Chapter 1

Introduction to Internal Combustion Engines

1.1 INTRODUCTION

The goals of this textbook are to describe how internal combustion engines work and provide insight into how engine performance can be modeled and analyzed. The main focus of the text is the application of the thermal sciences, including thermodynamics, combustion, fluid mechanics, and heat transfer, to internal combustion engines. An aspect upon which we will put considerable emphasis is the development of idealized models to represent the actual features of an operating engine.

Engineers use the methods and analyses introduced in the textbook to calculate the performance of proposed engine designs and to parameterize and correlate engines experiments. With the advent of high-speed computers and advanced measurement techniques, today's internal combustion engine design process has evolved from being purely empirical to a rigorous semi-empirical process in which computer based engineering software is used to evaluate the performance of a proposed engine design even before the engine is built and tested. In addition to detailed analysis, the textbook contains numerous computer routines for calculating the various thermal and mechanical parameters that describe internal combustion engine operation.

In this chapter we discuss the engineering parameters, such as thermal efficiency, mean effective pressure, and specific fuel consumption, that are used to characterize the overall performance of internal combustion engines. Major engine cycles, configurations, and geometries are also covered. The following chapters will apply the thermal science principles to determine an internal combustion engine's temperature and pressure profiles, work, volumetric efficiency, and exhaust emissions.

The internal combustion engine was invented and successfully developed in the late 1860s. It is considered one of the most significant inventions of the last century, and has had a significant impact on society, especially human mobility. The internal combustion engine has been the foundation for the successful development of many commercial technologies. Consider how the internal combustion engine has transformed the transportation industry, allowing the invention and improvement of automobiles, trucks, airplanes, and trains. The adoption and continued use of the internal combustion engine in different application areas has resulted from its relatively low cost, favorable power-to-weight ratio, high efficiency, and relatively simple and robust operating characteristics.

An internal combustion engine is an engine in which the chemical energy of the fuel is released inside the engine and used directly for mechanical work, as opposed to an external

Internal Combustion Engines: Applied Thermosciences, Fourth Edition. Allan T. Kirkpatrick.

Companion website: www.wiley.com/go/kirkpatrick/internal

Figure 1.1 Piston and connecting rod. (Courtesy Mahle, Inc.)

combustion engine in which a separate combustor is used to burn the fuel. The reciprocating piston-cylinder geometry is the primary geometry that has been used in internal combustion engines, and is shown in Figure 1.1. As indicated in the figure, a piston oscillates back and forth in a cyclic pattern in a cylinder, transmitting power to a drive shaft through a connecting rod and crankshaft mechanism. Valves or ports are used to control the flow of gas into and out of the engine. This configuration of a reciprocating internal-combustion engine, with an engine block, pistons, valves, crankshaft, and connecting rod, has remained basically unchanged since the late 1800s.

The main differences between a modern-day engine and one built 100 years ago can be seen by comparing their reliability, thermal efficiency, and emissions level. For many years, internal combustion engine research was aimed at improving thermal efficiency and reducing noise and vibration. As a consequence, the thermal efficiency has increased from about 10–20% at the beginning of the twentieth century to values as high as 50% today. Likewise, the power per unit volume has increased from about 0.5 kW/L to 50–100 kW/L.

Internal combustion engine efficiency continues to increase, driven both by legislation and the need to reduce operating costs. The primary US vehicle mileage standard is the Federal Corporate Average Fuel Economy (CAFE) standard. The CAFE standard for passenger vehicles and light-duty trucks was 27.5 miles per gallon (mpg) for a 20-year period from 1990 to 2010. The CAFE standards have risen in the last few years and are expected to double in the next decade. This increase in vehicle mileage requirements will require expanded use of techniques such as electronic control, engine downsizing, turbocharging, supercharging, variable valve timing, low-temperature combustion, and electric motors and transmissions.

Internal combustion engines have become the dominant prime mover technology in several areas. In 1900, most automobiles were steam or electrically powered, but by 1920 most automobiles were powered by gasoline engines. As of the year 2020, in the United States alone there are about 220 million motor vehicles powered by internal combustion engines, with about 12 million new vehicles built each year. In 1900, steam engines were used to power ships and railroad locomotives; today two- and four-stroke diesel engines are used. Prior to 1950, aircraft relied almost exclusively on piston engines. Today gas turbines are the power plant used in large planes, and piston engines continue to dominate the market in small planes.

Internal combustion engines have been designed and built to deliver power in the range from 0.01 kW to 20×10^3 kW, depending on their displacement. They compete in the marketplace with electric motors, gas turbines, and steam engines. The major applications are in the vehicular (see Figure 1.2), railroad, marine (see Figure 1.3), aircraft, stationary power, and home use areas. The vast majority of internal combustion engines are produced for vehicular applications, requiring a power output on the order of 100 kW.

Since 1970, with the recognition of the importance of environmental issues such as the impact of air quality on health, there has also been a great deal of work devoted to reducing the various emissions from engines. The emission levels of current internal combustion engines have decreased to about 5% of the emission levels 40 years ago. Currently, meeting emission requirements is one of the major factors in the design and operation of internal combustion engines. The major emissions from internal combustion engines include nitrogen oxides (NO_x), carbon monoxide (CO), hydrocarbons (HC), particulates (PM), and aldehydes. These combustion products are a significant source of air pollution, as the internal combustion engine is currently the source of about half of the NO_x, CO, and HC pollutants in the environment.

The emissions of carbon dioxide (CO_2), a primary combustion product of hydrocarbon-fueled internal combustion engines are now regulated, as CO_2 is the dominant contributor to climate change. There is increasing interest in carbon-free fuels for internal combustion engines, namely hydrogen and ammonia.

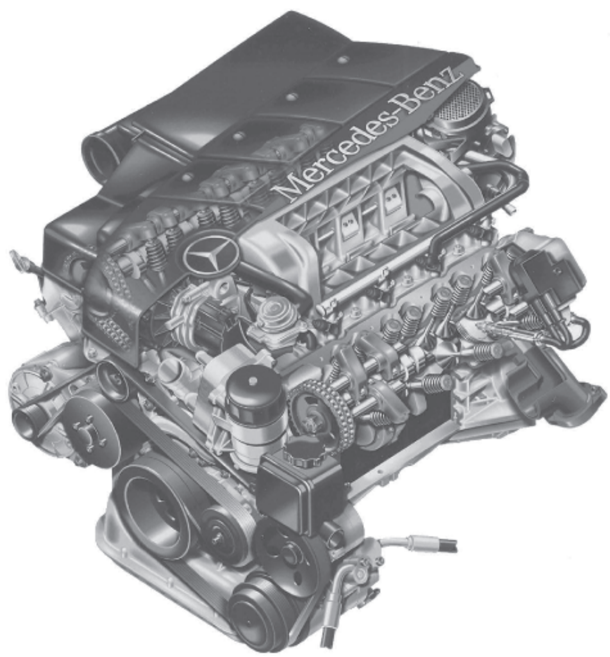

Figure 1.2 Automobile engine. (Courtesy Mercedes-Benz Photo Library.)

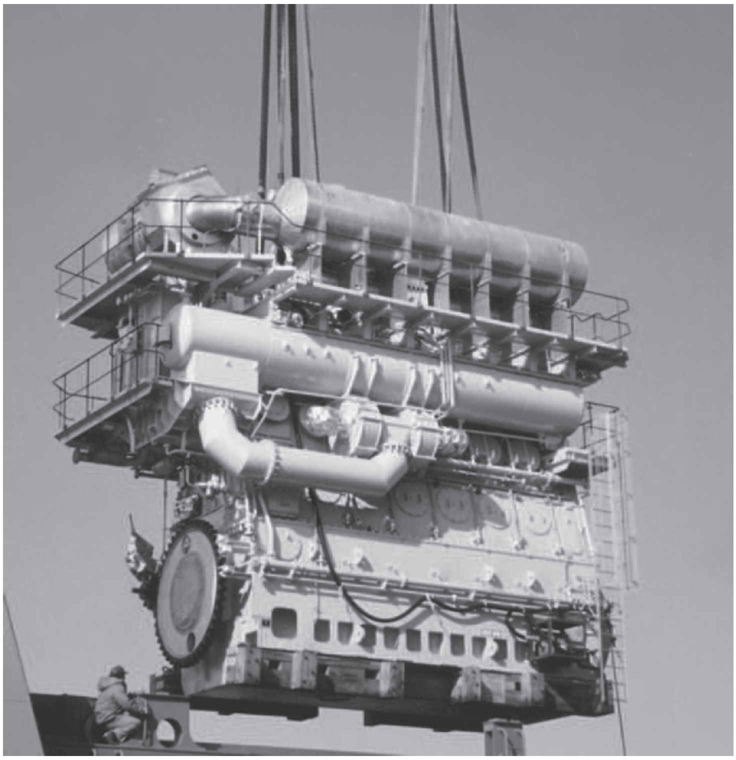

Figure 1.3 Marine engine. (Courtesy Man B&W Diesel.)

1.2 HISTORICAL BACKGROUND

In this section we briefly discuss a few of the major figures in the invention and development of the internal combustion engine. The ingenuity and creativity demonstrated by these early engineers in producing these successful inventions is truly inspiring to today's engine designers. In 1858, J. Lenior (1822–1900), a Belgian engineer, developed a two-stroke engine that developed 6 hp with an efficiency of about 5%. During the intake stroke, a gas–air mixture at atmospheric pressure was drawn into the engine and ignited by a spark, causing the cylinder pressure to increase during the latter half of the stroke, producing work. The return stroke was used to remove the combustion products through an exhaust valve. The Lenior engine was primarily used in stationary power applications.

In 1872, George Brayton (1830–1892), an American mechanical engineer, patented and commercialized a constant pressure internal combustion engine, Brayton's Ready Engine. The engine used two reciprocating piston-driven cylinders, a compression cylinder and an expansion cylinder. This cycle was also called the *flame cycle*, as ignition of the gas–air mixture was by a pilot flame, and the mixture was ignited and burned at constant pressure as it was pumped from the compression cylinder to the expansion cylinder. The Brayton piston engine was used on the first automobile in 1878. The Brayton cycle is the thermodynamic cycle now used by gas turbines, which use rotating fan blades to compress and expand the gas flowing through the turbine.

Nikolaus Otto (1832–1891), a German engineer, developed the Otto Silent Engine, the first practical four-stroke engine with in-cylinder compression, in 1876. With a compression ratio of 2.5, the gas engine produced 2 hp at 160 rpm, and had a brake efficiency of 14%. Nikolaus Otto is considered the inventor of the modern internal combustion engine, and the founder of the internal combustion engine industry. The concept of a four-stroke engine had been conceived and patented by A. de Rochas in 1861, however Otto is recognized as the first person to build and commercialize a working flame ignition engine. Otto had no formal engineering schooling; he was self-taught. He devoted his entire career to the advancement of the internal combustion engine. In 1872, he founded the first internal combustion engine manufacturing company, N. A. Otto and Cie, and hired Gottlieb Daimler and Wilhelm Maybach, who would go on to start the first automobile company, the Daimler Motor Company, in 1890. Otto's son Gustav founded the automotive company now known as BMW.

The first practical two-stroke engine was invented and built by Sir Dugald Clerk (1854–1932), a Scottish mechanical engineer, in 1878. Clerk graduated from Yorkshire College in 1876, and patented his two-stroke engine in 1881. He is well known for his career-long contributions to improvement of combustion processes in large-bore two-stroke engines. Clerk's engine was made of two cylinders – one a working cylinder to produce power and the other a pumping cylinder to compress and transfer the intake air and fuel mixture to the working cylinder. Poppet valves were used for intake flow, and a cylinder port uncovered by the piston on the expansion stroke was used to exhaust the combustion gases.

Many of these early internal combustion engines, such as the Lenior, Brayton, and Otto engines, were powered by coal gas, a mixture of methane, hydrogen, carbon monoxide, and other gases produced by the partial pyrolysis of coal. In the 1880s, crude oil refineries began producing gasoline and kerosene in quantities sufficient to create a market for liquid fueled internal combustion engines.

Gottlieb Daimler (1834–1900), a German engineer, is recognized as one of the founders of the automotive industry. He developed a high-speed, water-cooled four-stroke engine in 1883. The engine had a 70 mm bore and 100 mm stroke, and produced about 1 hp at 650 rpm. The gasoline fuel was vaporized and mixed with the intake air in a carburetor. It then passed by a spring loaded intake valve activated by sub-atmospheric cylinder pressure into the cylinder. The fuel–air mixture was ignited by a flame tube located just below the intake valve. The exhaust valve was operated by a cam lobe on the flywheel. In 1886, Daimler built the first four-wheeled automobile, and founded the Daimler Motor Company in 1890.

Karl Benz (1844–1929), a German engineer, successfully developed a 3.5 hp liquid fueled four-stroke engine with a carburetor and spark ignition in 1885. The ignition system consisted of an electrical induction coil with a rotary breaker driven by the engine and a removable spark plug fitted into the cylinder head, similar to what is found in today's engines. The engine was installed on a custom three wheeled vehicle in 1886, the first "horseless carriage." The transmission was a two-chain arrangement that connected the engine to the rear axle.

In 1897 Rudolph Diesel (1858–1913), a German engineer, developed the first practical four-stroke engine using direct injection of liquid fuel into the combustion chamber. The high compression ratio of the engine resulted in autoignition and combustion of the fuel–air mixture. Diesel graduated from Munich Polytechnic in 1880, and worked with his former professor, Carl von Linde, initially on ammonia Rankine cycle refrigeration, then worked with the MAN company to develop compression ignition engines. He designed his engines to follow Carnot's thermodynamic principles as closely as possible. Accordingly, his initial objective was to have constant temperature combustion; however, this was not realized in practice, and he adopted the strategy of constant pressure combustion.

Rudolph Diesel's single cylinder engine had a bore of 250 mm, stroke of 400 mm, for a 20-liter displacement. The diesel fuel was atomized using air injection, a technique where compressed air entrained diesel fuel in the injector and carried it into the cylinder. The engine operated at a speed of of 170 rpm, and produced 18 hp, with a an efficiency of 27% at full load. This is a much greater efficiency than the steam engines and spark-ignition engines in use at that time.

Sir Harry Ricardo (1885–1974), a mechanical engineering graduate of Cambridge and a prominent English engineer, patented the use of a spherical prechamber, the Ricardo "Comet" to greatly increase the fuel–air mixing rate, allowing diesel engines to be used in high speed, 2000 rpm and higher, engine vehicular applications. During his career, Ricardo also contributed to greater understanding of the role of turbulence, swirl, and squish in enhancing flame speed in both spark and diesel engines; commercialized sleeve valves for aircraft engines, developed an octane rating system for quantifying knock in spark engines; and founded what is now the Ricardo Consulting Engineers Company.

Early engines were air cooled, since they produced relatively low power. Natural convection water cooling using the thermosyphon principle, and forced convection cooling using water pumps was adopted after about 1910 for higher horsepower engines. Henry Ford's Model T engine of 1908, and the Wright Brother's Flyer engine of 1903 used natural convection water cooling.

The first multicylinder diesel engines for trucks were available by 1924. The first commercially available diesel powered automobile was the Mercedes 260D, initially introduced in 1936. It had a 2.6 L four-cylinder prechamber diesel engine, which produced 45 hp at 3000 rpm.

Engine configurations for automobiles in the first half of the twentieth century were primarily four-stroke, water-cooled, with four or six in-line cylinders, equipped with side valves. The valves were located at the side of the cylinder in a combustion pocket. The most common engine configuration used at the present time is the overhead valve configuration.

1.3 ENGINE CYCLES

The two major cycles currently used in internal combustion engines are termed spark-ignition and compression-ignition cycles, also known as Otto and Diesel cycles, named after the two men credited with their invention. As we will see in Chapter 2, the Otto cycle is modeled as a constant volume combustion cycle and the Diesel cycle is modeled as a constant pressure combustion cycle. These cycles can configured as either a two-stroke cycle in which the piston produces power on every downward stroke or a four-stroke cycle in which the piston produces power every other downward stroke.

Spark-Ignition Engine

As shown in Figure 1.4, the four-stroke spark-ignition engine has the following sequence of operations:

1. An intake stroke draws a combustible mixture of fuel and air past the throttle and the intake valve into the cylinder.
2. A compression stroke with the valves closed raises the temperature of the mixture. A spark ignites the mixture toward the end of the compression stroke.
3. An expansion or power stroke results from combustion of the fuel–air mixture.
4. An exhaust stroke pushes out the burned gases past the exhaust valve.

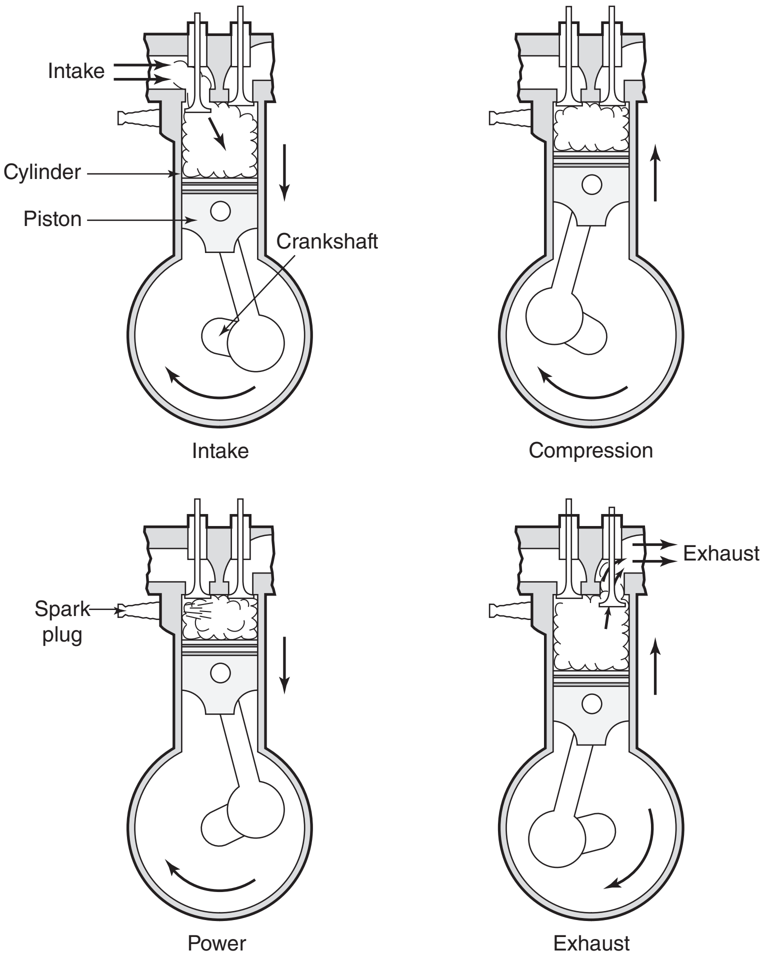

Figure 1.4 Four-stroke spark-ignition engine.

Air enters the engine through the intake manifold, a bundle of passages that evenly distribute the air mixture to individual cylinders. The fuel, typically gasoline, is mixed with the inlet air using a fuel injector or carburetor in the intake manifold, intake port, or directly injected into the cylinder, resulting in the cylinder filling with a homogeneous mixture. When the mixture is ignited by a spark, a turbulent flame develops and propagates through the mixture, raising the cylinder temperature and pressure. The flame is extinguished when it reaches the cylinder walls. If the initial pressure is too high, the compressed gases ahead of the flame will autoignite, causing a problem called knock. The occurrence of knock limits the maximum compression ratio and thus the efficiency of spark-ignition engines. The burned gases exit the engine past the exhaust valves through the exhaust manifold. The exhaust manifold channels the exhaust from individual cylinders into a central exhaust pipe.

In the spark-ignition cycle, a throttle is used to control the amount of air inducted. As the throttle is closed, the amount of air entering the cylinder is reduced, causing a proportional reduction in the cylinder pressure. Since the fuel flow is metered in proportion to the air flow, the throttle in a spark-ignition engine, in essence, controls the power.

Compression Ignition Engine

The four-stroke compression ignition engine has the following sequence:

1. An intake stroke draws inlet air past the intake valve into the cylinder.
2. A compression stroke raises the air temperature above the autoignition temperature of the fuel. Diesel fuel is sprayed into the cylinder near the end of the compression stroke.
3. Evaporation, mixing, ignition, and combustion of the diesel fuel occur during the later stages of the compression stroke and the expansion stroke.
4. An exhaust stroke pushes out the burned gases past the exhaust valve.

The inlet air in the diesel engine is unthrottled, and the combustion is lean. The power is controlled by the amount of fuel injected and the subsequent mixing of the fuel spray with the inlet air. The injection duration is proportional to the engine load. In order to ignite the fuel–air mixture, diesel engines are required to operate at a higher compression ratio, compared to spark-ignition (SI) engines, with typical values in the range of 15 to 20, resulting in a greater theoretical efficiency. Since the diesel fuel is mixed with cylinder air just before combustion is to commence, the knock limitation that occurs in SI engines is greatly reduced.

Diesel engine performance is limited by the time required to mix the fuel and air, as incomplete mixing and combustion results in decreased power, increased unburned hydrocarbon emissions, and visible smoke. As we shall see, many different diesel combustion chamber designs have been invented to achieve adequate mixing. There are two main types of diesel combustion systems, direct injection (DI) into the main cylinder and indirect injection (IDI) into a prechamber connected to the main cylinder. Direct injection engines predominate when the operating range of the engine is fairly narrow, such as in ships, locomotives, and electric power generation. Indirect injection engines tend to be used where the engine is expected to perform at high speeds over a wide range of loads, such as in an automobile.

With indirect injection, air is compressed into a prechamber during the compression stroke, producing a highly turbulent flow field, and thus high mixing rates when the diesel fuel is sprayed into the prechamber toward the end of the compression stroke. The combustion process is initiated in the prechamber, raising the pressure in the prechamber above that of the main chamber, which forces the combusting mixture of burning gases, fuel, and air back into the main chamber, resulting in the propagation of a highly turbulent swirling flame into the main chamber.

Since the mixing time is inversely proportional to the engine speed, diesel engines are classified into three classes, high speed, medium speed, and low speed. High-speed diesels are designed to operate at speeds of 1000 rpm or higher, have up to a 300 mm bore, and use high quality distillate fuels. Medium-speed diesels operate at speeds of 375–1000 rpm, have a medium bore typically between 200 and 600 mm, and can operate with a range of fuels. The low-speed class of diesel engines operate at speeds less than 375 rpm, are typically large bore (> 600 mm) two-stroke cycle engines, and use residual fuel oil. Each engine manufacturer has worked to optimize the design for a particular application, and that each manufacturer has produced an engine with unique characteristics illustrates that the optimum design is highly dependent on the specific application.

Two-Stroke Cycle

As the name implies, two-stroke engines need only two strokes of the piston or one revolution to complete a cycle. There is a power stroke every revolution instead of every two revolutions as for four-stroke engines. Two-stroke engines are mechanically simpler than four-stroke engines, and have a higher specific power, the power to weight ratio. They can use either spark or compression ignition cycles. One of the performance limitations of two-stroke engines is the scavenging process, simultaneously exhausting the burnt mixture and introducing the fresh fuel–air mixture into the cylinder. As we shall see, a wide variety of two-stroke engines have been invented to ensure an acceptable level of scavenging.

The principle of operation of a crankcase scavenged two-stroke engine, developed by Joseph Day (1855–1946), is illustrated in Figure 1.5. During compression of the crankcase scavenged two-stroke cycle, a subatmospheric pressure is created in the crankcase. In the example shown, this opens a reed valve letting air rush into the crankcase. Once the piston reverses direction during combustion and expansion begins, the air in the crankcase closes the reed valve so that the air is compressed. As the piston travels further, it uncovers holes or exhaust ports, and exhaust gases begin to leave, rapidly dropping the cylinder pressure to that of the atmosphere. Then the intake ports are opened and compressed air from the crankcase flows into the cylinder pushing out the remaining exhaust gases. This pushing out of exhaust by the incoming air is called scavenging.

Herein lies one problem with two-stroke engines: the scavenging is not perfect; some of the air will go straight through the cylinder and out the exhaust port, a process called short circuiting. Some of the air will also mix with exhaust gases and the remaining incoming air will push out a portion of this mixture. The magnitude of the problem is strongly dependent on the port designs and the shape of the piston top.

Less than perfect scavenging is of particular concern if the engine is a carbureted gasoline engine, for instead of air being in the crankcase there is a fuel–air mixture. Some of this fuel–air mixture will short circuit and appear in the exhaust, wasting fuel and increasing the hydrocarbon emissions. Carbureted two-stroke engines are used where efficiency is not of primary concern and advantage can be taken of the engine's simplicity; this translates into lower cost and higher power per unit weight. Familiar examples include motorcycles, chain saws, outboard motors, and model airplane engines. However, use in motorcycles is decreasing because they have poor emission characteristics. Two-stroke industrial engines are mostly diesel, and typically supercharged. With a two-stroke diesel or fuel injected

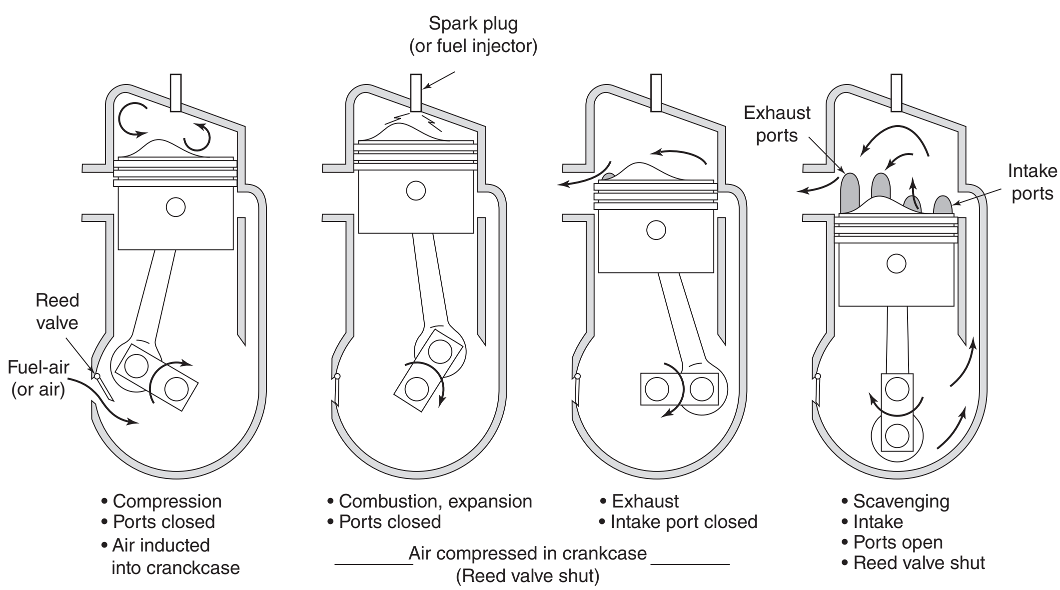

Figure 1.5 A cross-scavenged two-stroke cycle.

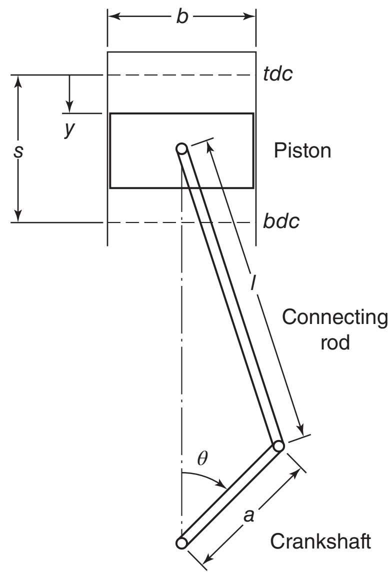

Figure 1.6 Engine slider crank geometry.

gasoline engine, air only is used for scavenging, so loss of fuel through short circuiting or mixing with exhaust gases is not a problem.

1.4 ENGINE PERFORMANCE PARAMETERS

Engine Geometry

A simple model of the crankshaft, connecting rod, piston, and head assembly of an engine cylinder is the flat-top slider crank mechanism shown in Figure 1.6. Of particular interest are the following geometric parameters: bore b, connecting rod length l, crank radius a, stroke s, clearance height c, and crank angle θ. The stroke s is twice the crankshaft radius a.

The term *top dead center* (tdc) of an engine refers to the crankshaft at a position $\theta = 0°$. The cylinder volume at this position is minimum and is called the clearance volume, V_c. The term *bottom dead center* (bdc) refers to the crankshaft at a position $\theta = 180°$. The cylinder volume at bottom dead center is the maximum volume V_1.

The compression ratio r is defined as the ratio of the maximum to minimum volume.

$$r = \frac{V_{\mathrm{bdc}}}{V_{\mathrm{tdc}}} = \frac{V_1}{V_c} \tag{1.1}$$

The displacement volume V_d is the difference between the maximum and minimum volume; for a single cylinder,

$$V_d = V_1 - V_c = \frac{\pi}{4} b^2 s \tag{1.2}$$

Useful expressions relating r, V_d, V_1, and V_c are

$$V_1 = V_{bdc} = \frac{r}{r-1} V_d \tag{1.3}$$

$$V_c = V_{tdc} = \frac{V_d}{r-1} \tag{1.4}$$

The piston clearance height c at top dead center is

$$c = \frac{s}{r - 1} \tag{1.5}$$

For multicylinder engines, the total displacement volume V_d is the product of the number of cylinders, n_c, and the volume of a single cylinder.

$$V_d = n_c \frac{\pi}{4} b^2 s \tag{1.6}$$

The mean piston speed $\overline{U}_p$ is an important parameter in engine design since stresses and other factors scale with piston speed rather than with engine speed. Since the piston travels a distance of twice the stroke per revolution it should be clear that

$$\overline{U}_p = 2Ns \tag{1.7}$$

The engine speed N refers to the rotational speed of the crankshaft and is in units of revolutions per second or revolutions per minute (rpm). The engine frequency, ω, also refers to the rotation rate of the crankshaft but in units of radians per second. Examples 1.1 and 1.2 provide unit conversions for engine rpm.

Engine Work, Power, Torque, and Mechanical Efficiency

The indicated work W_i of an engine is the net work done by the gas during a compression and expansion cycle. It is equal to the integral of the pressure over the cylinder volume:

$$W_i = \int PdV \tag{1.8}$$

The engine power $\dot{W}$ is the rate at which work W is done by the engine, and for an engine with n_c cylinders is

$$\dot{W} = n_c\ W\ N/2 \quad \text{(4 stroke)} \tag{1.9}$$

$$\dot{W} = n_c\ W\ N \quad \text{(2 stroke)} \tag{1.10}$$

since the four-stroke engine has two revolutions per power stroke and the two-stroke engine has one revolution per power stroke.

The brake power $\dot{W}_b$ is the power output of the engine measured by a dynamometer. Early dynamometers were simple brake mechanisms, hence the use of the term *brake*. The engine torque, τ, is a measure of the work done per unit rotation (radians) of the crank. As we shall see when discussing dynamometers in Chapter 12, the brake power and torque for both two- and four-stroke engines are related by

$$\dot{W}_b = 2\pi\tau N \tag{1.11}$$

The net brake power is from the complete engine; whereas gross brake power is from an engine without the cooling fan, muffler, and tail pipe. The brake power is less than the indicated power due to engine mechanical friction, pumping losses in the intake and exhaust, and accessory power needs, which are grouped as a friction power loss, $\dot{W}_f$:

$$\dot{W}_f = \dot{W}_i - \dot{W}_b \tag{1.12}$$

The ratio of the brake power to the indicated power is the mechanical efficiency, η_m:

$$\eta_m = \dot{W}_b/\dot{W}_i = 1 - \dot{W}_f/\dot{W}_i \tag{1.13}$$

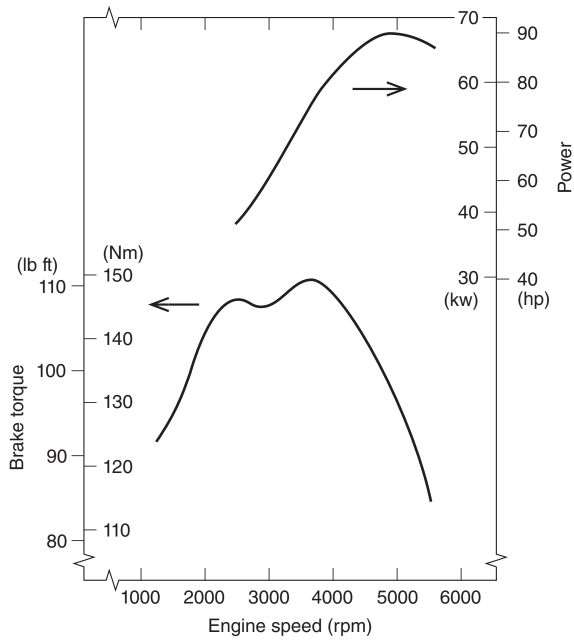

Figure 1.7 Wide-open throttle (WOT) performance of an automotive four-stroke engine.

The wide open throttle performance of a 2.0 L automotive four-stroke engine is plotted in Figure 1.7. As with most engines, the torque and power both exhibit maxima with engine speed. Viscous friction effects increase quadratically with engine speed, causing the torque curve to decrease at high engine speeds. The maximum torque occurs at lower speed than maximum power, since power is the product of torque and speed. Notice that the torque curve is rippled. This is due to both inlet and exhaust airflow dynamics and mechanical friction, discussed later.

Mean Effective Pressure

The mean effective pressure (mep), defined in Equation (1.14), is the work done per unit displacement volume. It is the average pressure that results in the same amount of work actually produced by the engine and has units of force/area. The mean effective pressure is a very useful parameter as it scales out the effect of engine size, allowing performance comparison of engines of different displacement.

$$\text{mep} = \frac{W}{V_d} \tag{1.14}$$

There are three useful mean effective pressure parameters – imep, bmep, and fmep. The indicated mean effective pressure (imep) is the net work per unit displacement volume done by the gas during compression and expansion. The name originates from the use of an ’indicator’ card used in the past to plot measured pressure versus volume. The pressure in the cylinder initially increases during the expansion stroke due to the heat addition from the fuel, and then decreases due to the increase in cylinder volume.

The brake mean effective pressure (bmep) is the external shaft work per unit volume done by the engine. The name originates from the *brake* dynamometer used to measure the torque produced by the rotating shaft. Typical values of bmep for vehicular engines depend on the load. A low-load bmep is about 5 bar, a mid-load bmep is about 10 bar, and a high-load bmep is about 20 bar. Use of turbo- or supercharging is generally required to produce a high-load bmep.

Based on torque, the bmep is

$$\begin{aligned} \text{bmep} &= \frac{4\pi\tau}{V_d} \quad \text{(4 stroke)} \\ &= \frac{2\pi\tau}{V_d} \quad \text{(2 stroke)} \end{aligned} \tag{1.15}$$

and in terms of power, the bmep is

$$\begin{aligned} \text{bmep} &= \frac{\dot{W}_b}{V_d\, N/2} \quad \text{(4 stroke)} \\ &= \frac{\dot{W}_b}{V_d\, N} \quad \text{(2 stroke)} \end{aligned} \tag{1.16}$$

The bmep can also be expressed in terms of piston area A_p, mean piston speed $\overline{U}_p$, and number of cylinders n_c:

$$\begin{aligned} \text{bmep} &= \frac{4\dot{W}_b}{n_c A_p \overline{U}_p} \quad \text{(4 stroke)} \\ &= \frac{2\dot{W}_b}{n_c A_p \overline{U}_p} \quad \text{(2 stroke)} \end{aligned} \tag{1.17}$$

The friction mean effective pressure (fmep) includes the mechanical engine friction, the pumping losses during the intake and exhaust strokes, and the work to run auxiliary components such as oil and water pumps. Accordingly, the friction mean effective pressure (fmep) is the difference between the imep and the bmep. Determination of the fmep is discussed further in Chapter 10.

$$\text{fmep} = \text{imep} - \text{bmep} \tag{1.18}$$

The bmep of two different displacement naturally aspirated automobile engines at wide-open throttle (WOT) is compared versus mean piston speed in Figure 1.8. Notice that when performance is scaled to be size independent, there is considerable similarity.

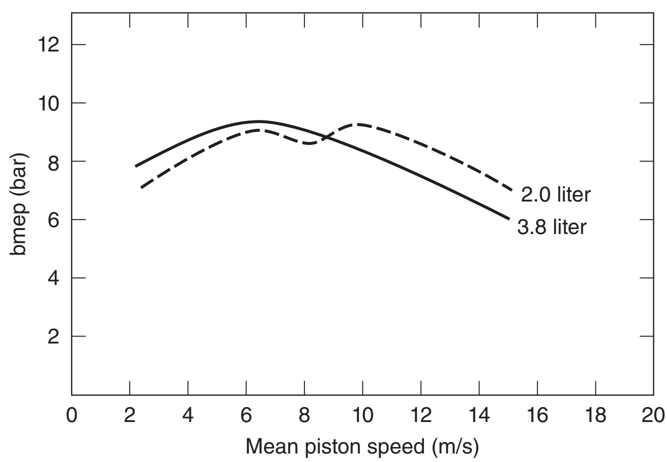

Figure 1.8 Brake mean effective pressure at WOT versus mean piston speed for two automotive engines.

Volumetric Efficiency

A performance parameter of importance for four-stroke engines is the volumetric efficiency, e_v. It is defined as the mass of air m_a inducted into the cylinder at bottom dead center divided by the mass that would occupy the cylinder volume V_d at a density ρ_i of the intake manifold air. The flow restrictions in the intake system, including the throttle, intake port, and valve, create a pressure drop in the inlet flow, which reduces the density and thus the mass of the gas in the cylinder. The volumetric efficiency is thus a mass ratio and not a volume ratio.

The volumetric efficiency for an engine operating at a speed N is

$$e_v = \frac{m_a}{\rho_i V_d} = \frac{\dot{m}_a}{\rho_i V_d N/2} \tag{1.19}$$

The factor of 2 in Equation (1.19) accounts for the two revolutions per cycle in a four-stroke engine. The ideal gas equation is used to determine the air density in the intake manifold,

$$\rho_i = \frac{P_i}{RT_i} \tag{1.20}$$

The parameter R is the ideal gas constant for air with a value of $R = 0.287$ kJ/kg-K. The intake manifold air density is used as a reference condition instead of the standard atmosphere, so that the effect of a supercharger is not included in the definition of volumetric efficiency. The standard atmosphere is assumed to have a temperature $T_a = 298$ K and pressure $P_a = 101.3$ kPa, with a corresponding density $\rho_a = 1.184$ kg/m^3.

For two-stroke cycles, an equivalent volumetric efficiency parameter is the delivery ratio D_r, which is defined in terms of the ambient air density ρ_a instead of the intake manifold density:

$$D_r = \frac{m_a}{\rho_a V_d} = \frac{\dot{m}_a}{\rho_a V_d N} \tag{1.21}$$

A representative plot of volumetric efficiency versus engine speed of an automotive four-stroke engine is shown in Figure 1.9. The shape and location of the peaks of the volumetric efficiency curve are very sensitive to the engine speed as well as the manifold configuration. Some configurations produce a flat curve, others produce a very peaked and asymmetric curve. As we will see later, the volumetric efficiency is also influenced by the valve size, valve lift, and valve timing. It is desirable to maximize the volumetric efficiency

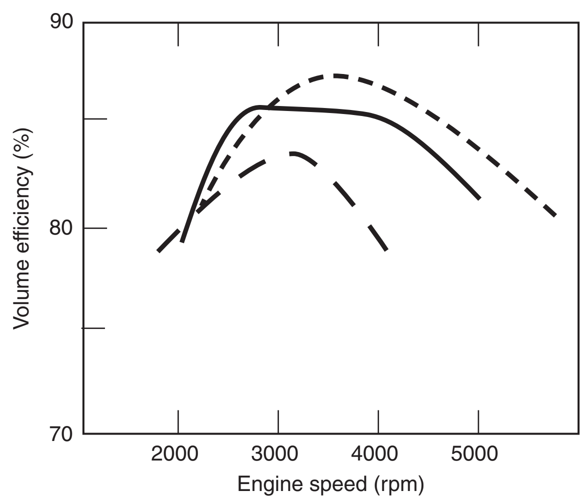

Figure 1.9 Effect of engine speed and intake manifold geometry on volumetric efficiency. Adapted from Armstrong and Stirrat (1982).

of an engine since the amount of fuel that can be burned and power produced for a given engine displacement (hence size and weight) is maximized. Although it does not influence in any way the thermal efficiency of the engine, the volumetric efficiency will influence the overall thermal efficiency of the system in which it is installed. As Example 1.1 below indicates, the volumetric efficiency is useful for determination of the air flowrate of an engine of a given displacement and speed.

EXAMPLE 1.1 ***Volumetric Efficiency***

A four-stroke 2.5 L (2.5×10^{-3} m^3) direct injection automobile engine is tested on a dynamometer at a speed of 2500 rpm. It produces a torque of 150 Nm, and its volumetric efficiency is measured to be 0.85. What is the brake power $\dot{W}_b$, and the mass air flowrate $\dot{m}_a$ through the engine? The intake manifold air pressure and temperature are 75 kPa and 313 K.

SOLUTION The engine power $\dot{W}_b$ is:

$$\dot{W}_b = 2\pi\tau N = (2\pi)(150)(2500/60) = 39.3 \text{ kW}$$

The intake manifold air density is

$$\rho_i = P/RT_i = 75/(0.287 \cdot 313) = 0.835 \text{ kg/m}^3$$

and the mass air flowrate $\dot{m}_a$ is:

$$\dot{m}_a = \frac{1}{2} e_v \rho_i V_d N = \frac{1}{2}(0.85)(0.835)(2.5 \times 10^{-3})(2500/60) = 3.70 \times 10^{-2} \text{ kg/s}$$

Specific Fuel Consumption

The specific fuel consumption is a comparative metric for the efficiency of converting the chemical energy of the fuel into work produced by the engine. As with the mean effective pressure, there are two specific fuel consumption parameters, brake and indicated. The brake-specific fuel consumption (bsfc) is the fuel flowrate $\dot{m}_f$, divided by the brake power $\dot{W}_b$. It has three parameters that are standard measurements in an engine test: the fuel flowrate, the torque, and the engine speed. The brake-specific fuel consumption for naturally aspirated automobile engines depends on the engine load and speed, and can have values ranging from about 175 to 400 g/kWh.

$$\text{bsfc} = \frac{\dot{m}_f}{\dot{W}_b} = \frac{\dot{m}_f}{2\pi\tau N} \tag{1.22}$$

The indicated specific fuel consumption (isfc) is the ratio of the mass of fuel injected during a cycle to the indicated cylinder work, and is used to compare engine performance in computational simulations that do not include the engine friction.

$$\text{isfc} = \frac{m_f}{W_i} \tag{1.23}$$

An engine performance map is used to present the effects of speed and load on engine performance, as shown in Figure 1.10. The engine speed N or the mean piston speed $\bar{U}_p$ is plotted on the x-axis, and the brake mean effective pressure (bmep) is plotted on the y-axis.

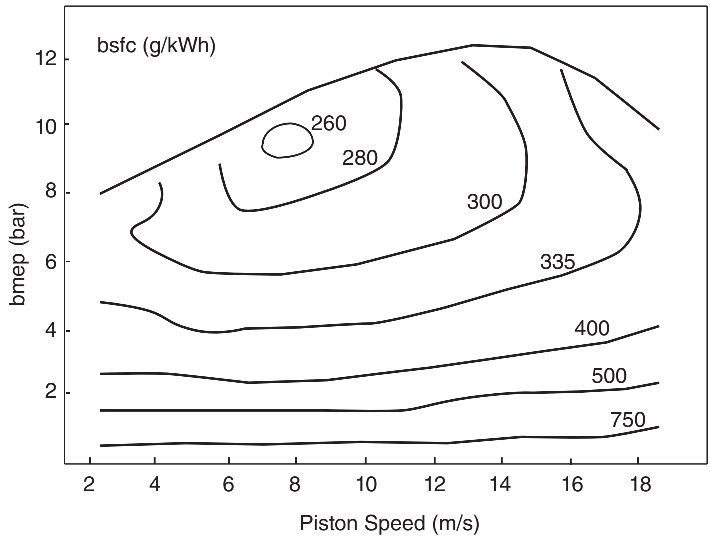

Figure 1.10 Performance map of bmep and bsfc versus mean piston speed for an automotive spark-ignition engine.

Contour lines of constant bsfc are plotted on this load-speed plane. The lines of constant bsfc are approximately independent of displacement for a given engine family, so engine performance maps can be used to match an engine with a given load. For a spark-ignition engine, the upper envelope on the map is the wide open throttle line. Its shape reflects variations in the volumetric efficiency with engine speed, although small changes in inlet air density are also involved.

The specific fuel consumption and engine efficiency are inversely related, so that the lower the specific fuel consumption, the greater the engine efficiency. Engineers use bsfc rather than thermal efficiency primarily because a more or less universally accepted definition of thermal efficiency does not exist. We will explore the reasons why in Chapter 4. Note for now only that there is an issue with assigning a value to the energy content of the fuel. Let us call that energy the heat of combustion q_c; the brake thermal efficiency η_t is then

$$\eta_t = \frac{\dot{W}_b}{\dot{m}_f\, q_c} = \frac{1}{\text{bsfc}\, q_c} \tag{1.24}$$

Inspection of Equation (1.24) shows that bsfc is a valid measure of efficiency provided q_c is held constant. Thus, two different engines can be compared on a bsfc basis provided that they are operated with the same fuel.

Air–Fuel and Equivalence Ratios

Since internal combustion engines require both a fuel and an oxidizer for the combustion process, another engine parameter is the air–fuel ratio, AF, expressed on a mass or a mass flow-rate basis.

$$\text{AF} = \frac{m_a}{m_f} = \frac{\dot{m}_a}{\dot{m}_f} \tag{1.25}$$

The reciprocal of the air–fuel ratio is the fuel–air ratio, FA:

$$\text{FA} = \frac{m_f}{m_a} = \frac{\dot{m}_f}{\dot{m}_a} \tag{1.26}$$

A dimensionless measure of the fuel–air ratio is the equivalence ratio, ϕ, which is the ratio of the actual fuel–air ratio to the stoichiometric fuel–air ratio. The word *stoichiometric* is from the Greek, meaning "element measure." A stoichiometric reaction of a hydrocarbon (HC) is defined such that the fuel burns completely and the only products are carbon dioxide (CO_2) and water (H_2O).

$$\phi = \frac{\text{FA}}{\text{FA}_s} \tag{1.27}$$

The equivalence ratio is used to characterize the fuel–air mixture composition. If $\phi = 1$ the mixture is stoichiometric, if $\phi < 1$ the mixture is lean, and if $\phi > 1$ the mixture is rich.

EXAMPLE 1.2 ***Engine Performance Parameters***

A six-cylinder, four-stroke automobile engine is being designed to produce 75 kW at 2000 rpm with a bsfc of 260 g/kWh and a bmep of 9.2 bar (920 kPa). The engine is to have equal bore and stroke, and will be fueled with a stoichiometric mixture of gasoline and air with an air–fuel ratio (AF) of 15.27. Gasoline has a heat of combustion $q_c = 44{,}510$ kJ/kg. (a) What is the design displacement volume V_d and bore b? (b) What is the mean piston speed at the design point? (c) What are the cycle average fuel flow and air flowrates and the fuel consumption per cycle per cylinder? (d) What is the work per cycle per cylinder? (e) What is the thermal efficiency?

SOLUTION

(a) The displacement volume V_d is

$$V_d = \frac{\dot{W}_b}{\text{bmep}\ N/2} = \frac{75}{(920)(2000/2)(1/60)} = 4.89 \times 10^{-3}\ \text{m}^3 = 4.89\ \text{L}$$

$$b = \left(\frac{V_d}{n_c}\frac{4}{\pi}\right)^{1/3} = \left(\frac{4.89 \times 10^{-3}}{6}\frac{4}{\pi}\right)^{1/3} = 101.1\ \text{mm}$$

Most automobile engines have a 90–100 mm bore and stroke.

(b) The mean piston speed is

$$\overline{U_p} = 2Ns = (2)(101.1 \times 10^{-3})(2000/60) = 6.76\ \text{m/s}$$

(c) The cycle average fuel consumption rate per cylinder is

$$\overline{\dot{m}_f} = \text{bsfc} \times \dot{W}_b/n_c = 260 \times 75/(6 \times 3600) = 0.903\ \text{g/s}$$

so the mass of fuel injected per cylinder per cycle is

$$m_f = \overline{\dot{m}_f}/(N/2) = 0.903/(2000/(2 \times 60)) = 5.42 \times 10^{-2}\ \text{g}$$

and the cycle average airflowrate is

$$\overline{\dot{m}_a} = \text{AF} \times \overline{\dot{m}_f} = (15.27)(0.903) = 13.8\ \text{g/s}$$

(d) The brake work per cycle per cylinder is

$$W_b = \dot{W}_b/(n_c\ N/2) = 75/(6 \times 2000/60/2) = 0.75\ \text{kJ}$$

(e) The brake thermal efficiency is

$$\eta_t = \frac{1}{\text{bsfc}\ q_c} = \frac{3600}{(0.260)(44,510)} = 0.31$$

Engine Kinematics

Assuming a flat piston top, the instantaneous cylinder volume, $V(\theta)$, at any crank angle is

$$V(\theta) = V_c + \frac{\pi}{4} b^2 y \tag{1.28}$$

where y is the instantaneous stroke distance from top dead center:

By reference to Figure 1.6

$$y = l + a - [(l^2 - a^2 \sin^2\theta)^{1/2} + a \cos\theta] \tag{1.29}$$

If the instantaneous volume $V(\theta)$ is nondimensionalized by the volume at bottom dead center, V_{bdc}, then the nondimensional volume $\tilde{V}(\theta)$ is

$$\tilde{V}(\theta) = \frac{V(\theta)}{V_{bdc}} = \frac{1}{r} + \frac{r-1}{r}\frac{y}{s} \tag{1.30}$$

We define a nondimensional parameter, ϵ, the ratio of the crankshaft radius a to the connecting rod length l, as

$$\epsilon = \frac{a}{l} = \frac{s}{2l} \tag{1.31}$$

The range of ϵ for the slider-crank geometries used in modern engines is about 0.25 to 0.33.

Therefore, the nondimensional piston displacement y/s is

$$\frac{y}{s} = \frac{1}{2}(1 - \cos\theta) + \frac{1}{2\epsilon}[1 - (1 - \epsilon^2 \sin^2\theta)^{1/2}] \tag{1.32}$$

and the nondimensional cylinder volume $\tilde{V}(\theta)$ is

$$\tilde{V}(\theta) = \frac{1}{r} + \frac{(r-1)}{2r}(1 - \cos\theta) + \frac{1}{2\epsilon r}[1 - (1 - \epsilon^2 \sin^2\theta)^{1/2}] \tag{1.33}$$

For $\epsilon < 1$, we can expand the $\sin^2\theta$ term in a Taylor series,

$$(1 - \epsilon^2 \sin^2\theta)^{1/2} \simeq 1 - \frac{1}{2}\epsilon^2 \sin^2\theta + O(\epsilon^4) \tag{1.34}$$

so

$$\frac{y}{s} \simeq \frac{1}{2}(1 - \cos\theta) + \frac{\epsilon}{4}\sin^2\theta \tag{1.35}$$

As $\epsilon \to 0$, the approximate volume $\tilde{V}(\theta)$ can then be expressed as a function only of the compression ratio r:

$$\tilde{V}(\theta) \simeq \frac{1}{r} + \frac{(r-1)}{2r}(1 - \cos\theta) \tag{1.36}$$

The cylinder volumes predicted by Equations (1.33) and (1.36) are compared in Figure 1.11 for a value of $\epsilon = 1/3$, using the Matlab® program `Volume.m` listed in the Appendix. Both equations give identical results at bottom dead center and top dead center. The approximate volume relation underpredicts the exact cylinder volume by about 18% at ±59 degrees near the middle of the stroke.

The instantaneous piston velocity U_p can be found by replacing θ with ωt and differentiating Equation (1.32) with respect to time t giving

$$U_p(\omega t) = \frac{dy}{dt} = \frac{\omega s \sin(\omega t)}{2}\left[1 + \frac{\epsilon \cos\omega t}{(1 - \epsilon^2 \sin^2\omega t)^{1/2}}\right] \tag{1.37}$$

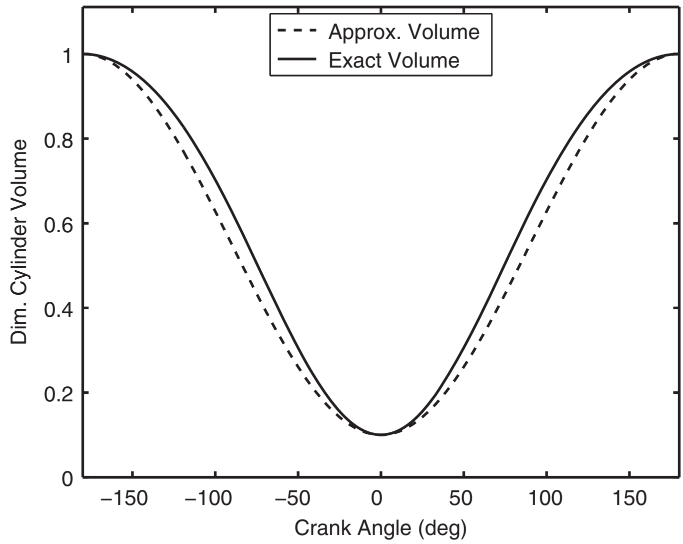

Figure 1.11 Cylinder volume vs. crank angle for $r = 10, \epsilon = 1/3$ (Equations (1.33) and (1.36)).

Equation (1.37) can be nondimensionalized by the mean piston speed $\overline{U}_p$, resulting in

$$\tilde{U}_p(\theta) = \frac{U_p}{\overline{U}_p} = \frac{\pi}{2}\sin\theta\left[1 + \frac{\epsilon\cos\theta}{(1-\epsilon^2\sin^2\theta)^{1/2}}\right] \tag{1.38}$$

Using the Matlab® program `Velocity.m` listed in the Appendix, the nondimensional velocity $\tilde{U}_p(\theta)$ is plotted versus crank angle from top dead center (tdc) to bottom dead center (bdc) in Figure 1.12 for a value of $\epsilon = 1/3$. The piston velocity is zero at tdc

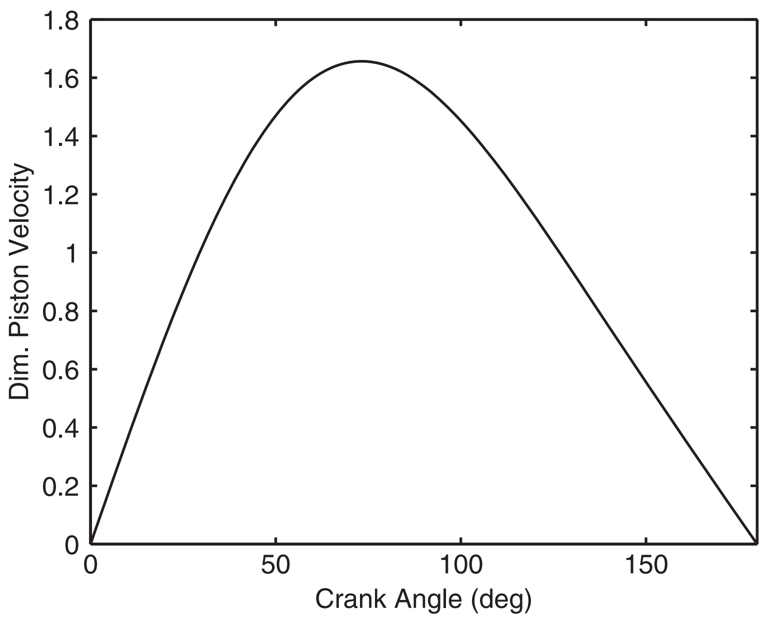

Figure 1.12 Nondimensional velocity vs. crank angle for $\epsilon = 1/3$ (Equation (1.38)).

and bdc. Due to the geometry of the slider crank mechanism, the velocity profile is nonsymmetric. For this example, the maximum nondimensional velocity $\tilde{U}_p(\theta) = 1.65$ occurs at $\theta = 72°$ after tdc.

If we neglect terms of $O(\epsilon^2)$, and use the trigonometric identity $\sin^2 \omega t = (1 - \cos 2\omega t)/2$, the piston velocity can be approximated as

$$U_p = \frac{dy}{dt} \simeq \frac{\omega s}{2}\left[\sin \omega t + \frac{\epsilon}{2} \sin 2\omega t\right] \tag{1.39}$$

The acceleration a_p is found by differentiating Equation (1.39) with respect to time

$$a_p = \frac{d^2 y}{dt^2} \simeq \frac{\omega^2 s}{2}[\cos \omega t + \epsilon \cos 2\omega t] \tag{1.40}$$

Note that the velocity and acceleration terms have two components, one varying with the same frequency ω as the crankshaft, known as the primary term, and the other varying at twice the crankshaft frequency 2ω, known as the secondary term. In the limit of an infinitely long connecting rod, i.e., $\epsilon \to 0$, the motion reduces to a simple harmonic at a frequency ω.

The reciprocating motion of the connecting rod and piston creates accelerations and thus inertial forces and moments that need to be considered in the choice of an engine configuration. In multicylinder engines, the cylinder arrangement and firing order are chosen to minimize the primary and secondary forces and moments. Complete cancellation is possible for the following four-stroke engines: in-line 6- and 8-cylinder engines; horizontally opposed 8- and 12-cylinder engines, and 12- and 16-cylinder V engines (Taylor 1985).

Scaling of Engine Performance

The performance characteristics of three different diesel engines are compared in Table 1.1. The engines are a four-cylinder 1.9 L automobile engine, a six-cylinder 5.9 L truck engine, and a six-cylinder 7.2 L military engine. Comparison of the data in the table indicates that the performance characteristics of piston engines are remarkably similar when scaled to be size independent. As Table 1.1 illustrates, the mean piston speed is about 12 m/s, the bmep

Table 1.1 Performance Comparison of Three Different Four-Stroke Turbocharged Diesel Engines

Parameter	1.9 L Automobile	5.9 L Truck	7.2 L Military
# Cylinders	4	6	6
Bore (mm)	82	102	110
Stroke (mm)	90	120	127
Displacement per cylinder (L)	0.475	0.983	1.20
Power (kW)	110	242	222
Mass (kg)	200	522	647
Engine speed (rpm)	4000	3200	2400
Mean piston speed (m/s)	12.05	12.78	10.16
Bmep (bar)	17.3	15.4	15.4
Power/Volume (kW/L)	57.9	41.0	30.8
Mass/Volume (kg/L)	105	88	90
Power/Mass (kW/kg)	0.55	0.46	0.35

is about 15 bar, the power/volume is about 40 kW/L, and the power/mass about 0.5 kW/kg for the three engines.

There is good reason for this; all engines in a given era tend to be made from similar materials. The small differences noted could be attributed to different service criteria for which the engine was designed. Advances in engine technology have allowed manufacturers to continue to increase the power/mass. The iron in engine blocks and cylinder heads has been replaced by aluminum, which has half the weight of iron, and intake manifolds are now made of composite materials. With turbocharging, engines for vehicles have also become smaller, with four- and six-cylinder engines replacing six- and eight-cylinder engines, respectively.

Since material stresses in an engine depend to a first order only on the bmep and mean piston speed, it follows that for the same stress limit imposed by the material, all engines should have the same bmep and mean piston speed. Finally, since the engines geometrically resemble one another independent of size, the mass per unit displacement volume is more or less independent of engine size.

1.5 ENGINE CONFIGURATIONS

Internal combustion engines can be built in many different configurations. For a given engine, using a four- or two-stroke Otto or Diesel cycle, the configurations are characterized by the piston-cylinder geometry, the inlet and exhaust valve geometry, the use of super or turbochargers, the type of fuel delivery system, and the type of cooling system. The reciprocating piston-cylinder combination remains the dominant configuration of the internal combustion engine.

Since the invention of the internal combustion engine, many different piston-cylinder geometries have been designed, as shown in Figure 1.13. The choice of a given arrangement depends on a number of factors and constraints, such as engine balancing and available volume. The in-line engine is the most prevalent because it is the simplest to manufacture and maintain. The V engine is formed from two in-line banks of cylinders set at an angle to each other, forming the letter V. A horizontally opposed or flat engine is a V engine with 180° offset piston banks. The W engine is formed from three in-line banks of cylinders set at an angle to each other, forming the letter W. A radial engine has all of the cylinders in one plane with equal spacing between cylinder axes. Radial engines are used in air-cooled aircraft applications since each cylinder can be cooled equally. Since the cylinders are in a plane, a

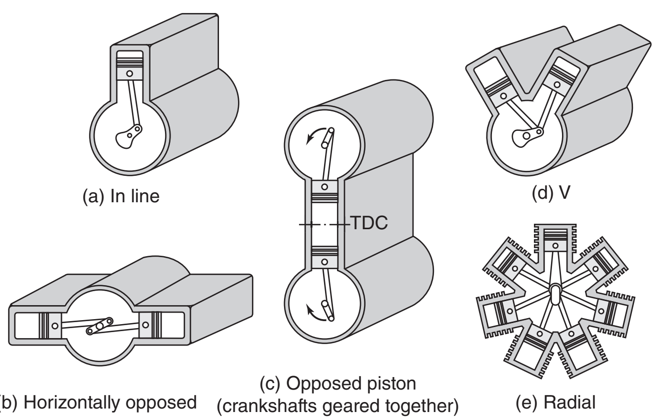

Figure 1.13 Various piston-cylinder geometries. (Adapted from Obert 1950.)

master connecting rod is used for one cylinder, and articulated rods are attached to the master rod. Alternatives to the reciprocating piston-cylinder arrangement have also been developed, such as the rotary Wankel engine, in which a triangular shaped rotor rotates eccentrically in a housing to achieve compression, ignition, and expansion of a fuel–air mixture.

Intake and Exhaust Valve Arrangement

Gases are admitted and expelled from the cylinders by valves that open and close at the proper times, or by ports that are uncovered or covered by the piston. There are many design variations for the intake and exhaust valve type and location. Poppet valves (see Figure 1.14) are the primary valve type used in internal combustion engines since they have excellent sealing characteristics. As shown in the pushrod configuration of Figure 1.14, springs are used to return the valve to a closed position. Sleeve and rotary valves have also been used, but do not seal the combustion chamber as well as poppet valves.

The poppet valves can be located either in the engine block or in the cylinder head, depending on airflow, cooling, and manufacturing considerations. Older engines and small four-stroke engines have the inlet and exhaust valves located in the block parallel to the cylinders, a configuration termed under-head or L-head. This configuration provides good cooling to the valves from the engine block coolant, however with undersquare (bore < stroke) engines the maximum valve diameter is limited, resulting in poor volumetric efficiency. The F-head configuration positions the intake valve in the cylinder head just above the cylinder, increasing the volumetric efficiency, with the exhaust valve remaining on the side.

Currently, most engines use valves located in the cylinder head, an overhead or I-head configuration, as this configuration has allows increased valve diameter resulting in good inlet and exhaust flow characteristics. However, overhead valves are more difficult to cool than L-head valves.

The valve timing is controlled by a camshaft that rotates at half the engine speed for a four-stroke engine. Lobes on the camshaft along with lifters, pushrods, and rocker arms control the valve motion. The inlet valves in early (circa 1910) engines were spring loaded,

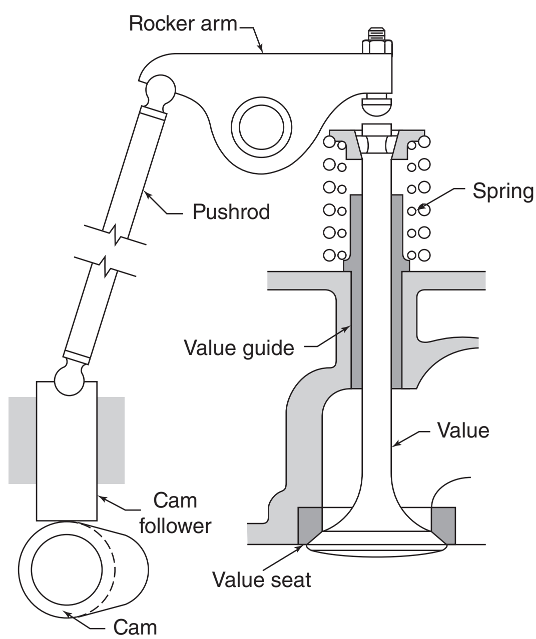

Figure 1.14 Poppet valve assembly. (Adapted from Taylor 1985.)

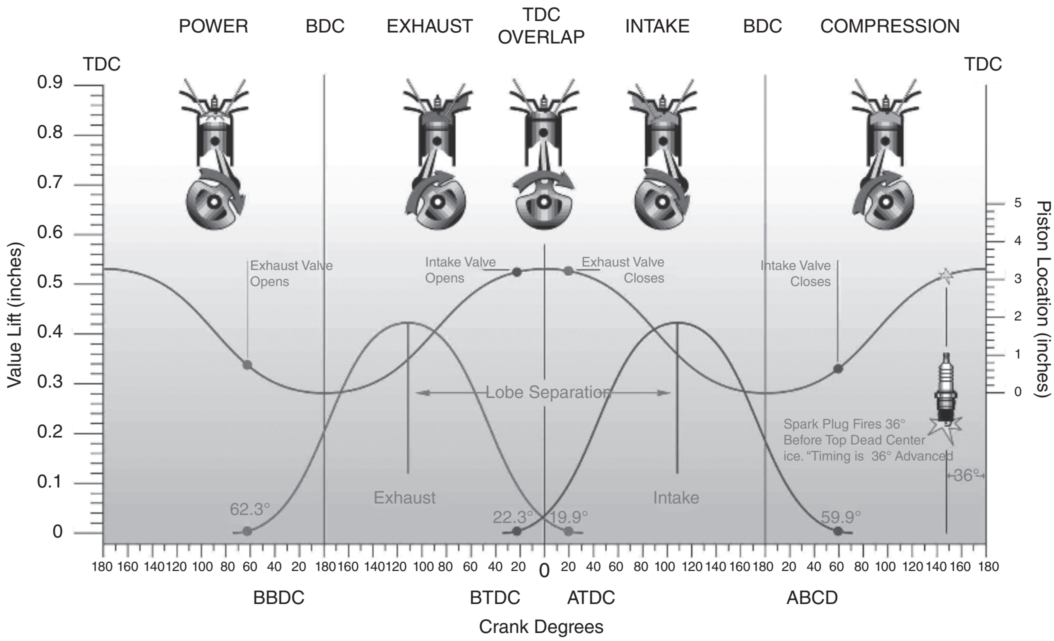

Figure 1.15 Poppet valve timing profile. (Courtesy of Competition Cams, Inc.)

and were opened during the inlet stroke by the atmosphere-cylinder pressure differential. Most automotive engines currently use an overhead camshaft to eliminate pushrods and simplify the valve train.

A valve timing profile is shown in Figure 1.15. The valve opening and closing angles are not necessarily symmetric about top and bottom dead center, due to fluid flow considerations discussed in Chapter 5. The valve timing can be varied to increase volumetric efficiency through the use of advanced camshafts that have moveable lobes, or with electric valves. With a change in the load and speed, the valve opening duration and timing can be adjusted to optimize power and/or efficiency.

Superchargers and Turbochargers

All the engines discussed so far are naturally aspirated, i.e., as the intake gas is drawn in by the downward motion of the piston. Engines can also be supercharged or turbocharged. Supercharging is mechanical compression of the inlet air to a pressure higher than standard atmosphere by a compressor powered by the crankshaft. The compressor increases the density of the intake air so that more fuel and air can be delivered to the cylinder to increase the power. The concept of turbocharging is illustrated in Figure 1.16. Exhaust gas leaving an engine is further expanded through a turbine that drives a compressor. The benefits are twofold: (1) the engine is more efficient because energy that would have otherwise been wasted is recovered from the exhaust gas; and (2) a smaller engine can be constructed to produce a given power because it is more efficient and because the density of the incoming charge is greater. The power available to drive the compressor when turbocharging is a nonlinear function of engine speed such that at low speeds there is little, if any, boost (density increase), whereas at high speeds the boost is maximum. It is also low at part throttle and high at wide-open throttle. These are desirable characteristics for an automotive engine since throttling or pumping losses are minimized. Most large and medium-size diesel engines are turbocharged to increase their efficiency. With the anticipated adoption

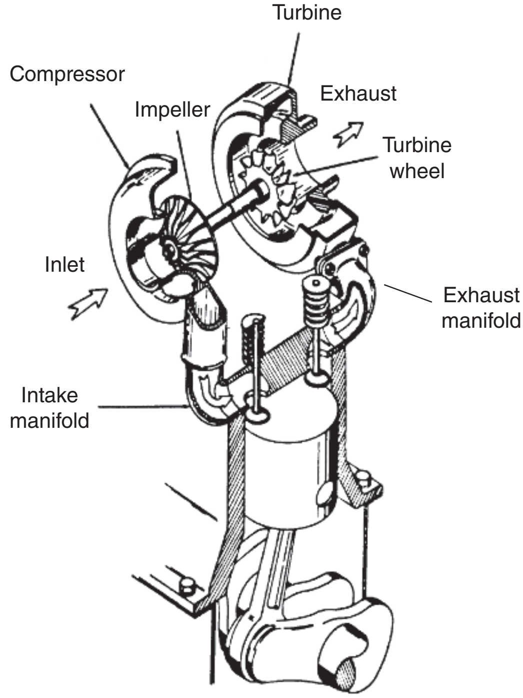

Figure 1.16 Turbocharger schematic. (Courtesy of Schwitzer.)

of 48V electrical systems in vehicles, there will be increased use of electrically powered superchargers used in conjunction with a turbocharger to reduce ”turbo lag.”

Fuel Injectors and Carburetors

Revolutionary changes have taken place with engine controls and fuel delivery systems in recent years and the progress continues. The ignition and fuel injection systems of the engine are now controlled by computers. Conventional carburetors in automobiles were replaced by throttle body fuel injectors in the 1980s, which in turn were replaced by port fuel injectors in the 1990s. Port fuel injectors are located in the intake port of each cylinder just upstream of the intake valve, so there is an injector for each cylinder. The port injector does not need to maintain a continuous fuel spray, since the time lag for fuel delivery is much less than that of a throttle body injector.

Direct injection spark-ignition engines are available on many production engines. With direct injection, the fuel is sprayed directly into the cylinder during the late stages of the compression stroke. Compared with port injection, direct injection engines can be operated at a higher compression ratio, and therefore will have a higher theoretical efficiency, since the combustion knock limitations are reduced. They can also be unthrottled, resulting in a greater volumetric efficiency at part load. The evaporation of the injected fuel in the combustion chamber will have a charge cooling effect, which will also increase volumetric efficiency.

Cooling Systems

Some type of cooling system is required to remove the approximately 30% of the fuel energy rejected as waste heat. Liquid and air cooling are the two main types of cooling

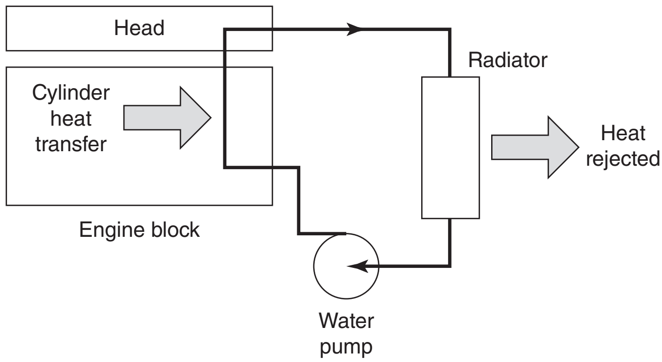

Figure 1.17 Liquid cooling system schematic.

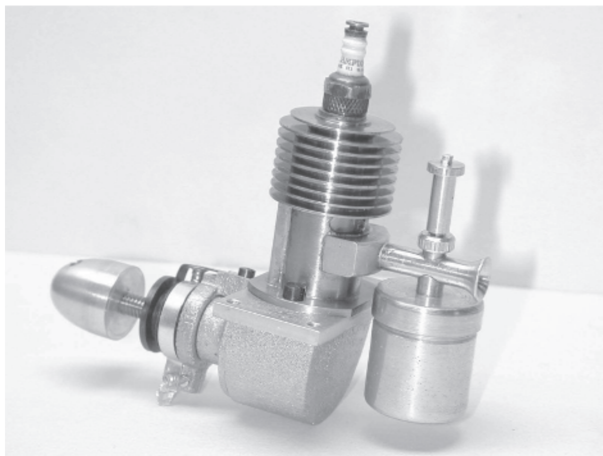

Figure 1.18 Air cooling of model airplane engine. (Courtesy R. Schroeder.)

systems. The liquid cooling system (see Figure 1.17) is usually a single loop where a water pump sends coolant to the engine block, and then to the head. Warm coolant flows through the intake manifold to warm it and thereby assist in vaporizing the fuel. The coolant will then flow to a radiator or heat exchanger, reject the waste heat to the atmosphere, and flow back to the pump. When the engine is cold, a thermostat prevents coolant from returning to the radiator, resulting in a more rapid warmup of the engine. Liquid-cooled engines are quieter than air-cooled engines, but have leaking, boiling, and freezing problems. Engines with relatively low power output, less than 20 kW, primarily use air cooling. Air-cooling systems use fins to lower the air side surface temperature (see Figure 1.18). There are historical examples of combined water and air cooling. The Mors, an early 1920s automobile, had a finned air-cooled cylinder and water-cooled heads.

1.6 EXAMPLES OF INTERNAL COMBUSTION ENGINES

Automotive Spark-Ignition Four-Stroke Engine

A photograph of a V-6 3.2 L automobile engine is shown as in Figure 1.19 and in cutaway view in Figure 1.20. The engine has a 89 mm bore and a stroke of 86 mm. The maximum power is 165 kW (225 hp) at 5550 rpm. The engine has a single overhead camshaft per piston bank with four valves per cylinder. The pistons are flat with notches for valve clearance.

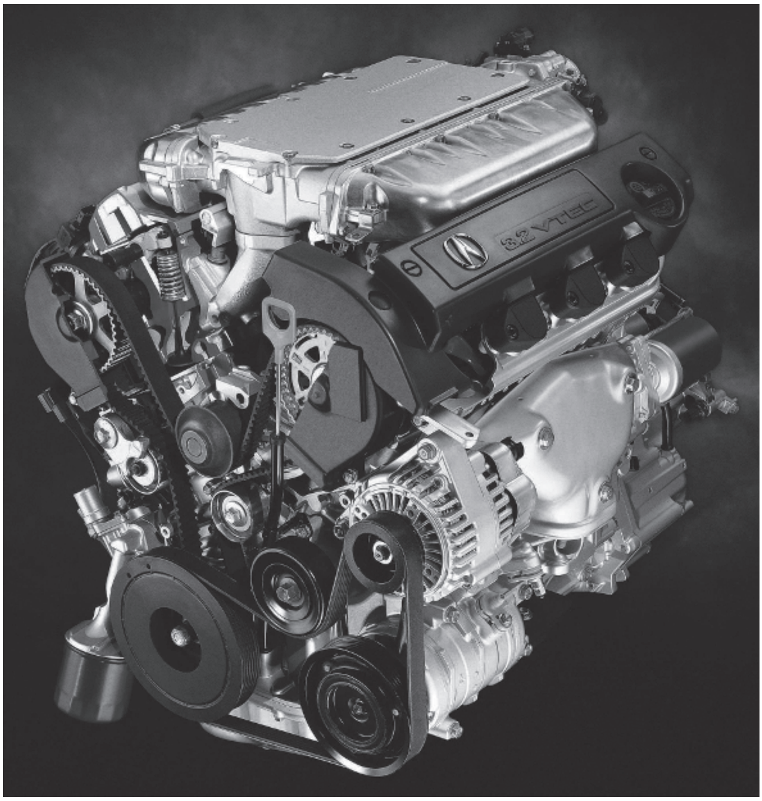

Figure 1.19 3.2 L V-6 automobile engine. (Courtesy of Honda Motor Co.)

The fuel is mixed with the inlet air by spraying the fuel into the intake port at the Y-junction just above the intake valves.

As shown in Figure 1.21, the overhead camshaft acts on both the intake and exhaust valves via rocker arms. The engine has variable valve timing applied to the intake valves with a shift from low-lift short duration cam lobes to high-lift long duration cam lobes above 3500 rpm. In the low-lift short duration cam operation the two intake valves have staggered timing, which creates additional swirl to increase flame propagation and combustion stability. Roller bearings are used on the rocker arms to reduce friction. The clearance volume is formed by an angled pent roof in the cylinder head, with the valves also angled.

Heavy-Duty Truck Diesel Engine

A heavy-duty truck diesel engine is shown in Figures 1.22. This engine is an inline six-cylinder turbocharged diesel engine with a 137-mm bore and 165-mm stroke for a total displacement of 14.6 L. The rated engine power is 373 kW (500 hp). The compression ratio is 16.5 to 1. The engine has electronically controlled, mechanically actuated fuel injectors, and an overhead camshaft. Note that the cylinder head is flat, with the diesel fuel injector mounted in the center of the combustion chamber. The inlet ports impart a swirl to the air in the combustion chamber to improve mixing with the radial fuel spray.

The top of the piston has a torus-shaped crater bowl, so that the initial combustion will take place in the piston bowl. The injection nozzles have three to six holes through which the fuel sprays into the piston bowl. The pressure required to spray the diesel fuel into the combustion chamber is of the order of 1000 bar, for adequate spray penetration into the

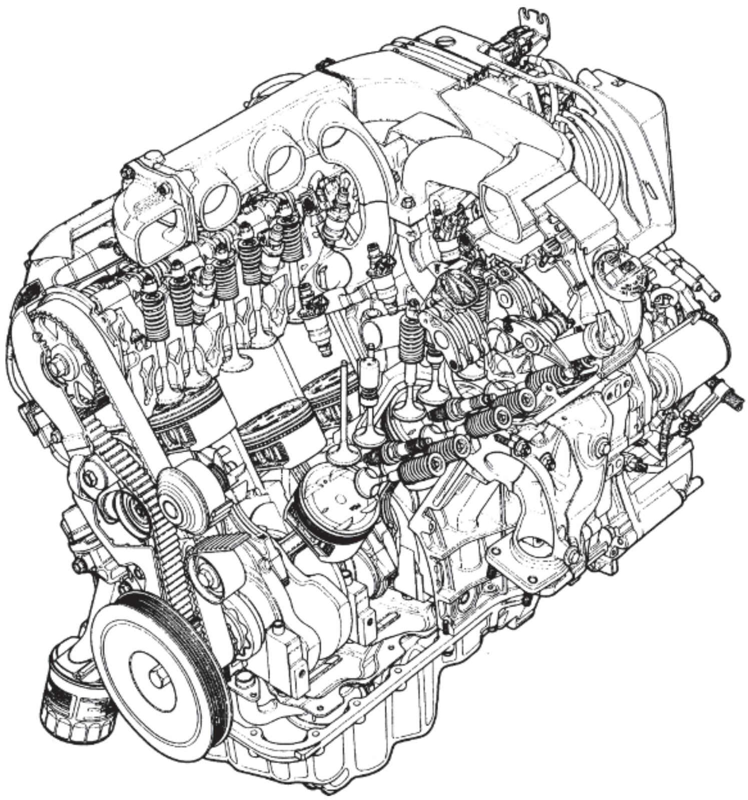

Figure 1.20 Cutaway view of 3.2 L V-6 automobile engine. (Courtesy of Honda Motor Co.)

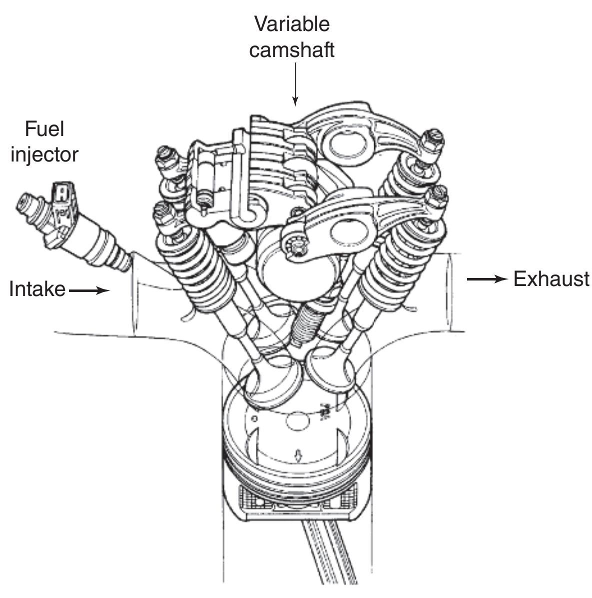

Figure 1.21 A variable valve timing mechanism. (Courtesy of Honda Motor Co.)

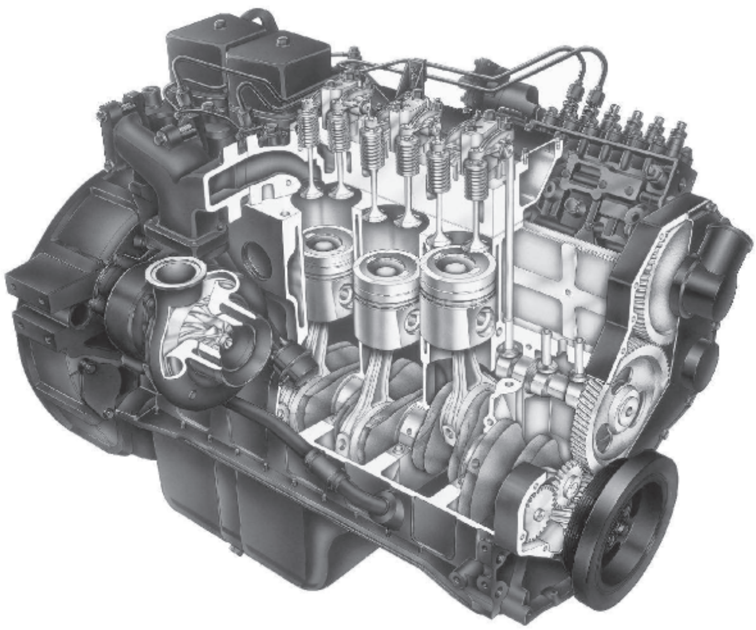

Figure 1.22 A 5.9 L L6 on-highway diesel engine. (Courtesy of PriceWebber.)

bowl and subsequent atomization of the diesel fuel. The fuel injection pressure is generated by a plunger driven by the camshaft rocker arm.

Stationary Gas Engine

A stationary natural gas engine is shown in Figures 1.23 and 1.24. Typical applications for stationary engines include co-generation, powering gas compressors, and power generation.

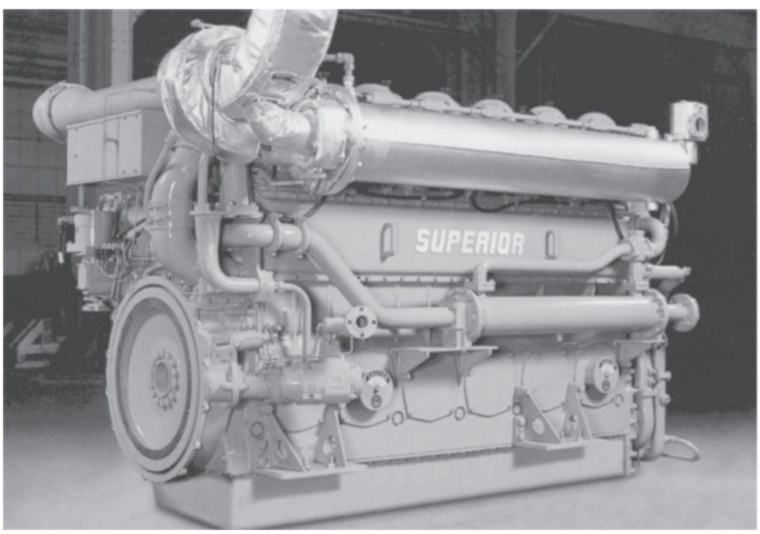

Figure 1.23 A 94 L L8 stationary natural gas engine. (Courtesy of Cooper Energy Services, Inc.)

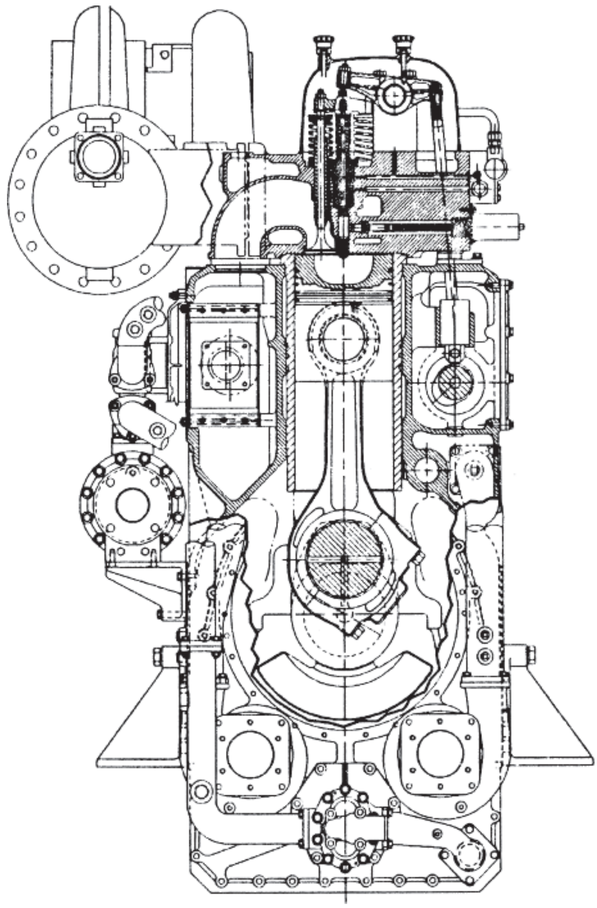

Figure 1.24 Cutaway view of 94 L L8 stationary natural gas engine. (Courtesy of Cooper Energy Services, Inc.)

The engine shown in Figure 1.23 is an in-line eight-cylinder turbocharged engine, with rated power of 1200 kW, bore of 240 mm, and stroke of 260 mm for a total displacement of 94 L. The compression ratio is 10.9:1. This type of engine is designed to operate at a constant speed condition, typically 1200 rpm. Each cylinder has two intake and two exhaust valves. The piston has a combustion bowl with a deep dish concentrated near the center of the piston, so most of the clearance volume is in the piston bowl.

Since natural gas engines are operated lean to reduce nitrogen oxides (NO_x), prechambers are used to initiate a stable combustion process. Pressurized natural gas is injected into a prechamber above the piston, and a spark plug in the prechamber is used to ignite the natural gas. The increase in pressure projects the burning mixture into the main combustion chamber, where the final stages of the combustion take place. Prechambers are also used in high-speed diesel engines to achieve acceptable mixing and more complete combustion.

1.7 ALTERNATIVE POWERTRAIN TECHNOLOGY

In this section, alternative powertrain technology, including electric motors, fuel cells, and gas turbines, are discussed in terms of a particular application where they have some advantage over the internal combustion engine.

Electric Motors

Electric motors compete with internal combustion engines in the range of powers less than about 500 kW. Driven by the need to adopt low-carbon technology both for CO_2 emission reduction and for improvement of outdoor air quality, the pace of change in vehicle electrification has been far faster than envisioned. Vehicle manufacturers are moving from viewing electric motors as a secondary or back-up source toward adopting electric motors as the primary power source and internal combustion engines as the secondary power source.

Electric vehicles have a number of advantages over internal combustion vehicles. Electric vehicles are quiet, have lower vibration levels, and cost less to operate, about 1 cent per mile versus 10 cents per mile for internal combustion vehicles. Electric motors have been developed that have high torque-speed characteristics superior to those of internal combustion engines, and also provide up to 150 kW per wheel. Most of the electric motors currently used in hybrid and electric vehicles are brushless DC motors, with rotor-mounted permanent magnets. However, use of AC induction motors, in which the rotating magnetic field is produced by electric currents in the stator, is increasing due to their lower cost, and less complex incorporation into the engine powertrain. Electric motor performance maps that contain contours of motor efficiency on a torque-speed plane are used to choose electric motors for vehicular applications.

Proponents of electric vehicles point out that almost any fuel, solar photovoltaic panels, or wind turbines can be used to generate the electricity used by an electric vehicle, reducing dependence on fossil fuels. There would be no local fossil fuel exhaust emissions emitted by the electric vehicle in an urban environment. However, if the electricity is generated by a power plant using coal as a fuel, the air pollution generated by the coal power plant would negate the air-quality advantage of the electric vehicle.

The main problem with electric vehicles is the batteries used for energy storage. It is generally recognized that a breakthrough in battery technology is required if electric vehicles are to become a significant part of the automotive fleet. Battery packs for vehicles are generally assembled from groups of individual lithium ion batteries, with a total mass of about 3500 kg, and have a life span of about 5 years. The battery pack capacity for automobiles varies from about 25–100 kWh, and fully electric urban buses are equipped with batteries with capacities from 600 to 1000 kWh. The electric vehicles that have been built to date have a limited range of only 100–200 mi (160–320 km), on the order of one-half of what can be easily realized with a gasoline engine–powered vehicle.

In addition, the volumetric energy density of a lithium ion battery is currently 0.5–1 MJ/L (150–300 Wh/L), with energy densities of the order of 5 MJ/L under development, significantly lower than gasoline or diesel fuels with energy densities of the order of 35 MJ/L. In cold weather, there is a degradation of battery performance of the order of 15-30%. A nationwide network of high voltage (240 to 950 V) charging stations is needed to compensate for the limited range of electric vehicles. The charging time for electric car batteries is at least two hours, depending on the charging station voltage, in comparison to a liquid fuel refueling time of the order of a few minutes.

Hybrid electric vehicles (HEV), which incorporate a small internal combustion engine with an electric motor and storage batteries, have reached the production stage, primarily due to their low fuel consumption and emission levels. A hybrid electric vehicle has an internal combustion engine to provide the energy to meet vehicle range requirements. The battery then provides the additional power needed for acceleration and climbing hills. The battery in an HEV vehicle typically has a capacity of about 50 MJ (14 kWh). Hybrid electric vehicles have a long history, as the first HEV, the Woods Dual Power automobile, was introduced in 1916. A similar engine-motor-battery combination has been used on diesel-electric submarines since 1900 to allow both surface and underwater operation.

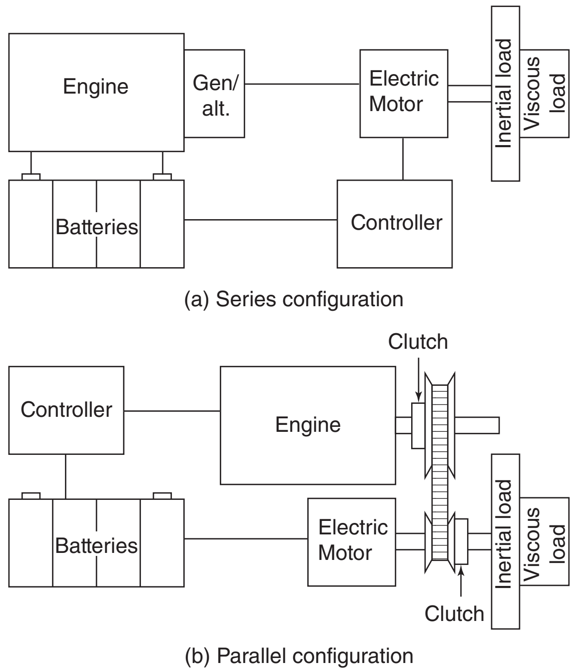

Figure 1.25 Hybrid electric vehicle powertrain configurations.

As shown in Figure 1.25, two elementary configurations for an HEV are series or parallel configurations. In a series configuration, only the electric motor with power from the battery or generator is used to drive the wheels. The internal combustion engine is maintained at its most efficient and lowest emission operating points to run the generator and charge the storage batteries. With the parallel configuration, the engine and electric motor can be used separately or together to power the vehicle. Some hybrid vehicles use an internal combustion engine to power the front wheels and an electric motor to power the rear wheels, and they synchronize them to provide all-wheel drive capability. The motors are used as generators during braking to increase vehicle efficiency.

Fuel Cells

A fuel cell converts the chemical energy in a fuel directly to electricity through electrochemical reactions. The first fuel cell was invented by W. Grove, an English scientist, in 1838. For vehicular applications, hydrogen is used as the fuel, and oxygen is the oxidizing agent. Fuel cell technology competes well in applications requiring reduced emissions, as recent developments in polymer-electrolyte membrane (PEM) technology indicate that a PEM fuel cell produces much lower CO_2 emissions relative to an internal combustion engine. A PEM fuel cell operates with hydrogen supplied to the anode, and oxygen supplied to the cathode.

Both the anode and cathode are composed of platinum particles embedded in a substrate surface of porous carbon. At the anode, the hydrogen is split into protons H^+ and electrons e^- in the reaction $H_2 \longrightarrow 2H^+ + 2e^-$. The protons migrate through a polymer electrolyte to the cathode, and the electrons provide the required current across the external load circuit, such as a battery. At the cathode, the oxygen reacts with the protons and electrons to form water, in the reaction $\frac{1}{2}O_2 + 2H^+ + 2e^- \longrightarrow H_2O$.

Current PEM fuel cell stacks are small enough to fit beneath a vehicle's floor next to the storage batteries and currently can deliver up to 125 kW to an electric motor. Studies indicate that the best opportunities for fuel cell adoption are in the commercial vehicle market, i.e., trucks and off-highway applications. Since there is presently no hydrogen fuel storage infrastructure, one option is on-board reforming of methanol fuel to hydrogen and CO_2. The reforming efficiency is about 60%, so coupled with a fuel cell efficiency of 70%, and a motor efficiency of 90%, the overall fuel cell engine efficiency is of the order of 40%, about the same as a high-efficiency internal combustion engine.

Gas Turbines

Gas turbine engines compete with internal combustion engines on the other end of the power spectrum, at powers greater than about 500 kW. The advantages offered depend on the application. Factors to consider are the efficiency and power per unit weight. A gas turbine consists basically of a compressor-burner-turbine combination that provides a supply of hot, high-pressure gas. This may then be expanded through a nozzle (turbojet), through a turbine, to drive a fan, and then through a nozzle (turbofan), through a turbine, to drive a propeller (turboprop), or through a turbine to spin a shaft in a stationary or vehicular application.

One advantage a gas turbine engine offers to the designer is that the hardware responsible for compression, combustion, and expansion are three different devices, whereas in a piston engine all these processes are done within the cylinder. The hardware for each process in a gas turbine engine can then be optimized separately; whereas in a piston engine compromises must be made with any given process, since the hardware is expected to do three tasks. However, it should be pointed out that turbochargers give the designer of conventional internal combustion engines some new degrees of freedom toward optimization.

With temperature limits imposed by materials, the reciprocating engine can have a greater peak cycle temperature than the gas turbine engine. In an internal combustion engine, the gases at any position within the engine vary periodically from hot to cold. Thus the average temperature during the heat transfer to the walls is neither very hot nor cold. On the other hand, the gas temperature at any position in the gas turbine is steady, and the turbine inlet temperature is always very hot, thus tending to heat material at this point to a greater temperature than anywhere in a piston engine.

The thermal efficiency of a gas turbine engine is highly dependent on the adiabatic efficiency of its components, which in turn is highly dependent on their size and their operating conditions. Large gas turbines tend to be more efficient than small gas turbines. That airliners are larger than automobiles is one reason gas turbines have displaced piston engines in airliners, but not in automobiles. Likewise gas turbines are beginning to penetrate the marine industry, though not as rapidly, as power per unit weight is not as important with ships as with airplanes.

Another factor favoring the use of gas turbines in airliners (and ships) is that the time the engine spends operating at part or full load is small compared to the time the engine spends cruising, therefore the engine can be optimized for maximum efficiency at cruise. It is a minor concern that at part load or at take-off conditions the engine's efficiency is compromised. Automobiles, on the other hand, are operated over a wide range of load and speed so a good efficiency at all conditions is better than a slightly better efficiency at the most probable operating condition and a poorer efficiency at all the rest.

Steam- or vapor-cycle engines are much less efficient than internal combustion engines, since their peak temperatures are about 800 K, much lower than the peak temperatures ($\approx 2500K$ of an internal combustion engine. They are used today almost totally in stationary applications and where the energy source precludes the use of internal combustion engines. Such energy sources include coal, waste feed stocks, nuclear, solar, and waste heat in the exhaust gas of combustion devices including internal combustion engines.

In some applications, engine emission characteristics might be a controlling factor. In the 1970s, in fact, a great deal of development work was done toward producing an automotive steam engine when it was not known whether the emissions from the internal combustion engine could be reduced enough to meet the standards dictated by concern for public health. However, the development of catalytic converters, as discussed in Chapter 9, made it possible for the internal combustion engine to meet emission standards at that time, and remain a dominant prime mover technology.

1.8 FURTHER READING

The references of this introductory chapter contain a listing of both historical and current books that will provide additional information about internal combustion engine design, analysis and performance. These books will give the reader a deeper appreciation of how much the technology of internal combustion engines has advanced in the last century. In chronological order, these books are: Clerk (1910), Ricardo (1941), Benson and Whitehouse (1979), Cummins (1989), Arcoumanis (1988), Lumley (1999), Pulkrabek (2003), Shi et al. (2011), Stone (2012), and Heywood (2018).

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1.10 HOMEWORK

1.1 Compute the mean piston speed, bmep (bar), torque (Nm), and the power per piston area for the engines listed in Table 1.2.

1.2 A six-cylinder, two-stroke airplane engine with a compression ratio $r = 9$ produces a torque of 1100 Nm at a speed of 2100 rpm. It has a bore b of 123 mm and a stroke s of 127 mm. (a) What is the displacement volume V_d and the clearance volume V_c of a cylinder? (b) What is the engine bmep, brake power, and mean piston speed?

Table 1.2 Engine Data for Homework Problems

Engine	Bore (mm)	Stroke (mm)	Cylinders	Speed (rpm)	Power (kW)
Marine	136	127	12	2600	1118
Truck	108	95	8	6400	447
Airplane	86	57	8	10500	522

1.3 A four-cylinder, 2.5 L four-stroke spark ignited engine is mounted on a dyno and operated at a speed of $N = 3000$ rpm. The engine has a compression ratio of 10:1 and mass air–fuel ratio of 15:1. The inlet air manifold conditions are 80 kPa and 313 K. The engine produces a torque of 160 Nm and has a volumetric efficiency of 0.82. (a) What is the brake power $\dot{W}_b$ (kW)? (b) What is the brake specific fuel consumption bsfc (g/kWh)? (c) What is the brake work W_b (kJ) per cylinder per revolution?

1.4 The volumetric efficiency of the fuel injected marine engine in Table 1.2 is 0.80 and the inlet manifold density is 50% greater than the standard atmospheric density of $\rho_{amb} = 1.18$ kg/m^3. If the engine speed is 2600 rpm, what is the inlet air mass flowrate $\dot{m}_a$ (kg/s)?

1.5 A 380 cc single-cylinder, two-stroke motorcycle engine is operating at 5500 rpm. The engine has a bore of 82 mm and a stroke of 72 mm. Performance testing gives a bmep = 6.81 bar, bsfc = 0.49 kg/kWh, and delivery ratio of 0.748. What is the fuel–air ratio, *FA*? Assume standard atmospheric conditions of 298 K and 101.3 kPa.

1.6 A 3.8 L four-stroke, four-cylinder fuel-injected automobile engine, with an equal bore and stroke and a compression ratio of 10:1, has a power output of 88 kW at 4000 rpm and volumetric efficiency of 0.85. The bsfc is 0.35 kg/kWh. (a) What is the bore b and V_{bdc}? (b) If the fuel has a heat of combustion of 42,000 kJ/kg, what are the bmep, thermal efficiency, and air–fuel ratio, *AF*? Assume standard atmospheric conditions of 298 K and 101.3 kPa.

1.7 A 4.0 L six-cylinder automobile engine is operating at 3000 rpm. The engine has a compression ratio of 10:1, and volumetric efficiency of 0.85. If the bore and stroke are equal, (a) what is the stroke s, (b) the mean piston speed $\bar{U}_p$, (c) the cylinder clearance volume V_c, and (d) the inlet air mass flowrate $\dot{m}_a$? Assume standard inlet air conditions of 298 K and 101.3 kPa.

1.8 A 10.0 L, eight-cylinder square four-stroke truck engine has a brake mean effective pressure of 11 bar, and operates at 2500 rpm with a volumetric efficiency of 0.85. Assume inlet air conditions of 298 K and 1 bar. (a) What is the total mass air flowrate into the engine, (b) the brake power produced by the engine, (c) the bore, and (d) the mean piston speed?

1.9 Chose an automotive, marine, or aviation engine of interest, and compute the engine's mean piston speed, bmep, and power/volume. Compare your calculated values with those presented in Table 1.1.

1.10 Compare the approximate, Equation (1.36), and exact, Equation (1.33), dimensionless cylinder volume versus crank angle profiles for $r = 8$, $s = 100$ mm, and $l = 150$ mm. What is the maximum error, and at what crank angle does it occur?

1.11 Plot the nondimensional piston velocity, Equation (1.37), for an engine with a stroke $s = 100$ mm and connecting rod length $l = 165$ mm. What is the maximum velocity, and at what crank angle does it occur?