Proof

Contents

CHAPTER 1

Introduction

1.1 Fundamental operating principles

The reciprocating internal combustion engine must be by far the most common form of engine or prime mover. As with most engines, the usual aim is to achieve a high work output with a high efficiency; the means to these ends are developed throughout this book. The term 'internal combustion engine' should also include open circuit gas turbine plant where fuel is burnt in a combustion chamber. However, it is normal practice to omit the prefix 'reciprocating'; none the less this is the key principle that applies to both engines of different types and those utilising different operating principles. The divisions between engine types and between operating principles can be explained more clearly if direct injection spark ignition (DISI) and Wankel-type engines are ignored initially; hence these are not discussed until section 1.4.

The two main types of internal combustion engine are: spark ignition (SI) engines, where the fuel is ignited by a spark; and compression ignition (CI) engines, where the rise in temperature and pressure during compression is sufficient to cause spontaneous ignition of the fuel. The spark ignition engine is also referred to as the petrol, gasoline or gas engine from its typical fuels, and the Otto engine, after the inventor. The compression ignition engine is also referred to as the Diesel or oil engine; the fuel is also named after the inventor.

During each crankshaft revolution there are two strokes of the piston, and both types of engine can be designed to operate in either four strokes or two strokes of the piston. The four-stroke operating cycle can be explained by reference to figure 1.1:

1. The induction stroke. The inlet valve is open, and the piston travels down the cylinder, drawing in a charge of air. In the case of a spark ignition engine the fuel is usually pre-mixed with the air.
2. The compression stroke. Both valves are closed, and the piston travels up the cylinder. As the piston approaches top dead centre (tdc), ignition occurs. In the case of compression ignition engines, the fuel is injected towards the end of the compression stroke.
3. The expansion, power or working stroke. Combustion propagates throughout the charge, raising the pressure and temperature, and forcing the piston down. At the end of the power stroke the exhaust valve opens, and the irreversible expansion of the exhaust gases is termed 'blow-down'.
4. The exhaust stroke. The exhaust valve remains open, and as the piston travels up the cylinder the remaining gases are expelled. At the end of the exhaust stroke, when the exhaust valve closes some exhaust gas residuals will be left; these will dilute the next charge.

The four-stroke cycle is sometimes summarised as 'suck, squeeze, bang and blow'. Since the cycle is completed only

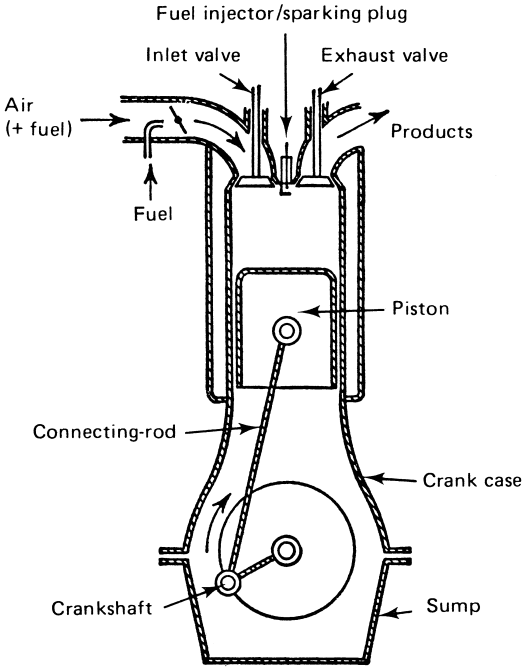


Figure 1.1
A four-stroke engine (reproduced with permission from Rogers and Mayhew, 1980a).

once every two revolutions the valve gear (and ignition or fuel injection equipment) have to be driven by mechanisms operating at half engine speed. Some of the power from the expansion stroke is stored in a flywheel, to provide the energy for the other three strokes.

The two-stroke cycle eliminates the separate induction and exhaust strokes; and the operation can be explained with reference to figure 1.2:

1 The compression stroke (figure 1.2a). The piston travels up the cylinder, so compressing the trapped charge. If the fuel is not pre-mixed, the fuel is injected towards the end of the compression stroke; ignition should again occur before top dead centre. Simultaneously, the underside of the piston is drawing in a charge through a spring-loaded non-return inlet valve.
2 The power stroke. The burning mixture raises the temperature and pressure in the cylinder, and forces the piston down. The downward motion of the piston also compresses the charge in the crankcase. As the piston approaches the end of its stroke the exhaust port is uncovered (figure 1.2b) and blowdown occurs. When the piston is at bottom dead centre (figure 1.2c) the transfer port is also uncovered, and the compressed charge in the crankcase expands into the cylinder. Some of the remaining exhaust gases are displaced by the fresh charge; because of the flow mechanism this is called 'loop scavenging'. As the piston travels up the cylinder, first the transfer port is closed by the piston, and then the exhaust port is closed.

For a given size engine operating at a particular speed, the two-stroke engine will be more powerful than a four-stroke engine since the two-stroke engine has twice as many power strokes per unit time. Unfortunately the efficiency of a two-stroke engine is likely to be lower than that of a four-stroke engine. The problem with two-stroke engines is ensuring that the induction and exhaust processes occur efficiently, without suffering charge dilution by the exhaust gas residuals. The spark ignition engine is particularly troublesome, since at part throttle operation the crankcase pressure can be less than atmospheric pressure. This leads to poor scavenging of the exhaust gases, and a rich air/fuel mixture becomes necessary for all conditions, with an ensuing low efficiency (see chapter 4, section 4.1).

These problems can be overcome in two-stroke direct injection engines by supercharging, so that the air pressure at inlet to the crankcase is greater than the exhaust back-pressure. This ensures that when the transfer port is opened, efficient scavenging occurs; if some air passes straight through the engine, it does not lower the efficiency since no fuel has so far been injected.

Originally engines were lubricated by total loss systems with oil baths around the main bearings or splash lubrication from oil in the sump. As engine outputs increased a circulating high-pressure oil system became necessary; this also assisted the heat transfer. In two-stroke spark ignition engines a simple system can be used in which oil is pre-mixed with the fuel; this removes the need for an oil pump and filter system.

An example of an automotive four-stroke compression ignition engine is shown in figure 1.3, and a two-stroke spark ignition motor cycle engine is shown in figure 1.4.

The size range of internal combustion engines is very large, especially for compression ignition engines. Two-stroke compression ignition engines vary from engines for models with swept volumes of about 1 cm^3 and a fraction of a kilowatt output, to large marine engines with a cylinder bore of about 1 m, up to 12 cylinders in-line, and outputs of up to 50 MW.

An example of a large two-stroke engine is the Sulzer RTA engine (see figure 1.5) described by Wolf (1982). The efficiency increases with size because the effects of clearances and cooling losses diminish. As size increases the operating speed reduces and this leads to more efficient combustion;

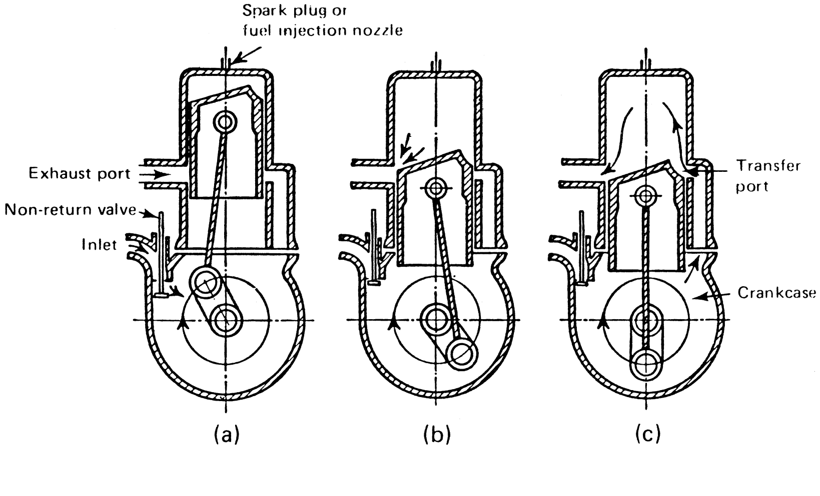


Figure 1.2
A two-stroke engine (reproduced with permission from Rogers and Mayhew, 1980a).

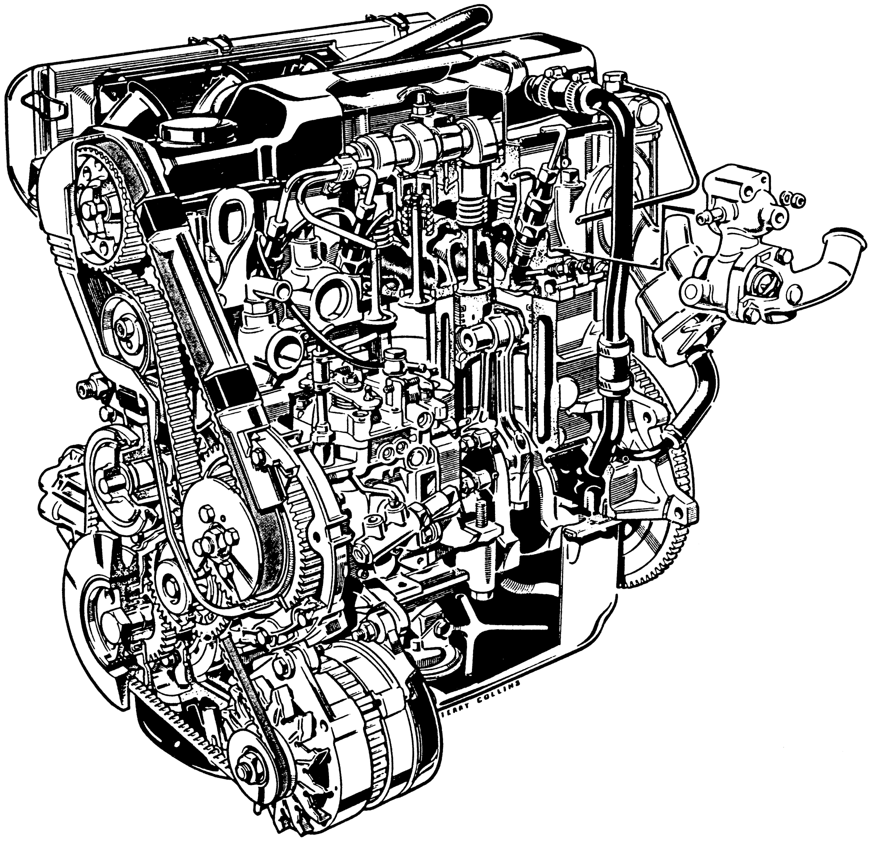


Figure 1.3
Ford 1.6-litre diesel engine (courtesy of Ford).

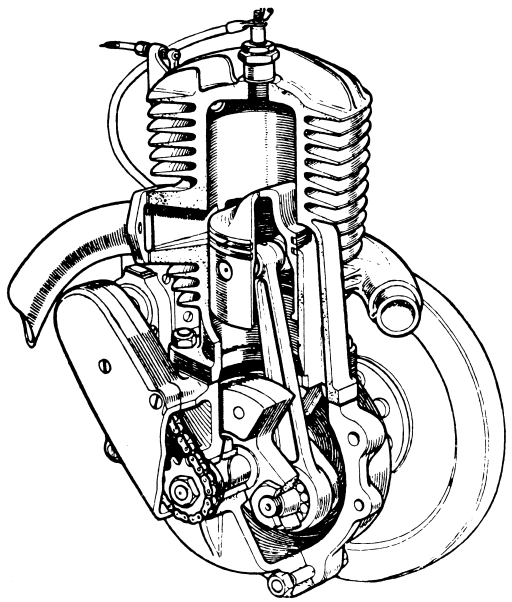
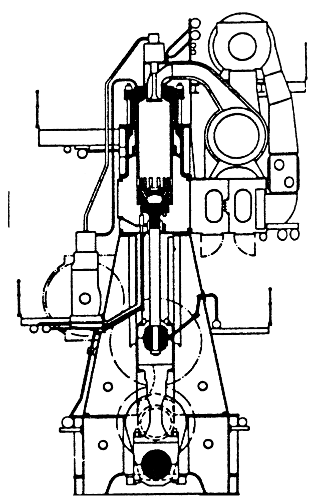


Figure 1.4
Two-stroke spark ignition engine.



Power/cylinder	2960 kW
Swept volume/cylinder	1.32 m^3
Speed	87 rpm
Peak cylinder pressure	76 bar
Mean effective pressure	8.65 bar

Figure 1.5
Sulzer RTA two-stroke compression ignition engine (courtesy of Wärtsilä).

also the specific power demand from the auxiliaries reduces. The efficiency of such an engine can exceed 50 per cent, and it is also capable of burning low-quality residual fuels. A further advantage of a low-speed marine diesel engine is that it can be coupled directly to the propeller shaft. To run at low speeds an engine needs a long stroke, yet a stroke/bore ratio greater than 2 leads to poor loop scavenging.

The Sulzer RTA engine has a stroke/bore ratio of about 3 and uses uniflow scavenging. This requires the additional complication of an exhaust valve. Figure 1.6 shows arrangements for loop, cross and uniflow scavenging. The four-stroke compression ignition engines have a smaller size range, from about 400 cm^3 per cylinder to 60 litres per cylinder with an output of 600 kW per cylinder at about 600 rpm with an efficiency of over 45 per cent.

The size range of two-stroke spark ignition engines is small, with the total swept volumes rarely being greater than 1000 cm^3. The common applications are in off-highway applications such as chainsaws, where the high output, simplicity and low weight are more important than their poor fuel economy and high emissions.

Since the technology of two-stroke engines is rather different from that of four-stroke engines, chapter 8 is devoted to two-stroke engines. The interest in two-stroke engines is because of the potential for higher specific power outputs and a more frequent firing interval. More detail can be found in the books published by Blair (1990, 1996).

Automotive four-stroke spark ignition engines usually have cylinder volumes in the range 50–500 cm^3, with the total swept volume rarely being greater than 5000 cm^3. Engine outputs are typically 50 kW/litre, a value that can be increased seven-fold by tuning and turbocharging.

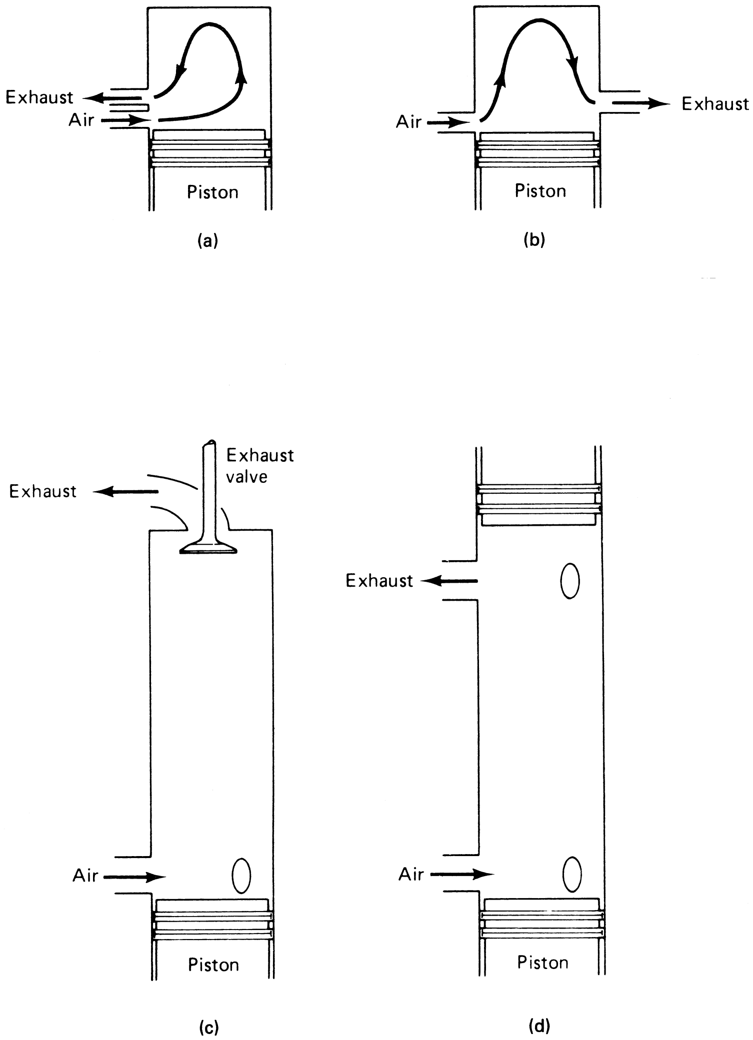


Figure 1.6
Two-stroke scavenging systems: (a) loop scavenging; (b) cross scavenging;
(c) uniflow scavenging with exhaust valve;
(d) uniflow scavenging with opposed pistons.

The largest spark ignition engines are gas engines; these are usually converted from large medium-speed compression ignition engines. High-output spark ignition engines have been developed for racing and in particular for aero-engines. A famous example of an aero-engine is the Rolls-Royce Merlin V12 engine. This engine had a swept volume of 27 litres and a maximum output of 1.48 MW at 3000 rpm; the specific power output was 1.89 kW/kg.

1.2 Early internal combustion engine development

As early as 1680 Huygens proposed to use gunpowder for providing motive power. In 1688 Papin described the engine to the Royal Society of London, and conducted further experiments. Surprising as it may seem, these engines did not use the expansive force of the explosion directly to drive a piston down a cylinder. Instead, the scheme was to explode a small quantity of gunpowder in a cylinder, and to use this effect to expel the air from the cylinder. On cooling, a partial vacuum would form, and this could be used to draw a piston down a cylinder – the so-called ‘atmospheric’ principle.

Papin soon found that it was much more satisfactory to admit steam and condense it in a cylinder. This concept was used by Newcomen who constructed his first atmospheric steam engine in 1712. The subsequent development of atmospheric steam engines, and the later high-pressure steam engines (in which the steam was also used expansively), overshadowed the development of internal combustion engines for almost two centuries. When internal combustion engines were ultimately produced, the technology was based heavily on that of steam engines.

Throughout the late 18th and early 19th century there were numerous proposals and patents for internal combustion engines; only engines that had some commercial success will be mentioned here.

The first engine to come into general use was built by Lenoir in 1860; an example of the type is shown in figure 1.7. The engine resembled a single-cylinder, double-acting horizontal steam engine, with two power strokes per revolution. Induction of the air/gas charge and exhaust of the burnt mixture were controlled by slide valves; the ignition was obtained by an electric spark. Combustion occurred on both sides of the piston, but considering just one combustion chamber the sequence was as follows:

1 In the first part of the stroke, gas and air were drawn in. At about half stroke, the slide valves closed and the mixture was ignited; the explosion then drove the piston to the bottom of the stroke.
2 In the second stroke, the exhaust gases were expelled while combustion occurred on the other side of the piston.

Question 2.12, in the next chapter, analyses the Lenoir engine by modelling it with an air standard cycle. This provides an upper bound on its efficiency, which can be seen to be about one-third of that of a typical spark ignition engine with compression.

The next significant step was the Otto and Langen atmospheric or free-piston engine of 1866; the fuel consumption was about half that of the Lenoir engine. The main features of the engine were a long vertical cylinder, a heavy piston and a racked piston rod (figure 1.8). The racked piston rod was engaged with a pinion connected to the output shaft by a ratchet. The ratchet was arranged to free-wheel on the upward stroke, but to engage on the downward stroke. Starting with the piston at the bottom of the stroke the operating sequence was as follows:

1 During the first tenth or so of the stroke, a charge of gas and air was drawn into the cylinder. The charge was ignited by a flame transferred through a slide valve, and the piston was forced to the top of its stroke without delivering any work, the work being stored as potential energy in the heavy piston.
2 As the cylinder contents cooled, the partial vacuum so formed, and the weight of the piston, transferred the work to the output shaft on the downward stroke. Exhaust occurred at the end of this stroke.

The piston had to weigh about 70 kg per kW of output, and by its nature the engine size was limited to outputs of a few kilowatts; none the less some 10 000 engines were produced within five years.

At the same time commercial exploitation of oil wells in the USA was occurring, as a result of the pioneer drilling by Drake in 1859. This led to the availability of liquid fuels that were

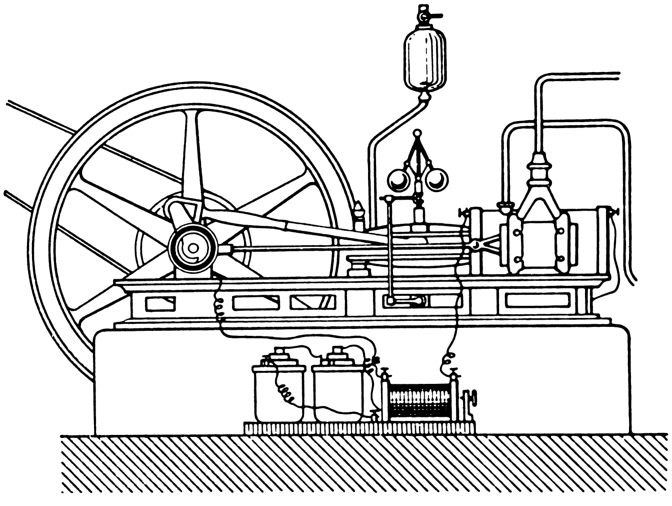


Figure 1.7
Lenoir gas engine of 1860 (reproduced with permission from Singer, History of Technology, Oxford University Press, 1978).

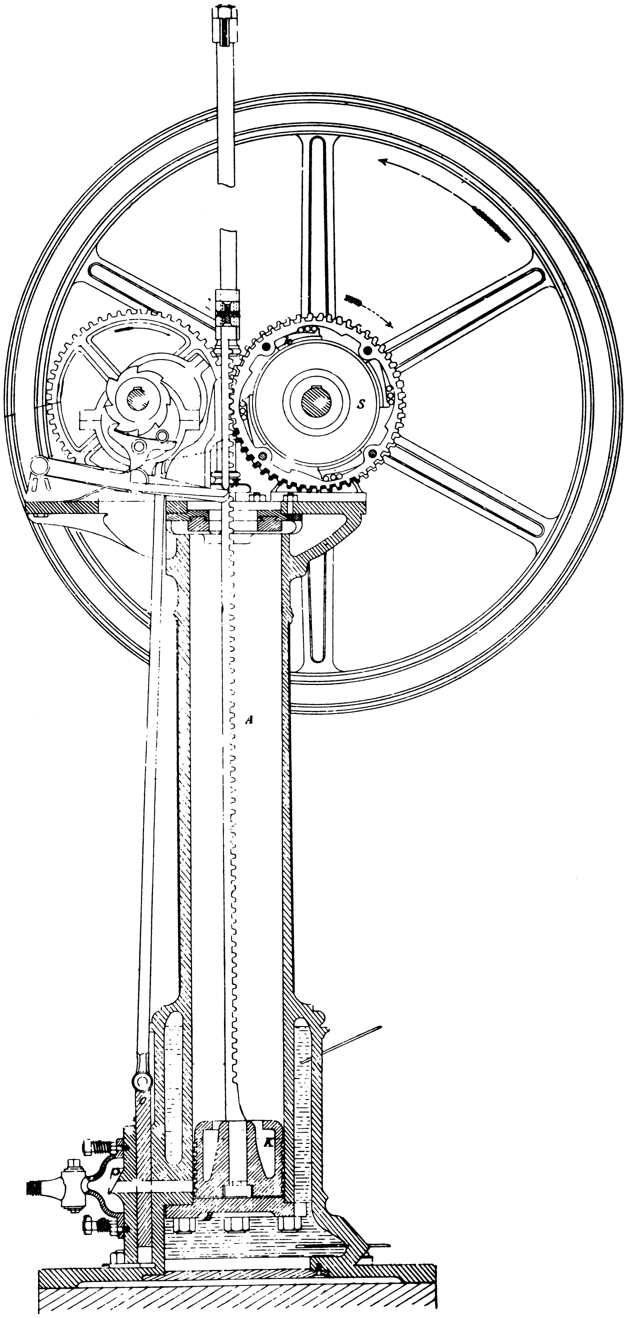


Figure 1.8
Otto and Langen free-piston engine.

much more convenient to use than gaseous fuels, since these often needed a dedicated gas-producing plant. Liquid fuels without doubt accelerated the development of internal combustion engines, and certainly increased the number of different types available, with oil products providing both the lubricant and the fuel. For the remainder of the 19th century any engine using a gaseous fuel was called a gas engine, and any engine using a liquid was called an oil engine; no reference was necessarily made to the mode of ignition or the different operational principles.

In 1876 the Otto silent engine using the four-stroke cycle was patented and produced. As well as being much quieter than the free-piston engine, the silent engine was about three times as efficient. Otto attributed the improved efficiency to a conjectured stratification of the charge. This erroneous idea was criticised by Sir Dugald Clerk, who appreciated that the improved efficiency was a result of the charge being compressed before ignition. Clerk subsequently provided the first analysis of the Otto cycle (see chapter 2, section 2.2.1).

The concept of compression before ignition can be traced back to Schmidt in 1861, but perhaps more remarkable is the work of Beau de Rochas. As well as advocating the four-stroke cycle, Beau de Rochas included the following points in 1862:

1 There should be a high volume-to-surface ratio.
2 The maximum expansion of the gases should be achieved.
3 The highest possible mixture pressure should occur before ignition.

Beau de Rochas also pointed out that ignition could be achieved by sufficient compression of the charge.

Immediately following the Otto silent engine, two-stroke engines were developed. Patents by Robson in 1877 and 1879 describe the two-stroke cycle with under-piston scavenging, while patents of 1878 and 1881 by Clerk describe the two-stroke cycle with a separate pumping or scavenge cylinder.

The quest for self-propelled vehicles needed engines with better power-to-weight ratios. Daimler was the first person to realise that a light high-speed engine was needed, which would produce greater power by virtue of its higher speed of rotation, 500–1000 rpm. Daimler's patents date from 1884, but his twin cylinder 'V' engine of 1889 was the first to be produced in quantity. By the end of the 19th century the petrol engine was in a form that would be currently recognisable, but there was still much scope for development and refinement.

The modern compression ignition engine developed from the work of two people, Akroyd Stuart and Rudolf Diesel. Akroyd Stuart's engine, patented in 1890 and first produced in 1892, was a four-stroke compression ignition engine with a compression ratio of about 3 – too low to provide spontaneous ignition of the fuel. Instead, this engine had a large uncooled pre-chamber or vaporiser connected to the main cylinder by a short narrow passage. Initially the vaporiser was heated externally, and the fuel then ignited after it had been sprayed into the vaporiser at the end of the compression stroke. The turbulence generated by the throat to the vaporiser ensured rapid combustion. Once the engine had been started the external heat source could be removed. The fuel was typically a light petroleum distillate such as kerosene or fuel oil; the efficiency of about 15 per cent was comparable with that of the Otto silent engine. The key innovations with the Akroyd Stuart engine were the induction, being solely of air, and the injection of fuel into the combustion chamber.

Diesel's concept of compressing air to such an extent that the fuel would spontaneously ignite after injection was published in 1890, patented in 1892 and achieved in 1893; an early example is shown in figure 1.9. Some of Diesel's aims were unattainable, such as a compression pressure of 240 bar, the use of pulverised coal and an uncooled cylinder. None the less, the prototype ran with an efficiency of 26 per cent, about twice the efficiency of any contemporary power plant and a figure that steam power plant achieved only in the 1930s.

Diesel injected the fuel by means of a high-pressure (70 bar) air blast, since a liquid pump for 'solid' or airless injection was not devised until 1910 by McKechnie. Air-blast injection necessitated a costly high-pressure air pump and storage vessel; this restricted the use of diesel engines to large stationary and marine applications. Smaller high-speed compression ignition engines were not used for automotive applications until the 1920s. The development depended on experience gained from automotive spark ignition engines, the development of airless quantity-controlled fuel injection pumps by Bosch (chapter 6, section 6.5.2), and the development of suitable combustion systems by people such as Ricardo.

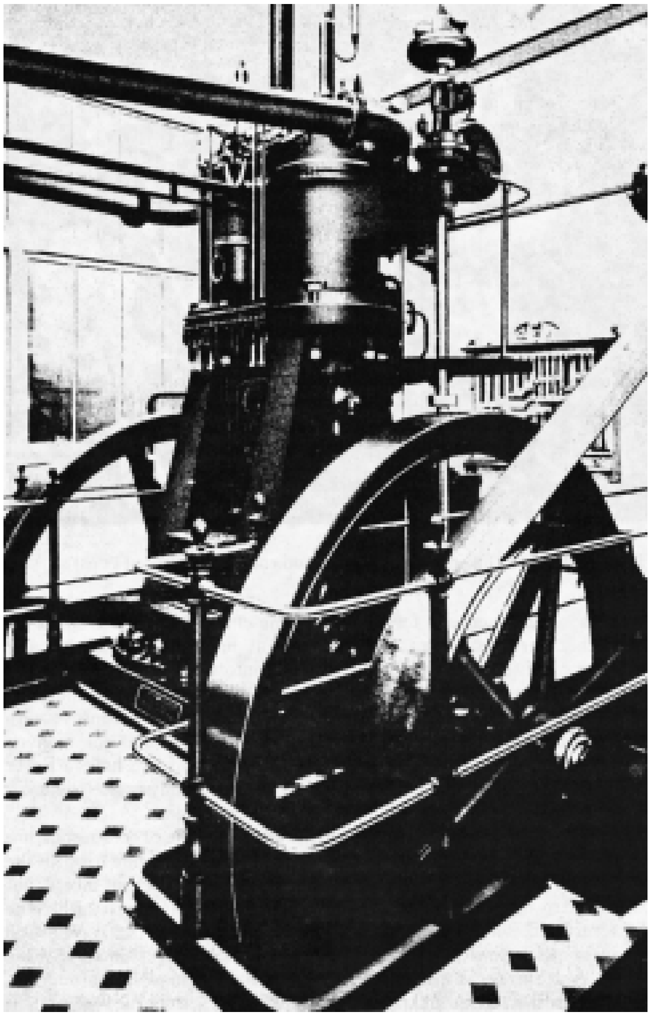


Figure 1.9
Early (1898) diesel engine (output: 45 kW at 180 rpm) (reproduced with permission from Singer, History of Technology, Oxford University Press, 1978).

1.3 Characteristics of internal combustion engines

The purpose of this section is to discuss one particular engine – the Ford 'Dover' direct injection in-line six-cylinder truck engine (figure 1.10). This turbocharged engine has a swept volume of 6 litres, weighs 488 kg and produces a power output of 114 kW at 2400 rpm.

In chapter 2 the criteria for judging engine efficiency are derived, along with other performance parameters, such as mechanical efficiency and volumetric efficiency, which provide insight into why a particular engine may or may not be efficient. Thermodynamic cycle analysis also indicates that, regardless of engine type, the cycle efficiency should improve with higher compression ratios. Furthermore, it is shown that for a given compression ratio, cycles that compress a fuel/air mixture have a lower efficiency than cycles that compress pure air. Cycle analysis also indicates the work that can be extracted by an exhaust gas turbine.

The type of fuel required for this engine and the mode of combustion are discussed in chapter 3. Since the fuel is injected into the engine towards the end of the compression stroke, the combustion is not pre-mixed but controlled by diffusion processes – the diffusion of the fuel into the air, the diffusion of the air into the fuel, and the diffusion of the combustion products away from the reaction zone. Turbulence is essential if these processes are to occur in the small time available. The main fuel requirement is that the fuel should readily self-ignite; this is the exact opposite to the requirements for the spark ignition engine. Fuel chemistry and combustion are discussed in chapter 3, along with additives, principally those that either inhibit or promote self-ignition. One of the factors limiting the output of this, and any other, diesel engine is the amount of fuel that can be injected before unburnt fuel leaves the engine as smoke (formed by agglomerated carbon particles). These and other engine emissions are discussed in sections 3.8 and 6.6.

A key factor in designing a successful compression ignition engine is the design of the combustion chamber, and the correct matching of the fuel injection to the in-cylinder air motion. These factors are discussed in chapter 6, the counterpart to chapter 4 which discusses spark ignition engines. The design and manufacture of fuel injection

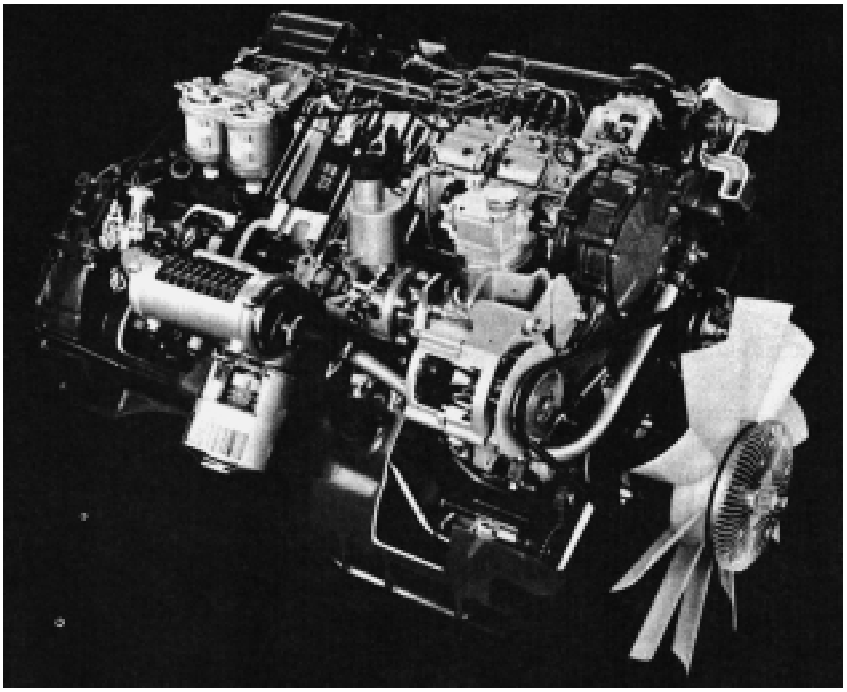


Figure 1.10
Ford Dover compression ignition engine (courtesy of Ford).

equipment are undertaken by specialist manufacturers; however, the final matching of the fuel injection equipment to the engine still has to be done experimentally. This turbocharged engine is likely to have a lower compression ratio than its naturally aspirated counterpart, in order to limit the peak pressures during combustion. One effect of this is that starting is made more difficult; the compression ratio is often chosen to be the minimum that will give reliable starting. None the less, cold weather or poor fuel quality can lead to starting difficulties, and methods to improve starting are discussed in section 6.4.

The induction and exhaust processes are controlled by poppet valves in the cylinder head. The timing of events derives from the camshaft, but is modified by the clearances and the elastic properties of the valve gear. The valve timing for the turbocharged engine will keep the valves open for longer periods than in the naturally aspirated version, since appropriate turbocharger matching can cause the inlet pressure to be greater than the exhaust. These aspects, and the nature of the flow in the inlet and exhaust passages, are discussed in chapter 7. In the turbocharged engine the inlet and exhaust manifold volumes will have been minimised to help reduce the turbocharger response time. In addition, care will have to be taken to ensure that the pressure pulses from each cylinder do not interfere with the exhaust system. The design of naturally aspirated induction and exhaust systems can be very involved if the engine performance is to be optimised. Pressure pulses can be reflected as rarefaction waves, and these can be used to improve the induction and exhaust processes. Since these are resonance effects, the engine speed at which maximum benefit occurs depends on the system design and the valve timing, see sections 7.4, 7.5 and 7.6.

Chapter 9 discusses the in-cylinder motion of the air in some detail. It provides definitions of turbulence and swirl, and explains how both can be measured. Chapter 6 should be read before chapter 9, as this is where the significance of swirl and turbulence is explained.

The turbocharger is another component that is designed and manufactured by specialists. Matching the turbocharger to the engine is difficult, since the flow characteristics of each machine are fundamentally different. The engine is a slow-running (but large) positive displacement machine, while the turbocharger is a high-speed (but small) non-positive displacement machine, which relies on dynamic flow effects. Turbochargers inevitably introduce a lag when speed or load is increased, since the flow rate can only increase as the rotor speed increases. These aspects, and the design of the compressor and turbine, are covered in chapter 10, along with applications to spark ignition engines. Turbochargers increase the efficiency of compression ignition engines since the power output of the engine is increased more than the mechanical losses.

Chapter 11 describes how computer modelling can be applied to internal combustion engines. Since the interaction of dynamic flow devices and positive displacement devices is complex, the chapter on computer modelling ends with an example on the computer modelling of a turbocharged diesel engine.

Some of the mechanical design considerations are dealt with in chapter 12. A six-cylinder in-line engine will have even firing intervals that produce a smooth torque output. In addition, there will be complete balance of all primary and secondary forces and moments that are generated by the reciprocating elements. The increased cycle temperatures in the turbocharged engine make the design and materials selection for the exhaust valve and the piston assembly particularly important. The increased pressures also raise the bearing loads and the role of the lubricant as a coolant will be more important. Computers are increasingly important in design work, as component weights are being reduced and engine outputs are being raised.

Chapter 13 is devoted to engine cooling systems. These are often taken for granted, but chapter 13 explains the principles, with particular emphasis on liquid cooled engines. There is also a description of how cooling systems can be designed to accelerate engine warm-up, and to provide higher efficiencies at steady-state operating points.

The use of computers is increasing in all aspects of engine work: modelling the engine to estimate performance, matching of the fuel injection equipment, selection and performance prediction of the turbocharger, estimation of vehicle performance (speed, fuel consumption etc.) for different vehicles, transmission and usage combinations.

None the less, when an engine such as the Dover diesel is being developed there is still a need to test engines. The designers will want to know about its performance at all loads and speeds. Some measurements, such as the fuel efficiency, are comparatively simple to make, and these techniques are introduced at the start of chapter 14. Chapter 14 also describes how exhaust emissions are measured, and the procedures needed for in-cylinder combustion analysis. The chapter ends with a description of computer-controlled test facilities.

1.4 Additional types of internal combustion engine

Two types of engine that fall outside the simple classification of reciprocating spark ignition or compression ignition engine are the Wankel engine and the direct injection spark ignition engine.

1.4.1 The Wankel engine

The Wankel engine is a rotary combustion engine, developed from the work of Felix Wankel. The mode of operation is best explained with reference to figure 1.11. The triangular rotor has a centrally placed internal gear that meshes with a sun gear that is part of the engine casing. An eccentric that is an integral part of the output shaft constrains the rotor to follow a planetary motion about the output shaft. The gear ratios are such that the output shaft rotates at three times the speed of the rotor, and the tips of the rotor trace out the two-lobe epitrochoidal shape of the casing. The compression ratio is dictated geometrically by the eccentricity of the rotor and the shape of its curved surfaces. The convex surfaces shown in the diagram minimise the sealed volumes, to give the highest compression ratio and optimum gas exchange. A recess in the combustion chamber provides a better-shaped combustion chamber.

The sequence of events that produces the four-stroke cycle is as follows. In figure 1.11a with the rotor turning in a clockwise direction a charge is drawn into space 1, the preceding charge is at maximum compression in space 2, and the combustion products are being expelled from space 3. When the rotor turns to the position shown in figure 1.11b, space 1 occupied by the charge is at a maximum, and further rotation will cause compression of the charge. The gases in space 2 have been ignited and their expansion provides the power stroke. Space 3 has been reduced in volume, and the exhaust products have been expelled. As in the two-stroke loop or cross scavenge engine there are no valves, and here the gas flow through the inlet and exhaust is controlled by the position of the rotor apex.

For effective operation the Wankel engine requires efficient seals between the sides of the rotor and its casing, and the more demanding requirement of seals at the rotor tips. Additional problems to be solved were cooling of the rotor, the casing around the spark plug and the exhaust passages. Unlike a reciprocating engine, only a small part of the Wankel engine is cooled by the incoming charge. Furthermore, the spark plug has to operate reliably under much hotter conditions. Not until the early 1970s were the sealing problems sufficiently solved for the engine to enter production. The advantages of the Wankel are its compactness, the apparent simplicity, the ease of balance and the potential for high outputs by running at high speeds.

The major disadvantages of the Wankel engine are its low efficiency (caused by limited compression ratios) and the high exhaust emissions resulting from the poor combustion chamber shape. By the mid 1970s concern over firstly engine emissions, and secondly fuel economy led to the demise of the Wankel engine. Experiments with other types of rotary combustion engine have not led to commercial development.

1.4.2 Direct injection spark ignition engines

Direct injection spark ignition engines, as the name implies, have the fuel injected directly into the cylinder. They can operate in two different regimes:

1 At full load, injection occurs during induction, so that

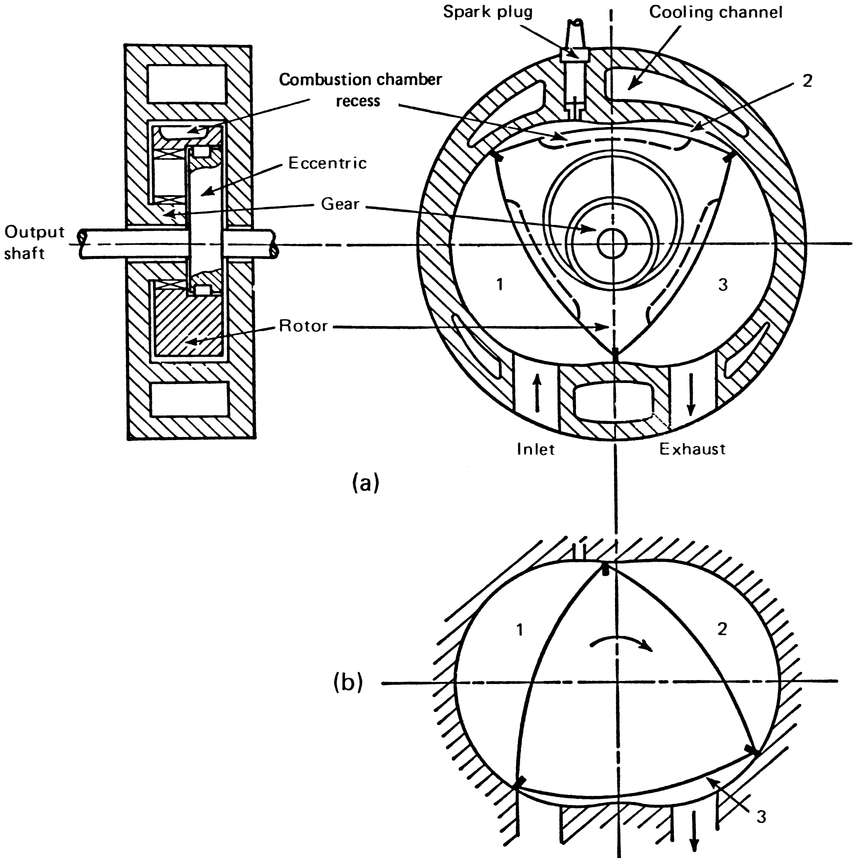


Figure 1.11
The Wankel engine (reproduced with permission from Rogers and Mayhew, 1980a).

there is time for a homogeneous air/fuel mixture to be formed. This early injection also facilitates evaporative cooling of the air and combustion chamber, so that a high mass of air is trapped in the cylinder. This enables the maximum power output to be comparable with a conventional spark ignition engine.

2 At part load, injection occurs during the compression stroke, and it has to be timed so that there is a stratified charge, with a flammable mixture formed at the spark plug, at the time that ignition is required. This gives the potential of a higher efficiency, since the throttling losses can be reduced or eliminated. There is thus the potential for the fuel economy to approach that of diesel engines.

The principle behind stratified charge engines is to have a readily ignitable mixture in the vicinity of the spark plug, and a weaker (possibly non-ignitable) mixture in the remainder of the combustion chamber. The purpose of this arrangement is to control the power output of the engine by varying only the fuel supply without throttling the air, thereby eliminating the throttling pressure-drop losses. The stratification of the charge was initially obtained by division of the combustion chamber to produce a pre-chamber that contains the spark plug. Typically fuel would also be injected into the pre-chamber, so that charge stratification is controlled by the timing and rate of fuel injection. Thus fuel supply is controlled in the same manner as compression ignition engines, yet the ignition timing of the spark controls the start of combustion.

Another means of preparing a stratified charge was to provide an extra valve to the pre-chamber, which controlled a separate air/fuel mixture. This was the method used in the Honda CVCC engine (figure 1.12), the first stratified charge engine in regular production.

However, of great significance is the potential for direct injection spark ignition (DISI) engines to achieve the specific output of gasoline engines, yet with fuel economy that is comparable to diesel engines. Mitsubishi launched such a DISI engine, and this has been described by Kume *et al.* (1996) and Ando (1997). The Mitsubishi engine used a combustion chamber formed by a carefully shaped piston, which controls the interaction between the air motion and fuel injection. More recently, much higher fuel injection pressures have been used (150 bar) so that a much more finely atomised fuel spray is formed, and the intention is to form a flammable

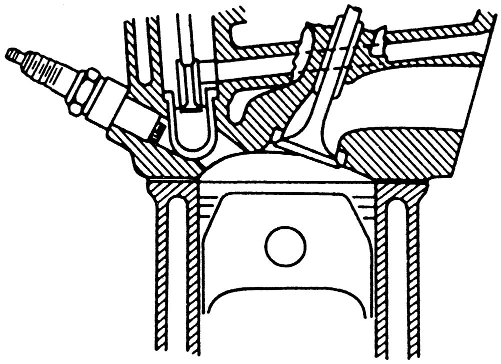


Figure 1.12
Honda CVCC engine, the first in regular production (from Campbell, 1978).

mixture in the region of the spark plug without fuel wetting the piston. When liquid fuel contacts the piston it leads to a 'pool fire' – a diffusion flame that is a source of particulate matter emissions.

The operation of direct injection spark ignition engine is discussed further in chapter 5. The design of stratified charge engines is complex, and the operating envelope for stratified charge operation is also restricted by emissions legislation. With weak mixtures, the conventional three-way catalyst will not reduce NO_x emissions, therefore a stratified charge engine has to operate with a very weak mixture, so that the engine-out exhaust emissions will satisfy the NO_x emissions requirements.

A proper discussion of stratified charge engines requires a knowledge of chapters 2–6.

1.5 Prospects for internal combustion engines

The future of internal combustion engines will be influenced by two factors: the future cost and availability of suitable fuels, and the development of alternative power plants.

Liquid fuels are by far the most convenient energy source for internal combustion engines, and the majority (over 99 per cent) of such fuels come from crude oil. The oil price is largely governed by political and taxation policy, and there is no reason to suppose that these areas of control will change.

In 2008 the world consumption of oil was about 84 million barrels per day. This figure has increased by 15 per cent in the last decade, with over half of the increase attributable to Asia. Current known oil reserves would then imply a supply of crude oil for another 40 years or so. However, exploration for oil continues and new reserves are being found; it must also be remembered that oil companies cannot justify expensive exploration work to demonstrate reserves for, say, the next 100 years. The ratio of oil reserves to the rate of production is shown in figure 1.13. This suggests a continuing equilibrium between supply and demand.

Internal combustion engines can also be fuelled from renewable energy sources. Spark ignition engines run satisfactorily on alcohol-based fuels, and compression ignition engines can operate on vegetable oils. Countries such as Brazil, with no oil reserves but plentiful sources of vegetation, are already operating an alcohol-fuelled policy.

The other major source of hydrocarbons is coal, and even conservative estimates show a 200-year supply from known reserves. One approach is to introduce a suspension of coal particles into the heavy fuel oil used by large compression ignition engines. A more generally applicable alternative is the preparation of fuels by 'liquefaction' or 'gasification' of coal. A

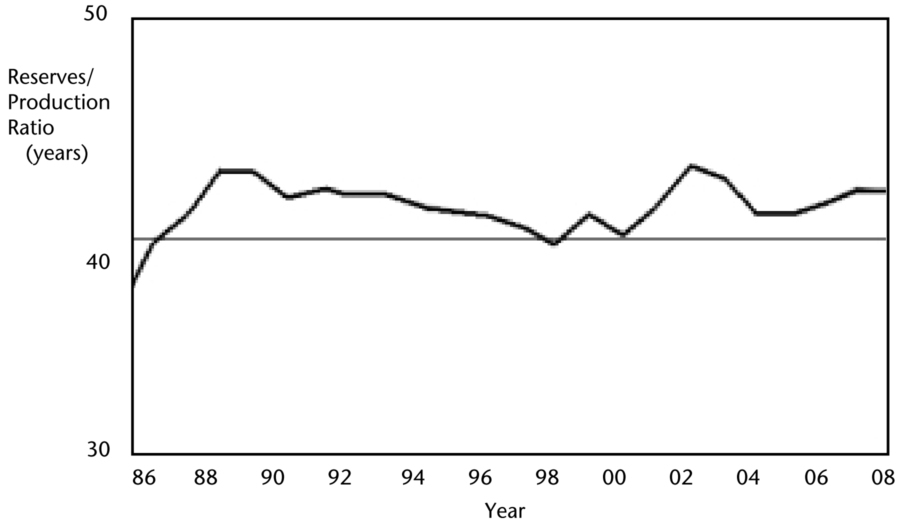


Figure 1.13
Worldwide reserves/production ratio (data derived from the BP Statistical Review of World Energy, 2009).

comprehensive review of alternative processes is given by Davies (1983), along with the yields of different fuels and their characteristics.

The preceding remarks indicate that the future fuel supply is assured for internal combustion engines, but that other types of power plant may supersede them; the following is a discussion of some of the possibilities.

Steam engines have been used in the past, and would have the advantages of external combustion of any fuel, with readily controlled emissions. However, if it is possible to overcome the low efficiency and other disadvantages it is probable that this would have already been achieved and they would already have been adopted; their future use is thus unlikely.

Stirling engines have been developed, with some units for automotive applications built by United Stirling of Sweden. The fuel economy at full and part load is comparable to compression ignition engines, but the cost of building the complex engine is about 50 per cent greater. As the Stirling engine has external combustion it too has the capability of using a wide range of fuels with readily controlled emissions.

Gas turbines present another alternative: conventionally they use internal combustion, but external combustion is also possible. For efficient operation a gas turbine would require a high efficiency compressor and turbine, high pressure ratio, high combustion temperatures and an effective heat exchanger. These problems have been solved for large aero and industrial gas turbines, but scaling down to even truck engine size changes the design philosophy. The smaller size would dictate the less efficient radial flow compressor, with perhaps a pressure ratio of 5:1 and a regenerative heat exchanger to preserve the efficiency. All these problems could be solved, along with reductions in manufacturing cost, by the development of ceramic materials. However, part load efficiency is likely to remain poor, although this is of less significance in truck applications. The application to private passenger vehicles is even more remote because of the importance of part load efficiency, and the reductions in efficiency that would follow from the smaller size. Marine application has been limited because of the low efficiency. In addition, the efficiency can deteriorate rapidly if the turbine blades corrode as a result of the combustion of salt-laden air.

Electric vehicles present an interesting possibility which is currently restricted by the lack of a suitable battery. Performance figures have been improved to give a typical speed of about 100 km per hour, and a range of 120 km. Battery technology is being improved, and lithium-ion and lithium-polymer battery systems are being developed, with two to three times the energy density of lead/acid batteries. However, batteries have less than 1 per cent of the energy density of hydrocarbon fuels. Electric vehicles could meet the majority of personal transport needs, and are viable for local delivery vehicles. However, electric vehicles are expensive, and their use has to be either subsidised or encouraged by fiscal or other means.

This leads to the possibility of a hybrid vehicle that has both an internal combustion engine and an electric motor. This is obviously an expensive solution but one that is versatile and efficient by using the motor and/or engine.

The simplest hybrid vehicles are those with parallel or series configurations. In the parallel configuration, the vehicle can be propelled by the electric motor and engine, either independently, or together. In the series configuration, the engine is connected to a generator, and the wheels are only connected to the electric motor.

Ingenious systems have been developed by Volkswagen, in which the electric motor/generator is integrated with the flywheel; see Walzer (1990). However, the greatest potential is offered by dual or split hybrid systems which combine elements of both the series and hybrid systems (Yamaguchi, 1997). During normal driving, some of the engine power can be sent via the generator and electric motor to a differential type gearbox, so that adjusting the electric motor speed effectively controls the overall gearing ratio from the engine to the wheels.

Perhaps the greatest challenge to internal combustion engines will come from fuel cells – devices that allow the direct conversion of fuels to electricity. These are of sufficient importance to be the subject of the next section.

1.6 Fuel cells

Fuel cells are electrochemical devices that permit the direct conversion of chemical energy to electricity, with the potential of a very high efficiency. As fuel cells are open-circuit devices, their maximum efficiency is not limited by their operating temperature and the Carnot cycle efficiency. This will become clearer in chapter 2, section 2.1, after the necessary thermodynamics has been developed. It will be seen shortly that fuel cells have quite specific fuel requirements, and an appreciation of how these can be obtained from conventional fuel cells will become clear by studying chapter 3.

The first fuel cell was built by Sir William Grove in 1839, who had recognised the occurrence of ‘reverse electrolysis’. The fuel cell used oxygen and hydrogen with platinum electrodes. The requirement for pure oxygen and pure hydrogen is a major inconvenience, and the most notable applications of fuel cells have been in space, where the fuels are available for propulsion. There are currently four basic fuel cell types with significant commercial potential:

SPFC	Solid Polymer Fuel Cells
PAFC	Phosphoric Acid Fuel Cells
MCFC	Molten Carbonate Fuel Cells, and
SOFC	Solid Oxide Fuel Cells.

Their operation and operating temperature ranges are summarised in figure 1.14.

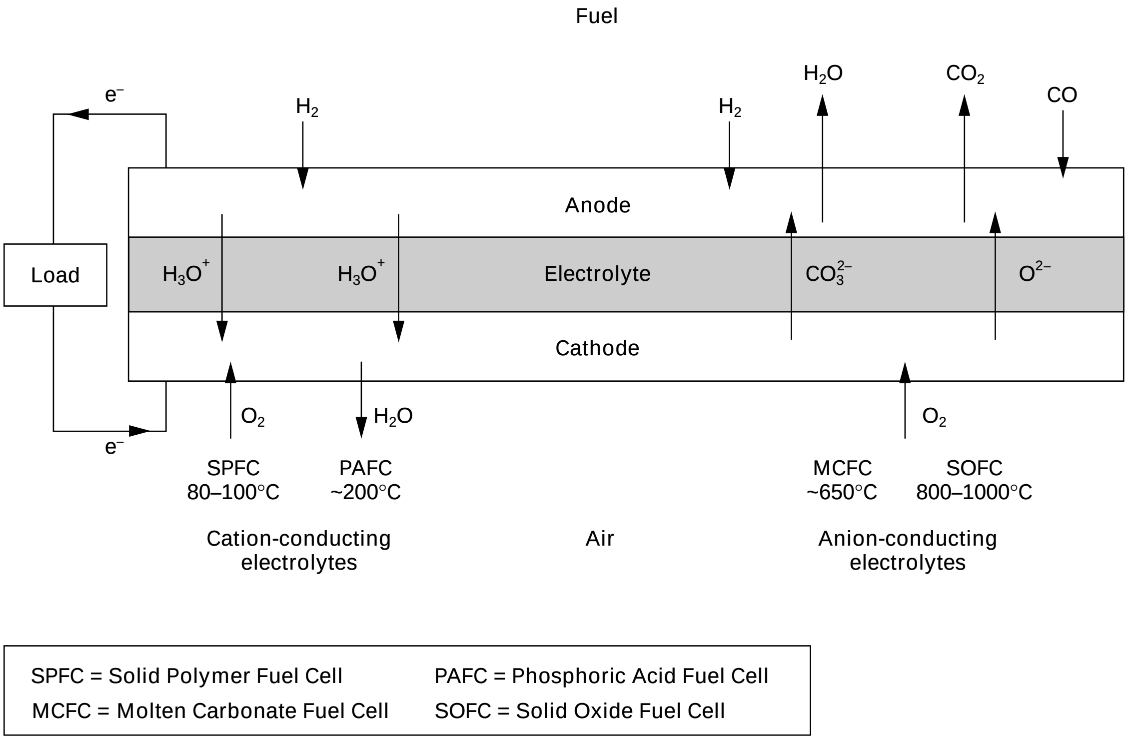


Figure 1.14
Basic fuel cell types (Gardner, 1997). ©IMechE/Professional Engineering Publishing Limited.

The electrolyte is a thin gas-tight ion conducting membrane with porous electrodes on each side. The membrane/electrolyte assembly (MEA) separates the fuel and the oxidant, and allows their chemical interaction. The name of the fuel cell type indicates the nature of the electrolyte, and figure 1.14 indicates a further subdivision into:

1 cation (positively charged ion) conducting electrolytes (SPFC and PAFC), in which the products appear on the air side, and
2 anion (negatively charged ion) conducting electrolytes (MCFC and SOFC), in which the products appear on the fuel side.

The cathode-seeking cation is a hydrated proton (a positively charged hydrogen atom H^+, attached to one or more water molecules). The anode-seeking anion is a carbonate ion (CO_3^{2-}) in the MCFC, and an oxygen ion (O^{2-}) in the SOFC. The SPFC and PAFC both require hydrogen (H_2) as their fuel, while the MCFC and SOFC both have the advantage that carbon monoxide (CO) can be used as well as hydrogen. In general, the reformation of hydrocarbon fuels yields both carbon monoxide and hydrogen, so it is a thermodynamic disadvantage if only the hydrogen can be utilised. Each fuel will produce a different open-circuit voltage, and it will be the lower voltage that will govern. Fortunately, in the case of carbon monoxide and hydrogen, these theoretical voltages are very close in the range of 800–900°C, so the SOFC is able to minimise this loss of electrical energy.

Electrical energy is extracted from the fuel cell, when the free electrons pass through an external load circuit, and not through the electrolyte. The electrolyte has to permit the passage of the ions (driven by the chemical potential arising from the oxidation of the fuel), but prevent the passage of the electrons.

The stationary and non-stationary applications of fuel cells are likely to employ different technologies. For vehicular applications there is limited scope for utilising 'waste heat', so the low-temperature solid polymer fuel cell (SPFC) is likely to be used. An efficiency (based on the lower calorific value of a hydrocarbon fuel) of about 30–40 per cent is likely. Although the fuel cell is at a comparatively low temperature (80–100°C), if on-board reformation of a hydrocarbon or oxygenate fuel is to be used to supply the hydrogen, then temperatures in the region of 200–300°C are required for the reformation of methanol, and temperatures in the region of 800°C would be needed for the reformation of hydrocarbons.

For stationary applications the 'waste heat' can be used, and this makes the solid oxide fuel cell (SOFC) the most attractive, since it operates at the highest temperature. Another advantage of high-temperature operation is that there is scope for internally reforming the fuel, so that (for example) methane (CH_4) can be the feedstock. In the SOFC the

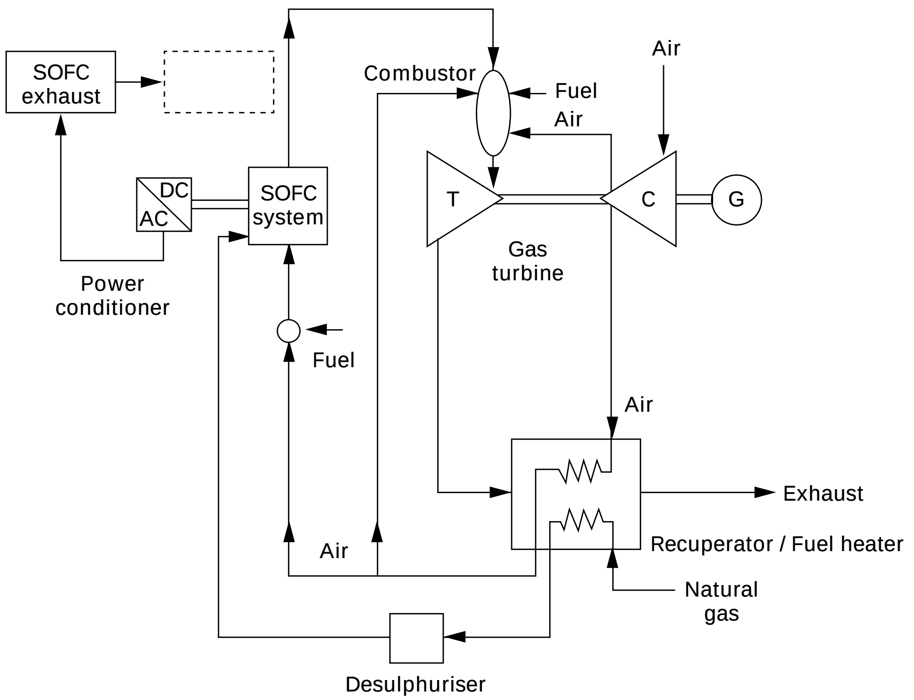


Figure 1.15
An integrated solid oxide fuel cell (SOFC)/gas turbine system (adapted from Bevc, 1997).

electrolyte is a ceramic, so great care is needed in the design to allow for the ceramic's brittle nature, and it is also necessary to avoid thermal shocks.

The power density of all fuel cells can be increased by operation at elevated pressures, and in the case of high-temperature fuel cells this can lead to direct integration with a gas turbine system. Figure 1.15 shows such a system that is discussed in detail by Bevc (1997). The waste heat from the turbine is used to heat both the fuel and the air, and some of this air goes directly to the gas turbine combustion chamber. The SOFC is designed to operate at pressures of up to 10 bar. The exhaust from the SOFC will contain some unoxidised fuel (CO and H_2), and some unreformed fuel (CH_4), but this oxidation can be completed in the gas turbine combustion chamber. The overall efficiency of such a SOFC/gas turbine system is predicted to be 60–70 per cent, but it will only be an economic proposition if the capital costs of the fuel cell are low enough.

1.7 Concluding remarks

The main application of internal combustion engines in terms of numbers is in automotive transport, and it is very difficult to make fair comparisons between conventional vehicles, hybrid vehicles and those with fuel cells. The results depend on the assumptions about the vehicle, its drive cycle, the type of fuel, and how it has been refined, transported and stored. There have been many 'well-to-wheel' studies, but very comprehensive studies are undertaken under the auspices of the European Commission Joint Research Centre and led by CONCAWE.

Figure 1.16 compares different vehicle systems (for which the different internal combustion engine technologies are reviewed in subsequent chapters):

PISI – Port Injection Spark Ignition (a conventional gasoline engine)
DISI – Direct Injection Spark Ignition (an alternative gasoline engine)
DICI – Direct Injection Compression Ignition (a conventional diesel engine)
DPF – Diesel Particulate Filter (the pressure drop across the filter and the need to regenerate the filter impose a fuel consumption penalty)
CNG – Compressed Natural Gas
C-H_2 – Compressed Hydrogen
L-H_2 – Liquefied Hydrogen

Figure 1.16a shows that diesel engines are more efficient than gasoline, and that vehicle energy consumption is reduced by hybridisation. The well-to-tank energy requirement is very low compared to the calorific value of the fuel, because of the ease of extracting, handling and refining liquid

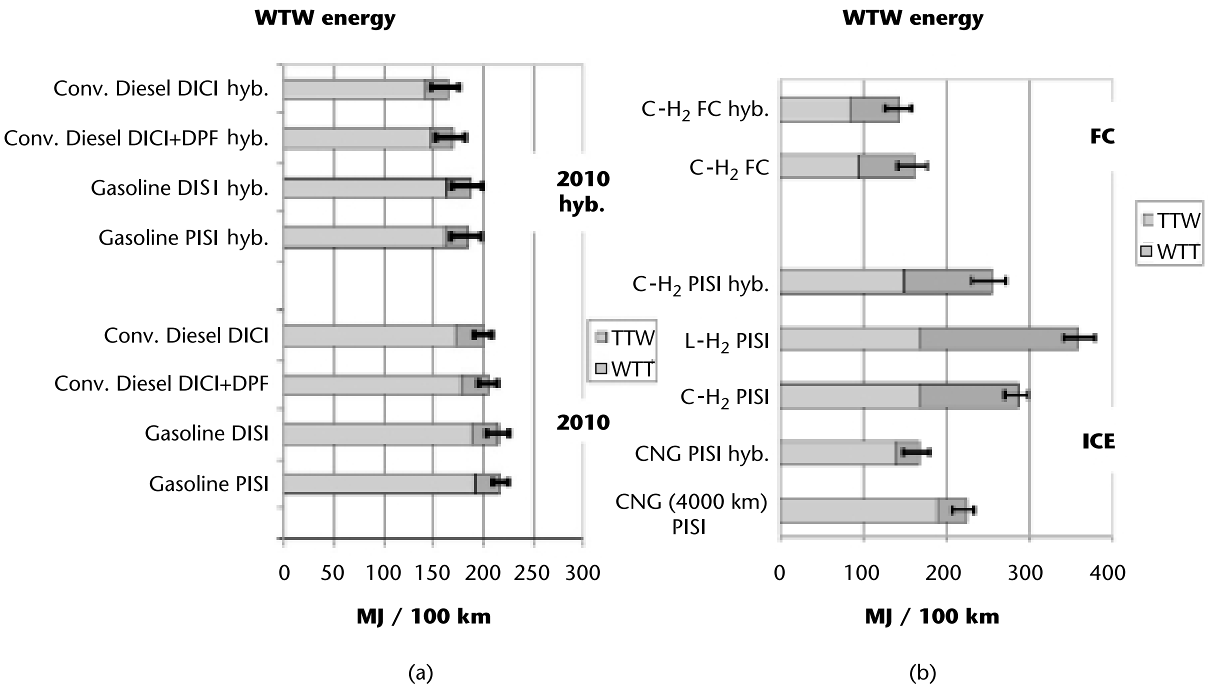


Figure 1.16
Well-to-wheel (WTW) performance of different vehicle systems, with the energy consumption apportioned between fuel production and distribution to the vehicle (WTT – well-to-tank) and vehicle usage (TTW – tank-to-wheel), adapted from CONCAWE (2007). (a) Conventional and hybrid vehicles with liquid fuels. (b) Gaseous fuelled vehicles, with internal combustion engines (ICE) and fuel cells (FC).

fuels. Figure 1.16b shows that Compressed Natural Gas (CNG) has a slightly lower energy consumption in a spark ignition engine than gasoline, but that the energy requirement for the fuel distribution is greater, leading to an overall lower well-to-wheel efficiency. The use of hydrogen with internal combustion engines leads to an overall increase in energy consumption because of the high energy cost in its production and storage. Liquid hydrogen (L-H_2) has a particularly high-energy requirement because at ambient pressure its boiling point is 20 K. Only a compressed hydrogen (C-H_2) fuel cell vehicle has the potential to achieve a lower energy consumption than that for a diesel vehicle. The CONCAWE modelling assumed that hydrogen was being produced from methane (CH_4, the major constituent of Natural Gas). When hydrogen is manufactured from methane (or any hydrocarbon) it is necessary to remove the carbon dioxide, and this could then be stored (‘sequestered’). The disadvantage of fuel cell vehicles would be their very high cost using current technology. The batteries on purely electric vehicles make them very expensive, and with hybrid vehicles the extra cost is divided between the batteries, electric motor(s)/generator(s) and the power electronics.

1.8 Question

1.1 What are the key technological developments for spark ignition engines in the first half of the 20th century?

Index