Chapter 1

Introduction to Internal Combustion Engines

1.1 INTRODUCTION

The main focus of this text is on the application of the engineering sciences, especially the thermal sciences, to internal combustion engines. The goals of the text are to familiarize the reader with engine nomenclature, describe how internal combustion engines work, and provide insight into how engine performance can be modeled and analyzed. An internal combustion engine is defined as 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 combustion engine in which a separate combustor is used to burn the fuel.

In this chapter, we discuss the engineering parameters that are used to characterize the overall performance of internal combustion engines. Major engine cycles, configurations, and geometries are covered. The following chapters will apply the principles of thermodynamics, combustion, fluid flow, friction, and heat transfer to determine an internal combustion engine's temperature and pressure profiles, work, thermal efficiency, and exhaust emissions.

An aspect upon which we have put considerable emphasis is the process of constructing idealized models to represent actual physical situations in an engine. Throughout the text, we will calculate the values of the various thermal and mechanical parameters that characterize internal combustion engine operation.

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 semiempirical 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. The development of a successful engine requires knowledge of methods and analyses introduced in the text which are used to parameterize and correlate experiments, and to calculate the performance of a proposed engine design.

The internal combustion engine was invented and successfully developed in the late 1860s. It is considered as 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. For example, 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

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

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.

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 comparison of 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 20th century to values as high as 50% today.

Internal combustion engine efficiency continues to increase, driven both by legislation and the need to reduce operating costs. The primary United States 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 will reach 35.5 mpg in 2016, and 54.5 mpg by 2025. This doubling of vehicle mileage requirements will require increased 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. For example, in 1900 most automobiles were steam or electrically powered, but by 1920 most automobiles were powered by gasoline engines. As of the year 2010, in the United States alone there are about 220 million motor vehicles powered by internal combustion engines. 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

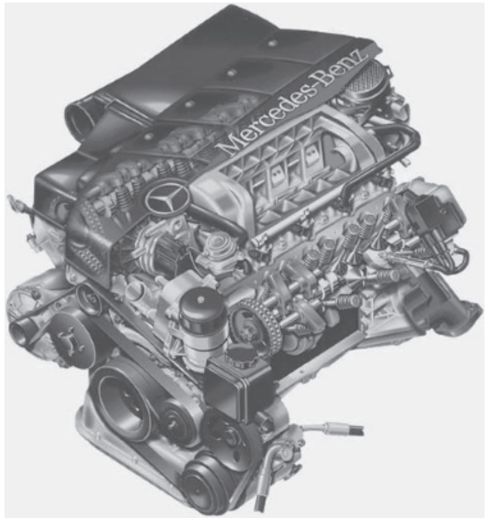

Figure 1.2 Automobile engine. (Courtesy Mercedes-Benz Photo Library.)

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 to 20×10^3 kW, depending on their displacement. They compete in the market place 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 emissions level of current internal combustion engines has decreased to about 5% of the emissions 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

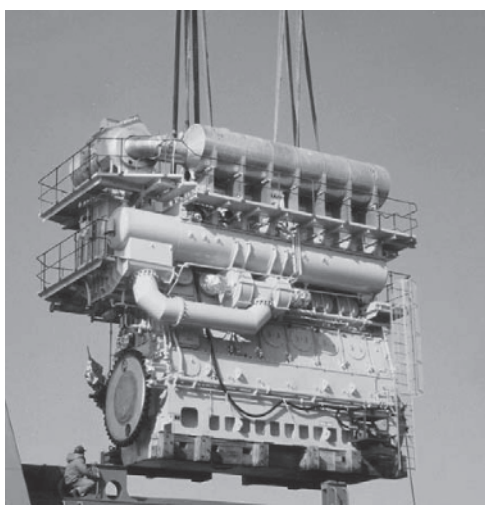

Figure 1.3 Marine engine. (Courtesy Man B&W Diesel.)

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 the source of about half of the NO_x, CO, and HC pollutants in the air. Carbon dioxide (CO_2), a primary gaseous combustion product of internal combustion engines is also a greenhouse gas, and is in the process of being regulated as well.

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, and 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 four-stroke gasoline-fueled engine in 1883. The liquid fuel was vaporized and mixed with the intake air in a carburetor before being drawn into the combustion chamber. The fuel air mixture was ignited by a flame tube. In 1886, he 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 two-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 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 L 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 170 rpm, and produced 18 hp, with 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. The first multi-cylinder diesel engines for trucks were available by 1924, and the first diesel-powered automobiles were available by 1936. 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.

These 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. For example, Henry Ford's Model T engine of 1908, and the Wright Brother's Flyer engine of 1903 used natural convection water cooling.

1.3 ENGINE CYCLES

The two major cycles currently used in internal combustion engines are termed Otto and Diesel, named after the two men credited with their invention. The Otto cycle is also known as a constant volume combustion or spark ignition cycle, and the Diesel cycle is also known as a constant pressure combustion or compression ignition 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.

Otto Cycle

As shown in Figure 1.4, the four-stroke Otto cycle has the following sequence of operations:

1. An intake stroke that 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 that raises the temperature of the mixture. A spark ignites the mixture toward the end of the compression stroke.
3. An expansion or power stroke resulting from combustion of the fuel–air mixture.
4. An exhaust stroke that pushes out the burned gases past the exhaust valve.

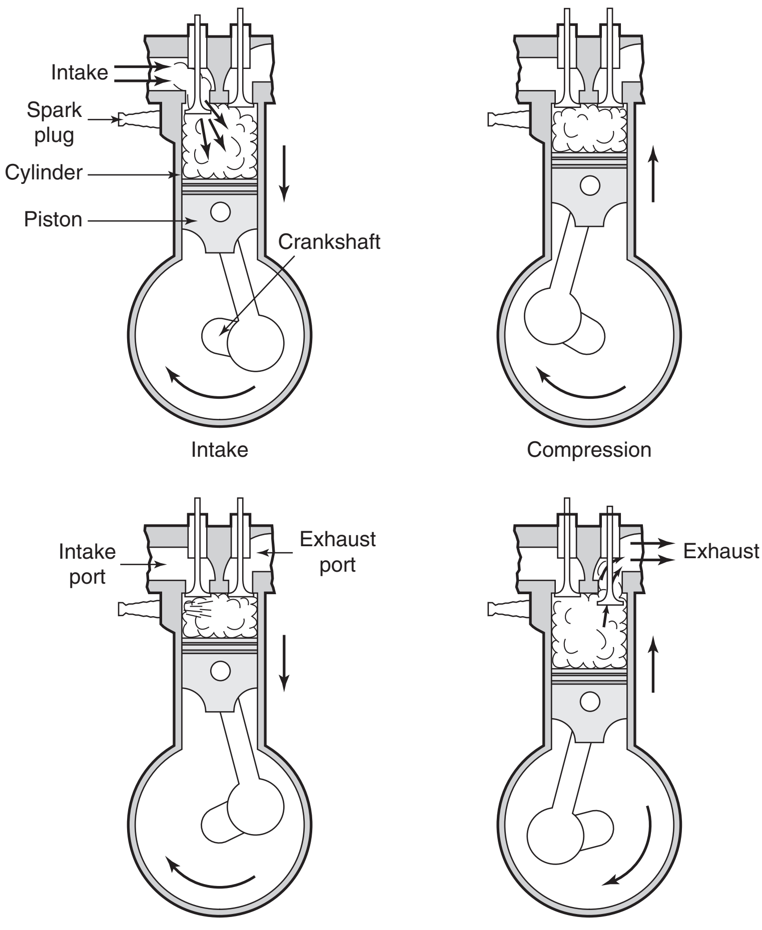

Figure 1.4 Four-stroke spark ignition cycle.

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 Otto cycle 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 Otto 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 airflow, the throttle in an Otto cycle, in essence, controls the power.

Diesel Cycle

The four-stroke Diesel cycle has the following sequence:

1. An intake stroke that draws inlet air past the intake valve into the cylinder.
2. A compression stroke that 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 during the later stages of the compression stroke and the expansion stroke.
4. An exhaust stroke that pushes out the burned gases past the exhaust valve.

There are two types of diesel combustion systems, direct injection (DI) into the main cylinder, and indirect injection (IDI) into a prechamber connected to the main cylinder. 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. Indirect injection engines tend to be used where the engine is expected to perform over a wide range of speeds and loads such as in an automobile. When the operating range of the engine is less broad such as in ships, trucks, locomotives, or electric power generation, direct injection engines predominate.

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 15–20, 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. 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 ranging from 375 to 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

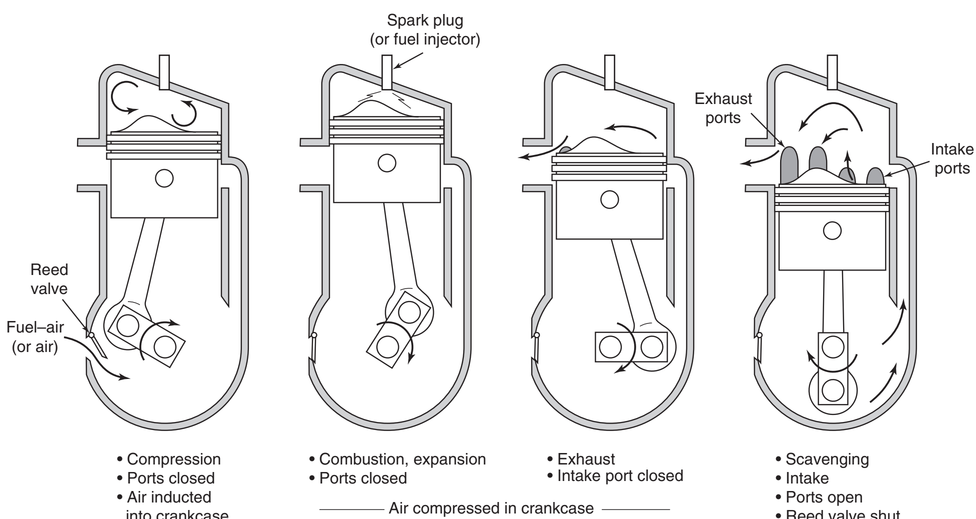

Figure 1.5 A cross-scavenged two-stroke cycle.

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 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

For any one cylinder, the crankshaft, connecting rod, piston, and head assembly can be represented by the 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; and crank angle, θ. The crank radius is one-half of the stroke. The top dead center (tdc) of an engine

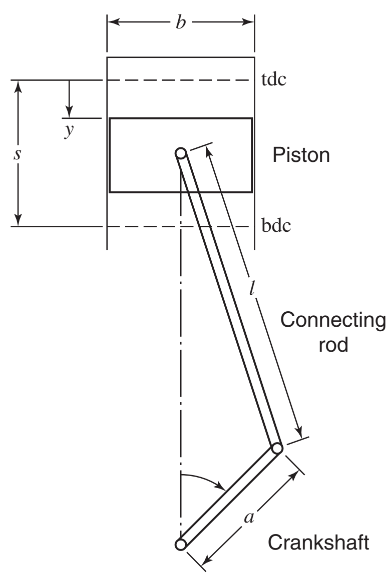

Figure 1.6 Engine slider–crank geometry.

refers to the crankshaft being in a position such that $\theta = 0°$. The cylinder volume in this position is minimum and is also called the clearance volume, V_c. Bottom dead center (bdc) refers to the crankshaft being at $\theta = 180°$. The cylinder volume at bottom dead center V_1 is maximum.

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

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

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

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

A useful expression relating V_d and V_{bdc} is

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

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

$$V_d = n_c \frac{\pi}{4} b^2 s \tag{1.4}$$

The mean piston speed $\bar{U}_p$ 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

$$\bar{U}_p = 2Ns \tag{1.5}$$

The engine speed, N, refers to the rotational speed of the crankshaft and is expressed in revolutions per minute. The engine frequency, ω, also refers to the rotation rate of the crankshaft but in units of radians per second.

Power, Torque, and Efficiency

The brake power, $\dot{W}_b$, is the rate at which work is done; and the engine torque, τ, is a measure of the work done per unit rotation (radians) of the crank. The brake power is the power output of the engine, and measured by a dynamometer. Early dynamometers were simple brake mechanisms. The brake power is less than the boundary rate of work done by the gas, called indicated power, partly because of friction. As we shall see when discussing dynamometers in Chapter 10, the brake power and torque are related by

$$\dot{W}_b = 2\pi\tau N \tag{1.6}$$

The net power is from the complete engine, whereas gross power is from an engine without the cooling fan, muffler, and tail pipe.

The indicated work W_i is the net work transferred from the gas to the piston during a cycle, which is the integral of the pressure over the cylinder volume:

$$W_i = \int P dV \tag{1.7}$$

and the indicated power $\dot{W}_i$, for an engine with n_c cylinders, is

$$\dot{W}_i = n_c W_i N/2 \quad \text{(four stroke engine)} \tag{1.8}$$

$$\dot{W}_i = n_c W_i N \quad \text{(two stroke engine)} \tag{1.9}$$

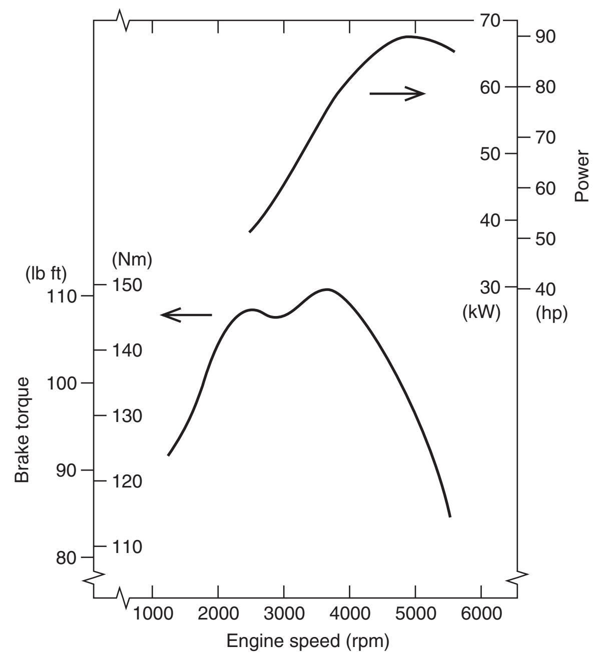

Figure 1.7 Wide open throttle (WOT) performance of an automotive four-stroke engine.

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 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, $\dot{W}_f$

$$\dot{W}_f = \dot{W}_i - \dot{W}_b \tag{1.10}$$

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

$$\eta_m = \dot{W}_b/\dot{W}_i = 1 - \dot{W}_f/\dot{W}_i \tag{1.11}$$

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) is the work done per unit displacement volume, and has units of force/area. It is the average pressure that results in the same amount of work actually produced by the engine. 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. There are three useful mean effective pressure parameters–mep, 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 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 measured bmep for naturally aspirated automobile engines depend on the load, with maximum values of about 10 bar, and greater values of about 20 bar for turbo or supercharged engines.

Based on torque, the bmep is

$$\begin{aligned} \text{bmep} &= \frac{4\pi\tau}{V_\text{d}} \quad \text{(four stroke engine)} \\ &= \frac{2\pi\tau}{V_\text{d}} \quad \text{(two stroke engine)} \end{aligned} \tag{1.12}$$

and in terms of power the bmep is

$$\begin{aligned} \text{bmep} &= \frac{\dot{W}_\text{b}}{V_\text{d}\, N/2} \quad \text{(four stroke engine)} \\ &= \frac{\dot{W}_\text{b}}{V_\text{d}\, N} \quad \text{(two stroke engine)} \end{aligned} \tag{1.13}$$

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

$$\begin{aligned} \text{bmep} &= \frac{4\dot{W}_\text{b}}{n_\text{c} A_\text{p} \bar{U}_\text{p}} \quad \text{(four stroke engine)} \\ &= \frac{2\dot{W}_\text{b}}{n_\text{c} A_\text{p} \bar{U}_\text{p}} \quad \text{(two stroke engine)} \end{aligned} \tag{1.14}$$

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.

$$\text{fmep} = \text{imep} - \text{bmep} \tag{1.15}$$

The bmep of two different displacement 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.

Volumetric Efficiency

A performance parameter of importance for four-stroke engines is the volumetric efficiency, e_v. It is defined as the mass of fuel and air inducted into the cylinder divided by the mass that would occupy the displaced volume at the density ρ_i in the intake manifold. 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

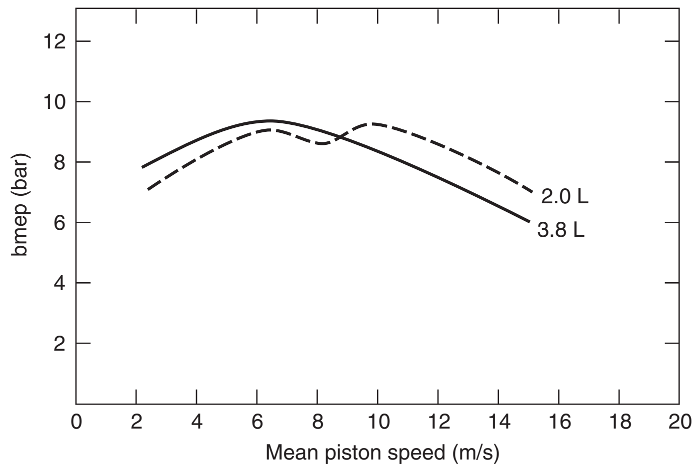

Figure 1.8 Brake mean effective pressure at WOT versus mean piston speed for two automotive engines.

cylinder. The volumetric efficiency is a mass ratio and not a volume ratio. The volumetric efficiency for an engine operating at a speed N is thus

$$e_v = \frac{\dot{m}_{in}}{\rho_i V_d N/2} \tag{1.16}$$

where

$$\dot{m}_{in} = \dot{m}_a + \dot{m}_f \tag{1.17}$$

In Equation 1.17, $\dot{m}_f$ is the flow rate of the fuel inducted in the intake manifold. For a direct injection engine, $\dot{m}_f = 0$. The factor of 2 accounts for the two revolutions per cycle in a four-stroke engine. The intake manifold density is used as a reference condition instead of the standard atmosphere, so that supercharger performance is not included in the definition of volumetric efficiency. For two-stroke cycles, a parameter related to volumetric efficiency called the delivery ratio is defined in terms of the airflow only and the ambient air density instead of the intake manifold density.

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

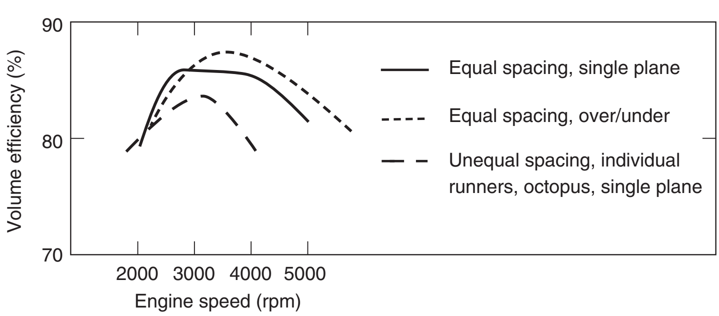

Figure 1.9 Effect of engine speed and intake manifold geometry on volumetric efficiency. Adapted from Armstrong and Stirrat (1982).

valve size, valve lift, and valve timing. It is desirable to maximize the volumetric efficiency 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 airflow rate of an engine of a given displacement and speed.

EXAMPLE 1.1 ***Volumetric efficiency***

A four-stroke 2.5 L 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 airflow rate $\dot{m}_a$ through the engine? The inlet air pressure and temperature are 75 kPa and 40°C.

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 inlet air density is

$$\rho_i = P/RT_i = 75{,}000/(287 \times 313) = 0.835\,\text{kg/m}^3$$

and the mass airflow rate $\dot{m}_a$ 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 flow rate $\dot{m}_f$, divided by the brake power $\dot{W}_b$. It has three terms that are standard measurements in an engine test: the fuel flow rate, the torque, and the engine speed:

$$\text{bsfc} = \frac{\dot{m}_f}{\dot{W}_b} = \frac{\dot{m}_f}{2\pi\tau N} \tag{1.18}$$

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.

$$\text{isfc} = \frac{m_f}{W_i} \tag{1.19}$$

Typical values of measured bsfc for naturally aspirated automobile engines depend on the engine load, with values ranging from about 200 to 400 g/kWh.

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 is then

$$\eta = \frac{\dot{W}_b}{\dot{m}_f\, q_c} = \frac{1}{\text{bsfc}\, q_c} \tag{1.20}$$

Inspection of Equation 1.20 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.

EXAMPLE 1.2 ***Engine Parameters Calculation***

A six-cylinder four-stroke automobile engine is being designed to produce 75 kW at 2000 rpm with a bsfc of 300 g/kWh and a bmep of 12 bar. The engine is to have equal bore and stroke, and fueled with gasoline with a heat of combustion of 44,510 kJ/kg. (a) What should be the design displacement volume and bore? (b) What is the mean piston speed at the design point? (c) What is the fuel consumption per cycle per cylinder? (d) What is the brake thermal efficiency?

SOLUTION (a) The displacement volume V_d is

$$V_d = \frac{\dot{W}_b}{\text{bmep}\, N/2} = \frac{75}{(1200)(2000/2)(1/60)} = 3.75 \times 10^{-3}\ \text{m}^3 = 3.75\ \text{L}$$

$$b = \left(\frac{V_d}{n_c}\frac{4}{\pi}\right)^{1/3} = \left(\frac{3.75 \times 10^{-3}}{6}\frac{4}{\pi}\right)^{1/3} = 92.7\ \text{mm}$$

Most automobile engines have approximately a 90 mm bore and stroke.

(b) The mean piston speed is

$$\bar{U}_p = 2Ns = (2)(9.27 \times 10^{-2})(2000/60) = 6.18\ \text{m/s}$$

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

$$\bar{\dot{m}}_f = \text{bsfc} \times \dot{W}_b/n_c = 300 \times 75/(6 \times 60) = 62.5\ \text{g/min}$$

so the mass of fuel injected per cylinder per cycle is

$$m_f = \bar{\dot{m}}_f/(N/2) = 62.5/(2000/2) = 6.25 \times 10^{-2}\ \text{g}$$

(d) The brake thermal efficiency is

$$\eta = \frac{1}{\text{bsfc}\, q_c} = \frac{3600}{(0.3)(44,510)} = 0.27$$

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

Scaling of Engine Performance

The performance characteristics of three different diesel engines is 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 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 tend to be made from similar materials. The small differences noted could be attributed to different service criteria for which the engine was designed. 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 form 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 1.10. 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 as 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

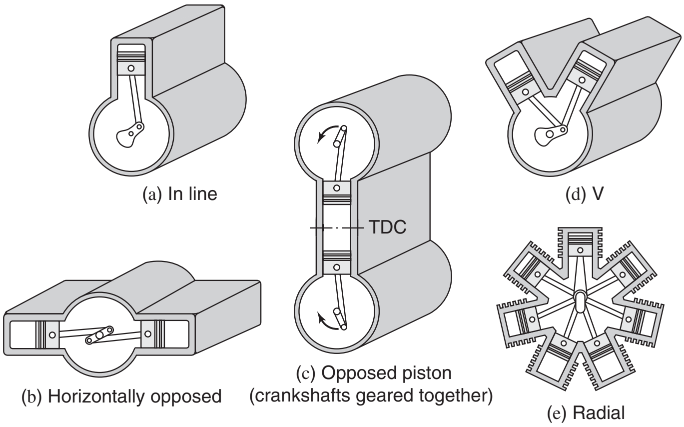

Figure 1.10 Various piston–cylinder geometries. Adapted from Obert (1950).

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 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.

Engine Kinematics

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

$$V(\theta) = V_c + \frac{\pi}{4} b^2 y \tag{1.21}$$

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

By reference to Figure 1.6

$$y = l + a - [(l^2 - a^2 \sin^2\theta)^{1/2} + a \cos\theta] \tag{1.22}$$

The instantaneous volume $V(\theta)$ can be nondimensionalized by the clearance volume at top dead center, V_{tdc}, resulting in

$$\tilde{V}(\theta) = \frac{V(\theta)}{V_{\text{tdc}}} = 1 + (r-1)\frac{y}{s} \tag{1.23}$$

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

$$\epsilon = \frac{a}{l} = \frac{s}{2l} \tag{1.24}$$

The value of ϵ for the slider–crank geometries used in modern engines is of order 1/3.

Therefore, the nondimensional piston displacement y/s is

$$\frac{y}{s} = \frac{1}{2}(1 - \cos\theta) + \frac{1}{2\epsilon}\left[1 - (1 - \epsilon^2 \sin^2\theta)^{1/2}\right] \tag{1.25}$$

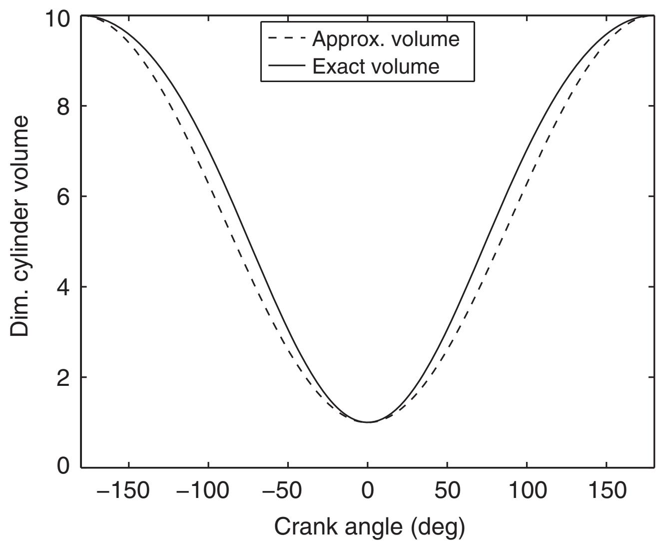

Figure 1.11 Cylinder volume versus crank angle for $r = 10, \epsilon = 1/3$ (Equations 1.26 and 1.29).

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

$$\tilde{V}(\theta) = 1 + \frac{(r-1)}{2}(1 - \cos\theta) + \frac{1}{2\epsilon}\left[1 - (1 - \epsilon^2 \sin^2\theta)^{1/2}\right] \tag{1.26}$$

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

$$(1 - \epsilon^2 \sin^2\theta)^{1/2} \simeq 1 - \frac{1}{2}\epsilon^2 \sin^2\theta + O(\epsilon^4) \tag{1.27}$$

so

$$\frac{y}{s} \simeq \frac{1}{2}(1 - \cos\theta) + \frac{\epsilon}{4}\sin^2\theta \tag{1.28}$$

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

$$\tilde{V}(\theta) \simeq 1 + \frac{(r-1)}{2}(1 - \cos\theta) \tag{1.29}$$

The cylinder volumes predicted by Equations 1.26 and 1.29 are compared in Figure 1.11 for a value of $\epsilon = 1/3$, using the Matlab® program `Volume.m` listed in Appendix F.1. Both equations give identical results at bottom dead center and top dead center, and since the second term of the expansion is relatively small, the approximate volume relation under-predicts the exact cylinder volume only by about 10% in the middle of the stroke.

The instantaneous piston velocity U_p can be found by replacing θ with ωt and differentiating Equation 1.25 with respect to time t giving

$$U_\text{p}(\omega t) = \frac{dy}{dt} = \frac{\omega s \sin(\omega t)}{2}\left[1 + \frac{\epsilon \cos\omega t}{(1 - \epsilon^2 \sin^2\omega t)^{1/2}}\right] \tag{1.30}$$

Equation 1.30 can be nondimensionalized by the mean piston speed $\bar{U}_\text{p}$, resulting in

$$\tilde{U}_\text{p}(\theta) = \frac{U_p}{\bar{U}_p} = \frac{\pi}{2}\sin\theta\left[1 + \frac{\epsilon\cos\theta}{(1 - \epsilon^2\sin^2\theta)^{1/2}}\right] \tag{1.31}$$

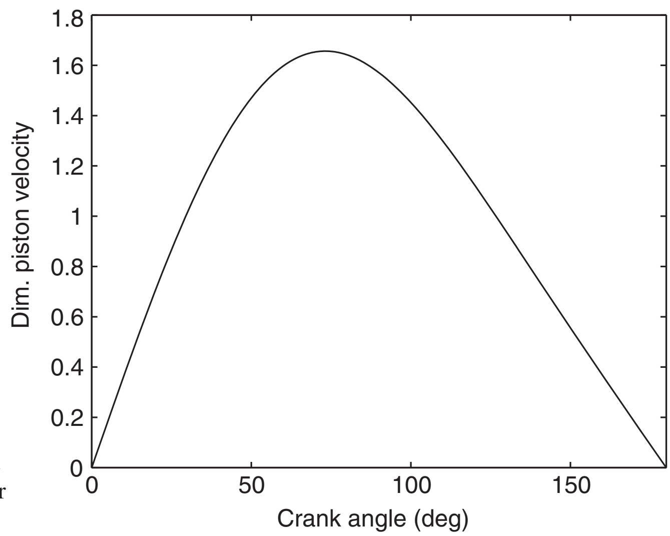

Figure 1.12 Nondimensional velocity versus crank angle for $\epsilon = 1/3$ (Equation 1.31).

Using the Matlab® program `Velocity.m` listed in the Appendix F.2, 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 and bdc. Due to the geometry of the slider–crank mechanism, the velocity profile is nonsymmetric, with the maximum nondimensional velocity of $\tilde{U}_p(\theta) = 1.65$ occurring at 72°atdc.

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

$$U_p = \frac{dy}{dt} \simeq \frac{\omega s}{2} \left[\sin \omega t + \frac{\epsilon}{2} \sin 2\omega t \right] \tag{1.32}$$

The acceleration a_p is found by differentiating Equation 1.32 with respect to time

$$a_\mathrm{p} = \frac{d^2 y}{dt^2} \simeq \frac{\omega^2 s}{2} [\cos \omega t + \epsilon \cos 2\omega t] \tag{1.33}$$

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).

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.

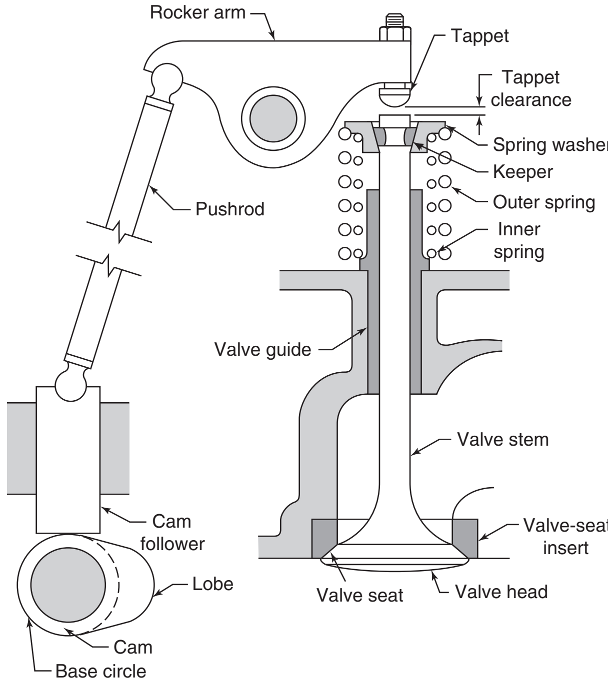

Figure 1.13 Poppet valve nomenclature (Taylor, 1985).

Poppet valves (see Figure 1.13) are the primary valve type used in internal combustion engines, since they have excellent sealing characteristics. Sleeve 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 manufacturing and cooling considerations. Older automobiles and small four-stroke engines have the valves located in the block, a configuration termed underhead or L-head. Currently, most engines use valves located in the cylinder head, an overhead or I-head configuration, as this configuration has good inlet and exhaust flow characteristics.

The valve timing is controlled by a camshaft that rotates at half the engine speed for four-stroke engine. A valve timing profile is shown in Figure 1.14. Lobes on the camshaft along with lifters, pushrods, and rocker arms control the valve motion. Some engines use an overhead camshaft to eliminate pushrods. 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, the valve opening duration and timing can be adjusted.

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 1.15. 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

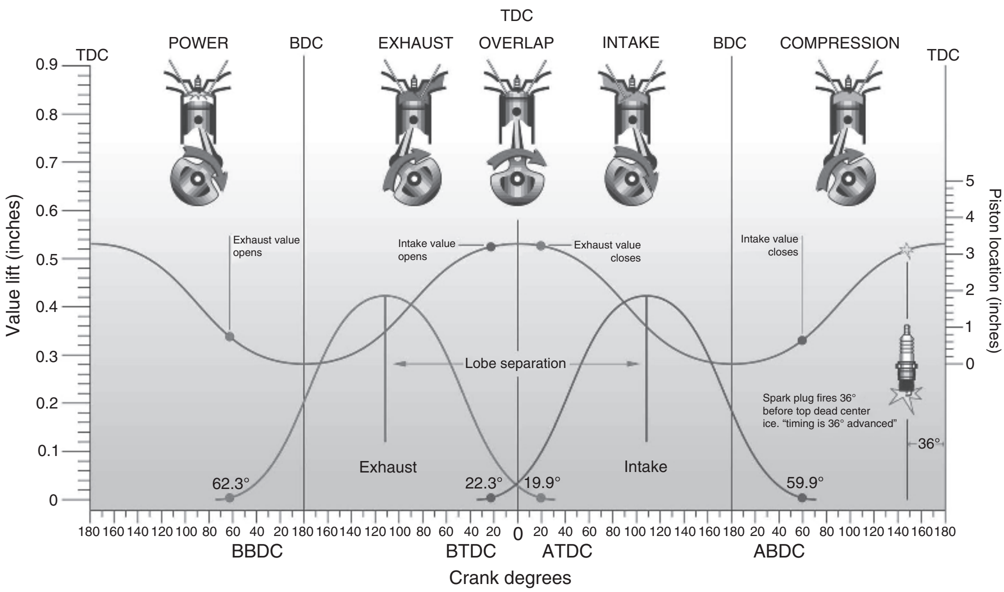

Figure 1.14 Poppet valve timing profile. (Courtesy of Competition Cams, Inc.)

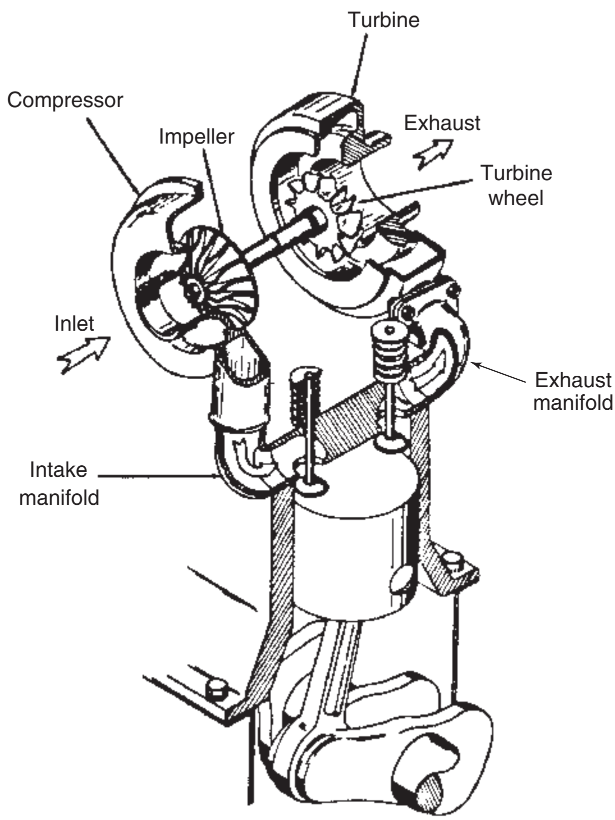

Figure 1.15 Turbocharger schematic. (Courtesy of Schwitzer.)

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-sized diesel engines are turbocharged to increase their efficiency.

Fuel Injectors and Carburetors

Revolutionary changes have taken place with computerized engine controls and fuel delivery systems in recent years and the progress continues. For example, the ignition and fuel injection is computer controlled in engines designed for vehicular applications. 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 they will not be knock limited. They will also be unthrottled, so they will have 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 its 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 systems. The liquid cooling system (see Figure 1.16) 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

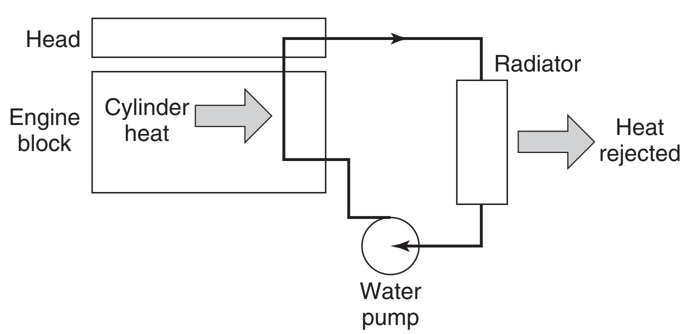

Figure 1.16 Liquid cooling system schematic.

Figure 1.17 Air cooling of model airplane engine. (Courtesy R. Schroeder.)

back to the pump. When the engine is cold, a thermostat prevents coolant from returning to the radiator, resulting in a more rapid warm-up 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 1.17). There are historical examples of combined water and air cooling. An early 1920s automobile, the Mors, 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 1.18 and in cutaway view in Figure 1.19. 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

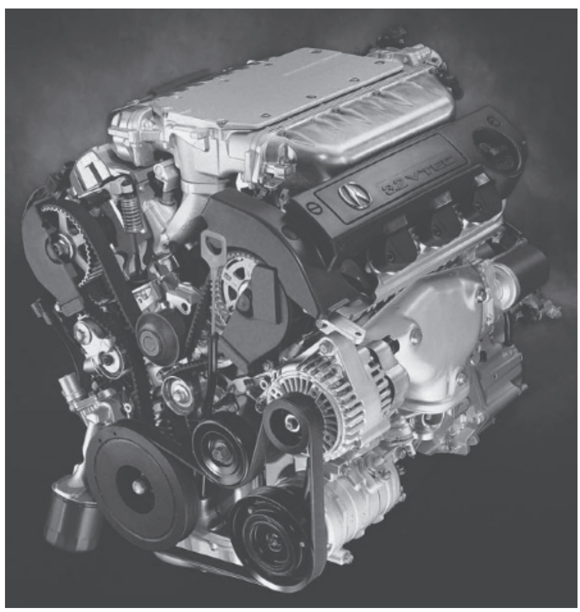

Figure 1.18 3.2 L V-6 automobile engine. (Courtesy of Honda Motor Co.)

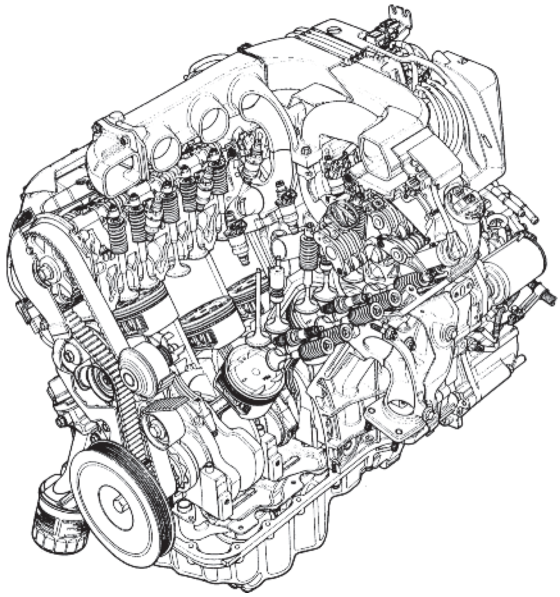

Figure 1.19 Cutaway view of 3.2 L V-6 automobile engine. (Courtesy of Honda Motor Co.)

piston bank with four valves per cylinder. The pistons are flat with notches for valve clearance. 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 1.20, 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

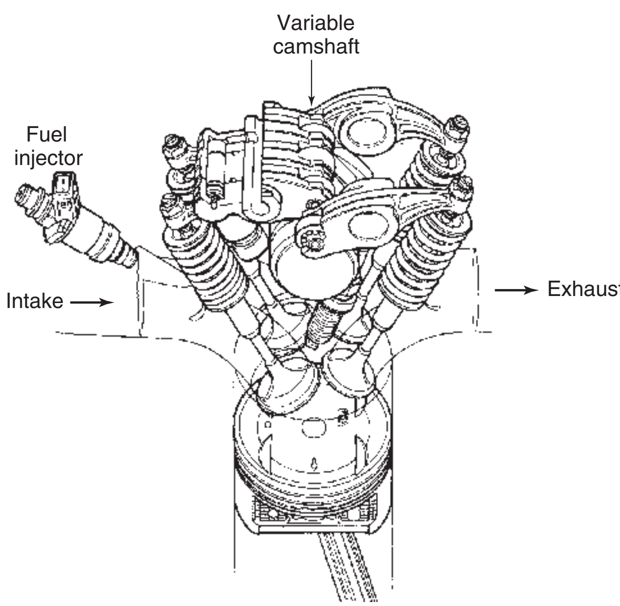

Figure 1.20 A variable valve timing mechanism. (Courtesy of Honda Motor Co.)

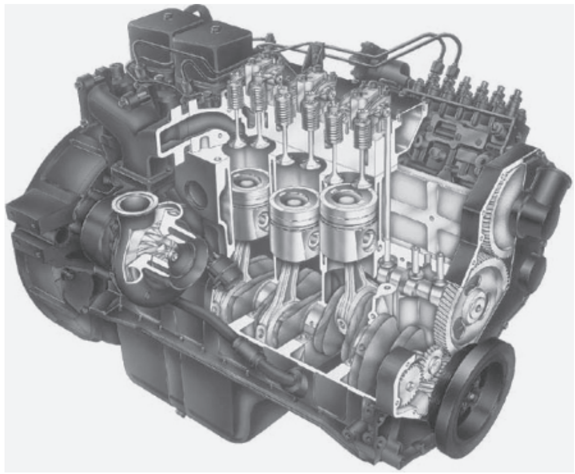

Figure 1.21 A 5.9 L L6 on-highway diesel engine. (Courtesy of PriceWebber.)

3500 rpm. In the low-lift short duration cam operation, the two intake valves have staggered timing that 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 1.21. 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 Mexican hat-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 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 1.22 and 1.23. Typical applications for stationary engines include cogeneration, powering gas compressors, and power generation. The engine shown in Figure 1.22 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 to 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

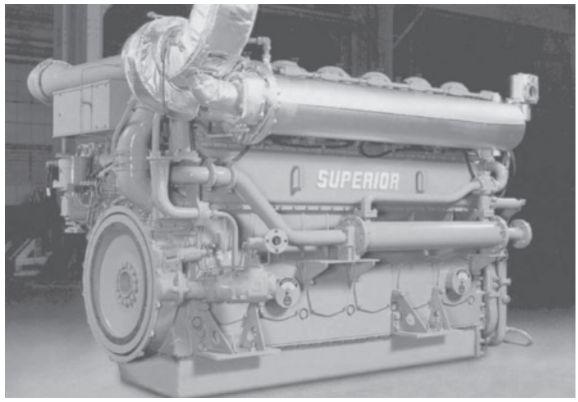

Figure 1.22 A 94 L L8 stationary natural gas engine. (Courtesy of Cooper Energy Services, Inc.)

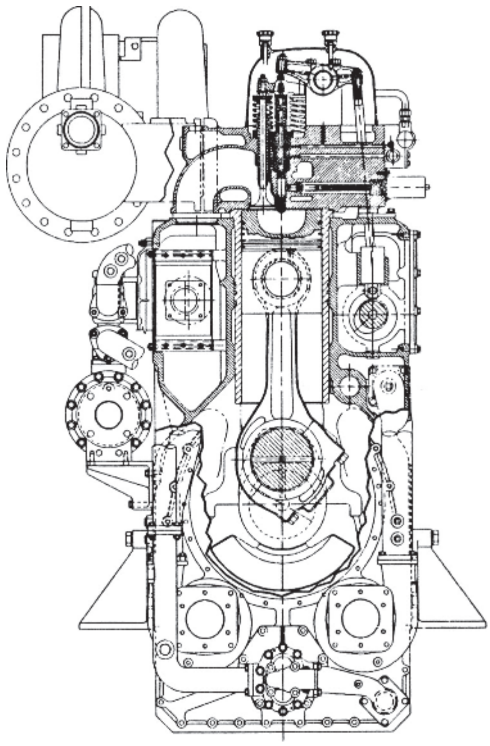

Figure 1.23 Cutaway view of 94 L L8 stationary natural gas engine. (Courtesy of Cooper Energy Services, Inc.)

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 POWER PLANTS

In this section, alternative power plants will be discussed in terms of a particular application where they dominate the field by having some advantage over the internal combustion engine.

First, consider electric motors which compete in the range of powers less than about 500 kW. They are used, for example, in forklifts operated within a factory or warehouse. Internal combustion engines are not applied in this case because they would build up high levels of pollutants such as carbon monoxide or nitric oxide. Electric motors are found in a variety of applications, such as where the noise and vibration of a piston engine or the handling of a fuel are unacceptable. Other examples are easy to think of in both indus trial and residential sectors. Electric motors will run in the absence of air, such as in outer space or under water; they are explosion proof; and they can operate at cryogenic temperatures. If one can generalize, one might state with respect to electric motors that internal combustion engines tend to be found in applications where mobility is a requirement or electricity is not available.

Proponents of electric vehicles point out that almost any fuel can be used to generate electricity, therefore we can reduce our dependence upon petroleum by switching to electric vehicles. There would be no exhaust emissions emitted throughout an urban environment. The emissions produced by the new electric generating stations could be localized geographically so as to minimize the effect. The main problem with electric vehicles is the batteries used for energy storage. The electric vehicles that have been built to date have a limited range of only 50–100 mi (80–160 km), on the order of one-fifth of what can be easily realized with a gasoline engine powered vehicle. 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. Batteries have about 1% of the energy per unit mass of a typical vehicular fuel, and a life span of about 5 years.

Hybrid electric vehicles (HEV), which incorporate a small internal combustion engine with an electric motor and storage batteries, have been the subject of recent research, and as of the year 2015, have reached the production stage, primarily due to their low fuel consumption and emission levels. A hybrid electric vehicle has more promise than an electric vehicle, since the HEV 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 fuels used in the HEV engines in current production include gasoline, diesel, and natural gas. Hybrid electric vehicles have a long history, as the first HEV, the Woods Dual Power automobile, was introduced in 1916.

As shown in Figure 1.24, the engine and electric motor are placed in either a series or parallel configuration. 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. The motors can be used as generators during braking to increase vehicle efficiency.

The fuel cell electric vehicle (FCEV) is currently in the development phase, and will be commercially available beginning in 2015. The chemical reaction in a fuel cell produces lower emissions relative to combustion in an internal combustion engine. Recent developments in proton exchange membrane (PEM) technology have been applied to vehicular fuel cells. Current PEM fuel cells are small enough to fit beneath a vehicle's floor next to storage batteries and deliver 50 kW to an electric motor. The PEM fuel cell requires a hydrogen fuel source to operate. Since there is presently no hydrogen fuel storage infrastructure, on-board reforming of methanol fuel to hydrogen and CO_2 is also required. 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 will be about 40%, about the same as high-efficiency internal combustion engines.

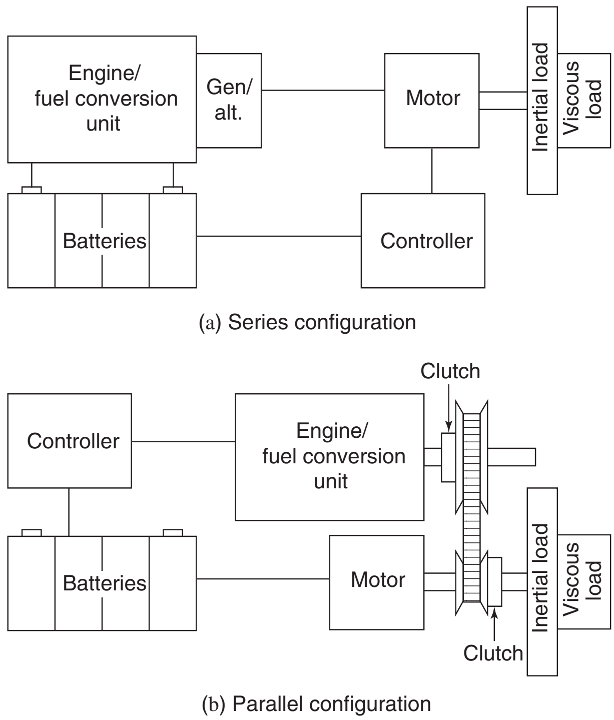

Figure 1.24 Hybrid electric vehicle powertrain configurations.

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 upon the adiabatic efficiency of its components, which in turn is highly dependent upon 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$2500 K) 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.

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 give the reader a deep 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), Heywood (1988), Cummins (1989), Arcoumanis (1998), Stone (1999), Lumley (1999), Pulkrabek (2003), Shi et al. (2011), Manning (2012), and Crolla et al. (2015).

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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	10,500	522

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1.9 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 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 and the clearance volume of a cylinder? (b) What is the engine bmep, brake power, and mean piston speed?

1.3 A four-cylinder 2.5 L 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)?

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 $\rho_{amb} = 1.17$ kg/m^3. If the engine speed is 2600 rpm, what is the air mass flow rate (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. (a) What is the fuel to air ratio? (b) What is the air mass flow rate (kg/s)?

1.6 A 3.8 L four-stroke four-cylinder fuel-injected automobile engine has a power output of 88 kW at 4000 rpm and volumetric efficiency of 0.85. The bsfc is 0.35 kg/kW h. If the fuel has a heat of combustion of 42 MJ/kg, what are the bmep, thermal efficiency, and air to fuel ratio? Assume atmospheric conditions of 298 K and 1 bar.

1.7 A 4.0 L six-cylinder 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, what is the stroke, the mean piston speed, cylinder clearance volume, and air mass flow rate into the engine? Assume standard inlet conditions.

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

1.9 Compare the approximate, Equation 1.29, and exact, Equation 1.26, 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?

1.10 Plot the dimensionless piston velocity for an engine with a stroke $s = 100$ mm and connecting rod length $l = 150$ mm.

1.11 Assuming that the mean effective pressure, mean piston speed, power per unit piston area, and mass per unit displacement volume are all size independent, how will the power per unit weight of an engine depend upon the number of cylinders if the total displacement is constant? To make the analysis easier, assume that the bore and stroke are equal.