine weight. The low weight, combined with the relative mechanical simplicity (especially of the fueling system), results in low initial cost. The high full load η_M and the high FA (relative to the diesel) result in high bmep and bp. However, the low CR results in low η_{ti} (especially at part load), which in turn causes low η_e, high part load bsfc, and poor low speed τ. Because η_{ti} increases with load, reasonable bsfc is attained near full load. Also, since the product of η_{ti} and N initially increases faster with engine speed than η_M decreases, then good medium speed torque is attained and can be significantly augmented by intake tuning such that η_v peaks at medium engine speed. Another disadvantage of the SI engine is that the near stoichiometric FA used results in high engine-out emissions of the gaseous regulated exhaust pollutants: NO_x (oxides of nitrogen), CO (carbon monoxide), and HCs (unburned hydrocarbons). On the other hand, the nature of the premixed combustion process results in particulate emission levels that are almost too low to measure.

One of the primary advantages of the diesel is that the high CR results in high η_{ti} and thus good full load bsfc and, because of the relationship between η_{ti} and load for the diesel, much higher part load bsfc than the SI engine. The higher η_{ti} and the high η_v produce high η_e and good low speed τ. The low volatility of diesel fuel results in lower evaporative emissions and a lower risk of accidental fire. The high CR also results in relatively low η_M and high engine weight. The high W and the mechanical sophistication (especially of the fuel-injection system) lead to high initial cost. The low η_M, coupled with the somewhat lower LHV$_p$ and the much lower FA in comparison to the SI engine, yield lower bmep. The low bmep and limited range of engine speeds yield low bp and low bp/D. The low bp and high W produce low bp/W. The diesel is ideally suited to supercharging, since

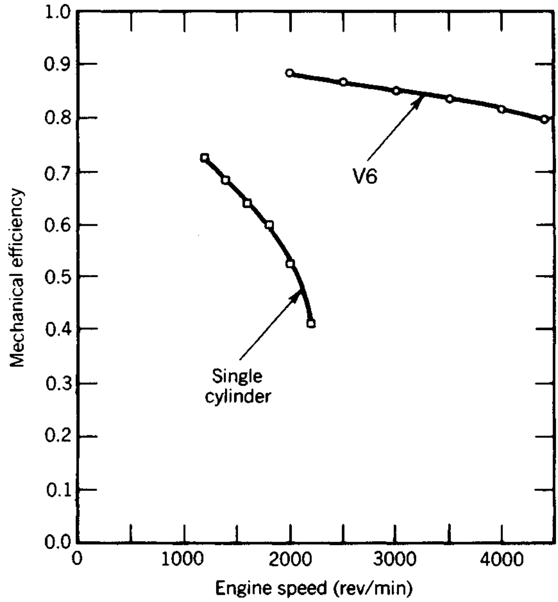

Fig. 59.11 Effect of engine speed on mechanical efficiency of 4-stroke SI V6 engine[13] and of 4-stroke SI single-cylinder research engine.

Table 59.4 Comparative Data for Motor Vehicle Engines[g]

Engine Type	CR	Peak N (rpm)	bmep (kPa)	bp/D (kW/liter)	bp/W (kW/kg)	bsfc (g/kWhr)	η_e (%)
Spark Ignition							
Motorcycles							
2-stroke	6.5–11	4500–7000[e]	400–550	20–50	0.17–0.40	600–400	14–18
4-stroke	6–10	5000–7500[e]	700–1000	30–60	0.18–0.40	340–270	25–31
Passenger cars							
2-stroke	6–8	4500–5400	450–600	30–45	0.18–0.40	480–340	17–18
4-stroke	7–11[c]	4000–7500[f]	700–1000	20–50	0.25–0.50	380–300	20–28
Rotary	8–9	6000–8000	950–1050	35–45	0.62–1.1	380–300	22–27
Trucks (4-stroke)	7–8	3600–5000	650–700	25–30	0.15–0.40	400–300	16–27
Diesel							
Passenger cars[a]	12–23[d]	4000–5000	500–750	18–22	0.20–0.40	340–240	23–28
Trucks—NA[b]	16–22	2100–4000	600–900	15–22	0.14–0.25	245–220	23–33
Trucks—TC[b]	15–22	2100–3200	1200–1800	18–26	0.14–0.29	245–220	—

[a]Exclusively IDI in United States. 2-stroke and 4-stroke DI may be used in other countries.
[b]NA: naturally aspirated, TC: turbocharged. Trucks are primarily DI.
[c]U.S. average about 9.
[d]U.S. average about 22.
[e]Many modern motorcycle engines have peak speeds exceeding 10,000 rpm.
[f]Production engines capable of exceeding 6000 rpm are rare.
[g]Adopted with permission from Adler and Bazlen.[3] Design trends to be read left to right.

supercharging inhibits knock. However, the lower exhaust temperatures of the diesel (500–600°C)[3] means that there is less energy available in diesel exhaust and thus it is somewhat more difficult to turbocharge the diesel. The diffusion-flame nature of the combustion process produces high particulate and NO_x levels. The overall fuel-lean nature of the combustion process produces relatively low levels of HCs and very low levels of CO. Diesel engines are much noisier than SI engines.

The IDI diesel is used in all diesel-powered passenger cars sold in the United States, which are attractive because of the high bsfc of the diesel in comparison to the SI engine. The IDI diesel passenger car has approximately 35% better fuel economy on a kilometers per liter of fuel basis (19% better on a per unit energy basis, since the density of diesel fuel is about 16% higher than that of gasoline) than the comparable SI-engine-powered passenger car.[17] The IDI diesel is currently used in passenger cars rather than the DI diesel because of the more limited engine speed range of the DI diesel and because the necessarily more sophisticated fuel-injection system leads to higher initial cost. Also, the IDI diesel is less fuel sensitive (less prone to knock), is generally smaller, runs more quietly, and has fewer odorous emissions than the DI diesel.[17]

The engine speed range of DI diesels is generally more limited than that of the IDI diesel. The DI diesel is easier to start and rejects less heat to the coolant. The somewhat higher exhaust temperature makes the DI diesel more suitable for turbocharging. At full load, the DI emits more smoke and is noisier than the IDI diesel. In comparison to the IDI diesel, the DI diesel has 15–20% better fuel economy.[17] This is a result of higher η_{ti} (which yields better bsfc and η_e) due to less heat loss because of the lower surface-to-volume ratio of the combustion chamber in absence of the IDI's prechamber.

The primary advantage of the 2-stroke engine is that it has one power stroke per revolution (X = 1) rather than one per every two revolutions (X = 2). This results in high τ, bp, and bp/W. However, the scavenging process results in very high HC emissions, and thus low η_c, bmep, bsfc, and η_e. When crankcase supercharging is used to improve scavenging, η_v is low,[3] being in the range of 0.3–0.7. Blowers may be used to improve scavenging and η_v, but result in decreased η_M. The mechanical simplicity of the valveless crankcase compression design results in high η_M, low weight, and low initial cost. Air-cooled designs have an even lower weight and initial cost, but have limited CR because of engine-cooling considerations. In fact, both air-cooled and water-cooled engines have high thermal loads because of the lack of no-load strokes. For the 2-stroke SI engine, the CR is also limited by the need to inhibit knock.

The primary advantages of the rotary SI engine are the resulting low vibration, high engine speed, and relatively high η_v. The high η_v produces high bmep and good low-speed τ. The high bmep and high attainable engine speed produce high bp. The valveless design results in decreased W. The high bp and low W result in very high bp/W. Because η_e and bsfc are independent of η_v, these parameters are approximately equal for the rotary SI and the 4-stroke piston SI engines.

59.4 EMISSIONS AND FUEL ECONOMY REGULATIONS*

Light-duty vehicles sold in the United States are subject to federal and state regulations regarding exhaust emissions, evaporative emissions, and fuel economy. Heavy-duty vehicles do not have to comply with any fuel economy standards, but are required to comply with standards for exhaust and evaporative emissions. Similar regulations are in effect in many foreign countries.

59.4.1 Light-Duty Vehicles

Light-duty-vehicle (LDV) emissions regulations are divided into those applicable to passenger cars and those applicable to light-duty trucks [defined as trucks of less than 6000 lb gross vehicle weight (GVW) rating prior to 1979 and less than 8500 lb after, except those with more than 45 ft^2 frontal area]. In turn, the light-duty-trucks (LDTs) are divided into four categories: LDT1s are "light light-duty trucks" (GVW < 6000 lb) that have a loaded vehicle weight (LVW) of <3750 lb and must meet the same emissions standards as passenger cars; LDT2s are also light light-duty trucks and have 3751 < LVW < 5750 lb; LDT3s are "heavy light-duty trucks" (GVW > 6000 lb) that have an adjusted loaded vehicle weight (ALVW) of 3751 < ALVW < 5750 lb; LDT4s are heavy light-duty trucks with 5751 < LVW < 8550 lb. Standards for the light-duty trucks in the LDT2–LDT4 categories are not presented or discussed for brevity, but the test procedure described below is used for these vehicles as well as for passenger cars.

U.S. federal and state of California exhaust emissions regulations for passenger cars are presented in Table 59.5. Emissions levels are for operation over the federal test procedure (FTP) transient driving cycle. The regulated emissions levels must be met at the end of the "useful life" of the vehicle. For passenger cars, the useful life is currently defined as five years or 50,000 miles, but the

*The regulations are specified in mixed English and metric units and these specified units are used in this section to avoid confusion.

Table 59.5 Federal and California Emissions Standards for Passenger Cars

	Useful Life (Miles)	THC (gm/mi)	NMHC[a] (gm/mi)	NMOG (gm/mi)	CO (gm/mi)	NO_x (gm/mi)	Particulates (gm/mi)	HCHO (mg/mi)
Precontrol avg.		10.6			84.0	4.0		
Fed. 1975–77	50K	1.5			15.0	3.1		
Cal. 1975–77	50K	0.9			9.0	2.0		
Fed. 1977–79	50K	1.5			15.0	2.0		
Cal. 1977–79	50K	0.41			9.0	1.5		
Fed. 1980	50K	0.41			7.0	2.0		
Cal. 1980	50K	—	0.39		9.0	1.0		
Fed. 1981	50K	0.41			3.4	1.0		
Cal. 1981	50K		0.39		7.0	0.7		
Fed. 1982–84	50K	0.41			3.4	1.0	0.6	
Cal. 1982	50K		0.39		7.0	0.7	0.6	
Cal. 1983–84	50K		0.39		7.0	0.4	0.6	
Fed. 1985–90	50K	0.41			3.4	1.0	0.2	
Cal. 1985–92	50K		0.39		7.0	0.4	0.2	
Fed. Tier 0 (1991–94)	50K[b]	0.41	—	—	3.4	1.0		—
	100K[c]	0.80	—	—	10.0	1.2		—
Cal. Tier 1	50K		0.25	—	3.4	0.4		15[d]
	100K		0.31	—	4.2	0.6		15
Fed. Tier 1[e] (1994–)	50K		0.25	—	3.4	0.4		—
	100K		0.31	—	4.2	0.6		—
Fed. Tier 2 (2003)	100K		0.125	—	1.7	0.2		—
TLEV—alt. fuel	50K		—	0.125	3.4	0.4		15
	100K		—	0.156	4.2	0.6		18
TLEV—bi-fuel[f]	50K		—	0.250	3.4	0.4		15
	100K		—	0.310	4.2	0.6		18
LEV—alt. fuel	50K		—	0.075	3.4	0.2		15
	100K		—	0.090	4.2	0.3		18
LEV—bi-fuel[f]	50K		—	0.125	3.4	0.2		15
	100K		—	0.156	4.2	0.3		18
ULEV—alt. fuel	50K		—	0.040	1.7	0.2		8
	100K		—	0.055	2.1	0.3		11
ULEV—bi-fuel[f]	50K		—	0.075	1.7	0.2		8
	100K		—	0.090	2.1	0.3		11

[a] Organic material nonmethane hydrocarbon equivalent (OMNHCE) for methanol vehicles.

[b] Passenger cars can elect Tier 0 with the lower useful life, but cannot elect Tier 0 at the full useful life.

[c] LDTs that elect Tier 0 must use the full useful life.

[d] This formaldehyde standard must be complied with at 50,000 miles rather than at the end of the full useful life.

[e] Federal Tier 1 standards also require that all light-duty vehicles meet a 0.5% idle CO standard and a cold (20°F FTP) CO standard of 10 gm/mi.

[f] Flexible fuel and bi-fuel vehicles must certify both to the dedicated (alt. fuel) standard (while using the alternative fuel) *and* the bi-fuel standard when using the conventional fuel (e.g., certification gasoline).

Clean Air Act Amendments (CAAA) of 1990 redefined the useful life of passenger cars and LDT1s as 10 years/100,000 miles and 120,000 miles for LDT2–LDT4 light duty trucks. The longer useful life requirement is being phased-in and all vehicles must meet the longer useful life emissions standards in 2003. The first two phases of the FTP are illustrated in Fig. 59.12. Following these two phases (together, these are the LA-4 cycle), the engine is turned off, the vehicle is allowed to hot soak for 10 minutes, the engine is restarted, and the first 505 seconds are repeated as phase 3. The LA-4 cycle is intended to represent average urban driving habits and therefore has an average vehicle

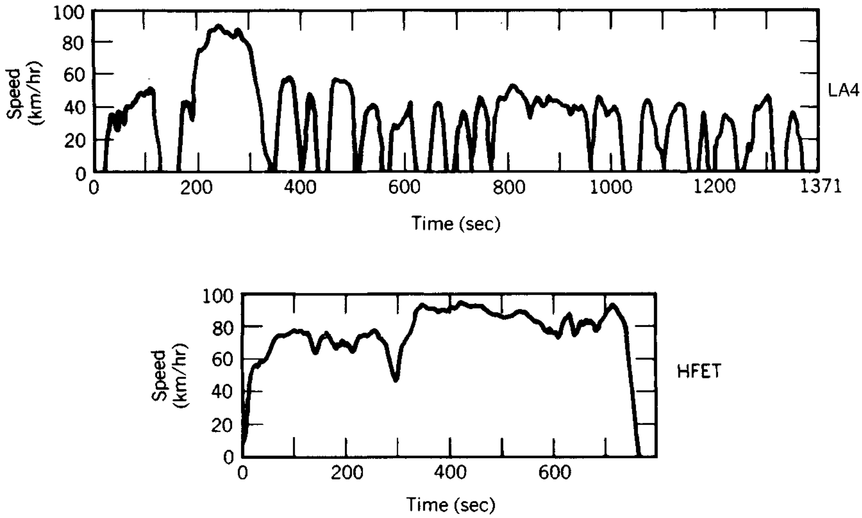

Fig. 59.12 FTP and HFET transient driving cycles.

speed of only 19.5 mi/hr (31.46 km/hr) and includes 2.29 stops per mile (1.42 per kilometer). The LA-4 test length is 7.5 miles (11.98 km) and the duration is 1372 seconds with 19% of this time spent with the vehicle idling. The motor vehicle manufacturer will commit at least two vehicles of each type to the certification procedure. The first vehicle will be used to accumulate 50,000 miles (or 100,000 or 120,000, depending upon the vehicle type and the useful life to which the vehicle is certified) so that a deterioration factor (DF) can be determined for each of the regulated species. The DF is determined by periodic FTP testing over the useful life and is intended to reveal how the emissions levels change with mileage accumulation. The second vehicle is subjected to the FTP test after 4000 miles of accumulation. The emission level of each regulated species is then multiplied by the DF for that species and the product is used to determine compliance with the regulatory standard. All data are submitted to the U.S. Environmental Protection Agency (EPA), which may perform random confirmatory testing, especially of the 4000-mile vehicles.

The 1981 standards represented a 96% reduction in HC and CO and 76% reduction in NO_x from precontrol levels. Beginning in 1980, the California HC standard was specified as either 0.41 grams of total hydrocarbons (THCs) per mile or 0.39 grams of nonmethane hydrocarbons (NMHCs) per mile, in recognition that methane is not considered to be photochemically reactive and therefore does not contribute to smog formation.[17,18] The federal CAAA of 1990 also recognized that methane does not contribute to smog formation and thus instituted a NMHC standard, rather than a THC standard, as part of the new emissions standards phased in from 1994 to 1996. Similarly, because the goal of the hydrocarbon standards is control of ambient ozone, the current California rules regulate Nonmethane Organic Gases (NMOG) rather than NMHCs. Furthermore, in recognition that each of the organic gases in vehicle exhaust has a different efficiency in ozone-formation chemistry, the California rulemaking does not regulate measured NMOG, but rather "reactivity-adjusted" NMOG. The measured NMOG is multiplied by a reactivity adjustment factor (RAF) and the resulting product is then judged against the NMOG standard. The RAF is both fuel-specific and emissions level-specific. The California emissions levels are TLEV (transitional low-emission vehicle), LEV (low-emission vehicle), ULEV (ultra-low-emission vehicle), and ZEV (zero-emission vehicle, or electric vehicle). For LEV vehicles, CARB has established the following RAFs: 1.0 for gasoline, 0.94 for California Phase 2 reformulated gasoline, 0.43 for CNG, 0.50 for LPG, and 0.41 for M85. The particulate emissions standards shown in Table 59.5 apply only to light-duty diesel (LDD) vehicles; light-duty gasoline (LDG) vehicles emit essentially immeasurable quantities of particulates, and therefore are not required to demonstrate compliance with the particulate standards.

LDVs are also required to meet evaporative emissions standards, as determined using a SHED (sealed housing for evaporative determination) test. The precontrol average for evaporative emissions was 50.6 grams per test, which is approximately equivalent to 4.3 gm HC per mile.[19] The federal regulations are 6 gm/test for 1978–80 and 2 gm/test thereafter. The California standard is also 2 gm/test. Evaporative emissions from the LDD are essentially immeasurable and thus are not required to demonstrate compliance with this regulation. California allows LDDs a 0.16 gm/mi credit on exhaust HCs in recognition of the negligible evaporative HC emissions of these vehicles. In the early

1990s, although retaining the standard of 2 gm/test, California instituted new rules requiring use of a new SHED test design, requiring that the test period increase from one day to four days, requiring that the maximum temperature increase to 104°F from 84°F and compliance for 120,000 miles/10 years instead of 50,000 miles/5 years. A similar federal SHED test will be implemented in the near future.

Beginning in 1978, the motor vehicle manufacturers were required to meet the corporate average fuel economy (CAFE) standards shown in Table 59.6 (a different, and lower, set of standards is in effect for "low-volume" manufacturers). The CAFE standard is the mandated average fuel economy, as determined by averaging over all vehicles sold by a given manufacturer. The CAFE requirement is based upon the "composite" fuel economy (mpg_c, in mi/gal), which is a weighted calculation of the fuel economy measured over the FTP cycle (mpg_u, "urban" fuel economy) plus the fuel economy measured over the highway fuel economy test (HFET, as shown in Fig. 59.12) transient driving cycle (mpg_h):

$$mpg_c = \frac{1.0}{\dfrac{0.55}{mpg_u} + \dfrac{0.45}{mpg_h}} \tag{59.38}$$

The manufacturer is assessed a fine of $5 per vehicle produced for every 0.1 mpg below the CAFE standard. Beginning in 1985, an adjustment factor was added to the measured CAFE to account for the effects of changes in the test procedure. Additionally, the motor vehicle manufacturer is subjected to a "gas guzzler" tax for each vehicle sold that is below a minimum composite fuel economy, with the minimum level being: 1983, 19.0 mpg; 1984, 19.5 mpg; 1985, 21.0 mpg; 1986 on, 22.5 mpg. In 1986, the tax ranged from $500 for each vehicle with 21.5–22.4 mpg up to the maximum of $3850 for each vehicle with less than 12.5 mpg.

The federal and state regulations are intended to require compliance based on the best available technology. Therefore, future standards may be subjected to postponement, modification, or waiver, depending on available technology. For more detailed information regarding the standards and measurement techniques, refer to the U.S. Code of Federal Regulations, Title 40, Parts 86 and 88, and to the Society of Automotive Engineers (SAE) Recommended Practices.[20]

59.4.2 Heavy-Duty Vehicles

The test procedure for heavy-duty vehicles (HDVs) is different from that for the LDV. Because many manufacturers of heavy-duty vehicles offer the customer a choice of engines from various vendors, the engines rather than the vehicles must be certified. Thus, the emissions standards are based on grams of pollutant emitted per unit of energy produced (g/bhp-hr, mixed metric and English units, with bhp meaning brake horsepower) during engine operation over a prescribed transient test cycle. The federal standards for 1991 and newer heavy-duty engines are divided into two gross vehicle weight categories for spark ignition engines (less than and greater than 14,000 lb), but a single set of standards is used for diesel engines. The standards also depend upon the fuel used (diesel, gasoline, natural gas, etc.), as presented in Table 59.7. Beginning in 1985, the heavy-duty gasoline (HDG) vehicle was required to meet an idle CO standard of 0.47% and an evaporative emissions SHED Test standard of 3.0 g/test for vehicles with GVW less than 14,000 lb and 4.0 g/test for heavier vehicles.

Table 59.6 CAFE Standards for Passenger Cars Required by the Energy Policy and Conservation Act of 1975

Model Year	Average mpg
1978	18.0
1979	19.0
1980	20.0
1981	22.0[a]
1982	24.0[a]
1983	26.0[a]
1984	27.0[a]
1985	27.5[a]

[a] Set by the Secretary of the U.S. Department of Transportation.

Table 59.7 Emissions Standards[a] for 1991–2000 Model Year Heavy-Duty Engines (in g/bhp-hr)

GVW Category	Engine	Fuel	THCs[b]	NMHCs	CO	NO_x[c]	PM
≤ 14,000 lb	SI	gasoline	1.1	NA	14.4	5.0	NA
≤ 14,000 lb	SI	methanol	1.1	NA	14.4	5.0	NA
≤ 14,000 lb	SI	CNG	NA	0.9	14.4	5.0	NA
≤ 14,000 lb	SI	LPG	1.1	NA	14.4	5.0	NA
> 14,000 lb	SI	gasoline	1.9	NA	37.1	5.0	NA
> 14,000 lb	SI	methanol	1.9	NA	37.1	5.0	NA
> 14,000 lb	SI	CNG	NA	1.7	37.1	5.0	NA
> 14,000 lb	SI	LPG	1.9	NA	37.1	5.0	NA
any (> 8501 lb)	CI	diesel	1.3	NA	15.5	5.0	0.25[d]
any (> 8501 lb)	CI	methanol	1.3	NA	15.5	5.0	0.25[d]
any (> 8501 lb)	CI	CNG	NA	1.2	15.5	5.0	0.10
any (> 8501 lb)	CI	LPG	1.3	NA	15.5	5.0	0.10

[a] Other standards apply to clean fuel vehicles.
[b] Organic material hydrocarbon equivalent (OMHCE) for methanol.
[c] NO_x standard decreases to 4.0 g/bhp-hr beginning in 1998.
[d] Particulate standard decreased to 0.10 g/bhp-hr beginning in 1994.

Prior to 1988, diesels were required only to meet smoke opacity standards (rather than particulate mass standards) of 20% opacity during acceleration, 15% opacity during lugging, and 50% peak opacity. Particulate mass standards for heavy duty diesels came into effect in 1988.

59.4.3 Nonhighway Heavy-Duty Standards

Stationary engines may be subjected to federal, state, and local regulations, and these regulations vary throughout the United States.[19] Diesel engines used in off-highway vehicles and in railroad applications may be subjected to state and local visible smoke limits, which are typically about 20% opacity.[19]

SYMBOLS

AF	air–fuel mass ratio [-]*
AF_s	stoichiometric air–fuel mass ratio [-]
ALVW	adjusted loaded vehicle weight = (curb weight + GVW)/2 [lb]
API	American Petroleum Institute gravity [°]
ASTM	American Society of Testing and Materials
BDC	bottom dead center
bmep	break mean effective pressure [kPa]
bp	brake power [kW]
bsfc	brake specific fuel consumption [g/kW-hr]
CARB	California Air Resources Board
CI	compression ignition (diesel) engine
CN	cetane number [-]
CO	carbon monoxide
CO_2	carbon dioxide
CR	compression ratio (by volume) [-]
C_xH_y	average fuel molecule, with x atoms of carbon and y atoms of hydrogen
D	engine displacement [liters]
DI	direct injection diesel engine
DISC	direct injection stratified charge SI engine

*Standard units, which do not strictly conform to metric practice, in order to produce numbers of convenient magnitude.

DF	emissions deteriorator factor [-]
F	force on dyno torque arm with engine being motored [N]
FA	fuel–air mass ratio [-]
FA_s	stoichiometric fuel–air mass ratio [-]
fp	friction power [kW]
FTP	federal test procedure transient driving cycle
GVW	gross vehicle weight rating
HCs	hydrocarbons
$h_{f,i}^{298}$	standard enthalpy of formation of gaseous species i at 298 K [MJ/kmole of species i]*
$h_{v,i}^{298}$	enthalpy of vaporization of species i at 298 K [MJ/kmole of species i]
ΔH_{RX}^{298}	enthalpy of reaction (heat of combustion) at 298 K [MJ/kg of fuel]
HDD	heavy-duty diesel vehicle
HDG	heavy-duty gasoline vehicle
HDV	heavy-duty vehicle
HFET	highway fuel economy test transient driving cycle
HV	heating value [MJ/kg of fuel]
HHV_p	constant-pressure higher heating value [MJ/kg of fuel]
HHV_v	constant-volume higher heating value [MJ/kg of fuel]
HV_p	constant-pressure heating value [MJ/kg of fuel]
HV_v	constant-volume heating value [MJ/kg of fuel]
IDI	indirect injection diesel engine
ILEV	inherently low emission vehicle
imep	indicated mean effective pressure [kPa]
ip	indicated power [kW]
isfc	indicated specific fuel consumption [g/kW-hr]
k	ratio of specific heats
K	dyno constant [N/kW-min]
L	force on dyno torque arm with engine running [N]
LDD	light-duty diesel vehicle
LDG	light-duty gasoline vehicle
LDT	light-duty truck
LDV	light-duty vehicle
LEV	low-emission vehicle
LHV_p	constant-pressure lower heating value [MJ/kg of fuel]
LHV_v	constant-volume lower heating value [MJ/kg of fuel]
LVW	loaded vehicle weight (curb weight plus 300 lb)
$\dot{m}_A$	air mass flow rate into engine [g/hr]
$\dot{m}_F$	fuel mass flow rate into engine [g/hr]
MON	octane number measured using Motor method [-]
mpg_c	composite fuel economy [mi/gal]
mpg_h	highway (HFET) fuel economy [mi/gal]
mpg_u	urban (FTP) fuel economy [mi/gal]
MW	molecular weight [kg/kmole]
N	engine rotational speed [rpm]
N_2	molecular nitrogen
NMHC	nonmethane hydrocarbons
NMOG	nonmethane organic gases (NMHCs plus carbonyls and alcohols)
NO_x	oxides of nitrogen
O_2	molecular oxygen

*A kmole is a mole based upon a kg; also called a kg-mole.

ON	octane number of fuel [-]
P	pressure [kPa]
R	dyno torque arm length [m]
RAF	reactivity adjustment factor
RON	octane number measured using Research method [-]
r_P	pressure ratio (end of combustion/end of compression) [-]
r_T	temperature ratio (end of combustion/end of compression) [-]
r_v	volume ratio (end of combustion/end of compression) [-]
s	entropy [kJ/kg-K]
S	piston stroke [nm]
SAE	Society of Automotive Engineers
sg_F	specific gravity of fuel
SHED	sealed housing for evaporative determination
SI	spark ignition engine
T	milliliters of tetraethyl lead per gallon of iso-octane [ml/gal]
T	temperature [K]
TA	percent theoretical air
TDC	top dead center
THCs	total hydrocarbons
TLEV	transitional low emission vehicle
ULEV	ultra low emission vehicle
v	specific volume [m^3/kg]
W	engine mass [kg]
X	crankshaft revolutions per power stroke
x	atoms of carbon per molecule of fuel
y	atoms of hydrogen per molecule of fuel
Y_i	mole fraction of species i in exhaust products [-]
ZEV	zero emission vehicle
α	1.0 for HHV, 0.0 for LHV [-]
β	1.0 for liquid fuel, 0.0 for gaseous fuel [-]
η_c	combustion efficiency [-]
η_e	overall engine efficiency [-]
η_M	mechanical efficiency [-]
η_{ti}	indicated thermal efficiency [-]
η_v	volumetric efficiency [-]
ϕ	equivalence ratio, FA/FA_s [-]
ρ_A	density of air at engine inlet [kg/m^3]
ρ_F	density of fuel [kg/m^3]
ρ_w	density of water [kg/m^3]
λ	excess air ratio, AF/AF_s [-]
τ	engine output torque [N-m]

REFERENCES

1. C. F. Taylor, *The Internal Combustion Engine in Theory and Practice,* Vol. 2, MIT Press, Cambridge, MA, 1979.
2. R. S. Benson and N. D. Whitehouse, *Internal Combustion Engines,* Pergamon Press, New York, 1979.
3. U. Adler and W. Baslen (eds.), *Automotive Handbook,* Bosch, Stuttgart, 1978.
4. L. C. Lichty, *Combustion Engine Processes,* McGraw-Hill, New York, 1967.
5. E. F. Obert, *Internal Combustion Engines and Air Pollution,* Harper and Row, New York, 1973.
6. Fuels and Lubricants Committee, "Diesel Fuels—SAE J312 APR82," in *SAE Handbook,* Vol. 3, pp. 23.34–23.39, 1983.
7. Fuels and Lubricants Committee, "Automotive Gasolines—SAE J312 JUN82," in *SAE Handbook,* Vol. 3, pp. 23.25–23.34, 1983.

8. Engine Committee, "Engine Power Test Code—Spark Ignition and Diesel—SAE J1349 DEC80," in *SAE Handbook,* Vol. 3, pp. 24.09–24.10, 1983.
9. Engine Committee, "Small Spark Ignition Engine Test Code—SAE J607a," in *SAE Handbook,* Vol. 3, pp. 24.14–24.15, 1983.
10. C. F. Taylor, *The Internal Combustion Engine in Theory and Practice,* Vol. 1, MIT Press, Cambridge, MA, 1979.
11. A. S. Campbell, *Thermodynamic Analysis of Combustion Engines,* Wiley, New York, 1979.
12. R. D. Matthews, "Relationship of Brake Power to Various Efficiencies and Other Engine Parameters: The Efficiency Rule," *International Journal of Vehicle Design* **4**(5), 491–500 (1983).
13. R. D. Matthews, S. A. Beckel, S. Z. Miao, and J. E. Peters, "A New Technique for Thermodynamic Engine Modeling," *Journal of Energy* **7**(6), 667–675 (1983).
14. B. D. Wood, *Applications of Thermodynamics,* Addison-Wesley, Reading, MA, 1982.
15. P. N. Blumberg, G. A. Lavoie, and R. J. Tabaczynski, "Phenomenological Model for Reciprocating Internal Combustion Engines," *Progress in Energy and Combustion Science* **5**, 123–167 (1979).
16. J. N. Mattavi and C. A. Amann (eds.), *Combustion Modeling in Reciprocating Engines,* Plenum, New York, 1980.
17. Diesel Impacts Study Committee, *Diesel Technology,* National Academy Press, Washington, DC, 1982.
18. R. D. Matthews, "Emission of Unregulated Pollutants from Light Duty Vehicles," *International Journal of Vehicle Design* **5**(4), 475–489 (1983).
19. Environmental Activities Staff, *Pocket Reference,* General Motors, Warren, MI, 1983.
20. Automotive Emissions Committee, "Emissions," in *SAE Handbook,* Vol. 3, Section 25, 1983.
21. Fuels and Lubricants Committee, "Alternative Automotive Fuels—SAE J1297 APR82," in *SAE Handbook,* Vol. 3, pp. 23.39–23.41, 1983.
22. R. C. Weast (ed.), *CRC Handbook of Chemistry and Physics,* 55th ed., CRC Press, Cleveland, OH, 1974.
23. ASTM Committee D-2, *Physical Constants of Hydrocarbons C_1 to C_{10},* American Society for Testing and Materials, Philadelphia, PA, 1963.
24. E. M. Shelton, *Diesel Fuel Oils, 1976,* U.S. Energy Research and Development Administration, Publication No. BERC/PPS-76/5, 1976.
25. C. M. Wu, C. E. Roberts, R. D. Matthews, and M. J. Hall, "Effects of Engine Speed on Combustion in SI Engines: Comparisons of Predictions of a Fractal Burning Model with Experimental Data," SAE Paper 932714, also in SAE Transactions: *Journal of Engines* **102**(3), 2277–2291, 1994.
26. R. D. Matthews, J. Chiu, and D. Hilden, "CNG Compositions in Texas and the Effects of Composition on Emissions, Fuel Economy, and Driveability of NGVs," SAE Paper 962097, 1996.
27. W. E. Liss and W. H. Thrasher, *Variability of Natural Gas Composition in Select Major Metropolitan Areas of the United States,* American Gas Association Laboratories Report to the Gas Research Institute, GRI Report No. GRI-92/0123, 1991; also see SAE Paper 9123464, 1991.
28. R. H. Pahl and M. J. McNally, Fuel Blending and Analysis for the Auto/Oil Air Quality Improvement Research Program, SAE Paper 902098, 1990.
29. S. C. Mayotte, V. Rao, C. E. Lindhjem, and M. S. Sklar, *Reformulated Gasoline Effects on Exhaust Emissions: Phase II: Continued Investigation of Fuel Oxygenate Content, Oxygenate Type, Volatility, Sulfur, Olefins, and Distillation Parameters,* SAE Paper 941974, 1994.
30. J. Kubesh, S. R. King, and W. E. Liss, *Effect of Gas Composition on Octane Number of Natural Gas Fuels,* SAE Paper 922359, 1992.
31. K. Owen and T. Coley, *Automotive Fuels Handbook,* Society of Automotive Engineers, Warrendale, PA, 1990.

BIBLIOGRAPHY

Combustion and Flame, The Combustion Institute, Pittsburgh, PA, published monthly.

Combustion Science and Technology, Gordon and Breach, New York, published monthly.

International Journal of Vehicle Design, Interscience, Jersey, C.I., UK, published quarterly.

Proceedings of the Institute of Mechanical Engineers (Automobile Division), I Mech E, London.

Progress in Energy and Combustion Science, Pergamon, Oxford, UK, published quarterly.

SAE Technical Paper Series, Society of Automotive Engineers, Warrendale, PA.

Symposia (International) on Combustion, The Combustion Institute, Pittsburgh, PA, published biennially.

Transactions of the Society of Automotive Engineers, SAE, Warrendale, PA, published annually.