

## Reciprocating Internal-combustion Engines

Theoretical power cycles are considered in Chapter 5, and the  $p$ - $v$  diagrams analysed are similar to those obtained from actual reciprocating engines. There are, however, fundamental mechanical and thermodynamic differences between the cycles, which make comparison less valuable than might be expected.

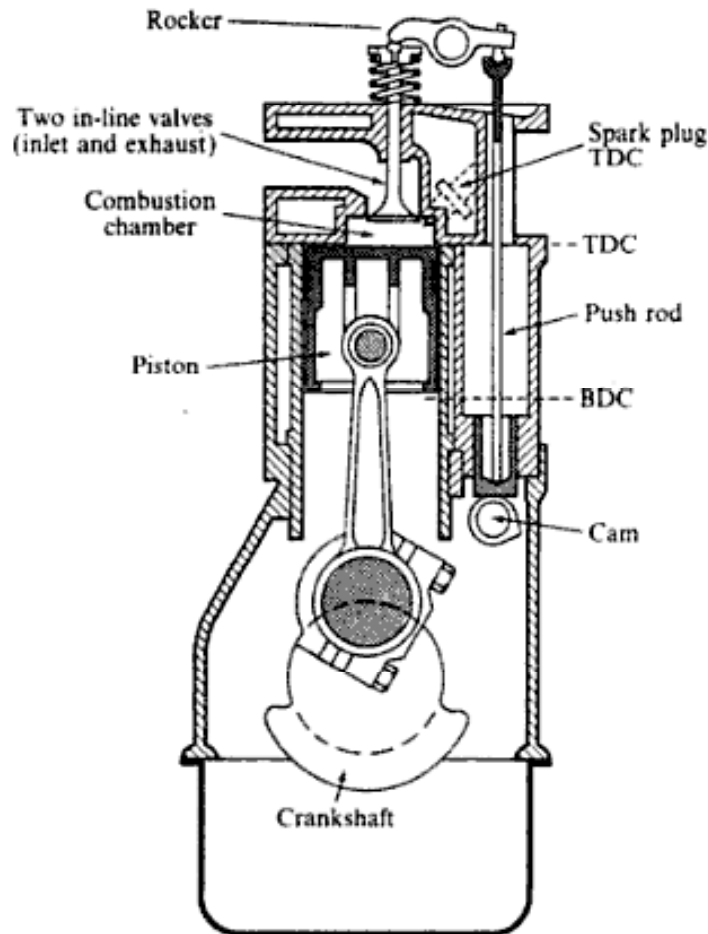
In the theoretical cycles there is no chemical change in the working fluid, which is assumed to be air, and the heat exchanges in the cycle are made externally to the working fluid. In the practical cycle the heat supply is obtained from the combustion of a fuel in air and thus the air charge is consumed during combustion and the combustion products must be exhausted from the cylinder before a fresh charge of air can be induced for the next cycle. The practical cycle consists of the exhaust and induction processes together with the compression and expansion processes as in the theoretical cycle. Further differences between the ideal and the actual cycles are discussed in section 13.8.

The reciprocating engine mechanism consists of a piston which moves in a cylinder and forms a movable gas-tight plug, a connecting-rod and a crankshaft (see Fig. 13.1). If the engine has more than one cylinder then the cylinders, pistons, etc. are identical, and all the connecting-rods are fastened to a common crankshaft. The angular positions of the crank-pins are such that the cylinders contribute their power strokes in a selected and regular sequence. By means of this arrangement the reciprocating motion of the piston is converted to rotary motion at the crankshaft.

There are many types and arrangements of engines, and some classification is necessary to describe a particular engine adequately. The methods of classification are as follows:

- (i) By the fuel used and the way in which the combustion is initiated. Petrol engines and gas engines have spark ignition (SI). Diesel engines or oil engines have compression ignition (CI). In the SI engine the air and the fuel are mixed before compression. In the CI engine the air only is compressed, and the fuel is injected into the air which is then at a sufficiently high temperature to initiate combustion.
- (ii) By the way in which the cycle of processes is arranged. This is defined by the number of complete strokes of the piston required for one complete

**Fig. 13.1** Reciprocating IC engine



cycle. The stroke of the piston is the distance it moves from the position most extreme from the crankshaft to that nearest it. This takes place over half a revolution of the crankshaft. In petrol engine practice the extreme positions of the piston are referred to as *top dead centre* (TDC), and *bottom dead centre* (BDC) (see Fig. 13.1). In oil-engine practice they are referred to as *outer dead centre* and *inner dead centre* respectively. An engine which requires four strokes of the piston (i.e. two revolutions of the crankshaft) to complete its cycle is called a *four-stroke cycle engine*. An engine which requires only two strokes of the piston (i.e. one crankshaft revolution) is called a *two-stroke cycle engine*.

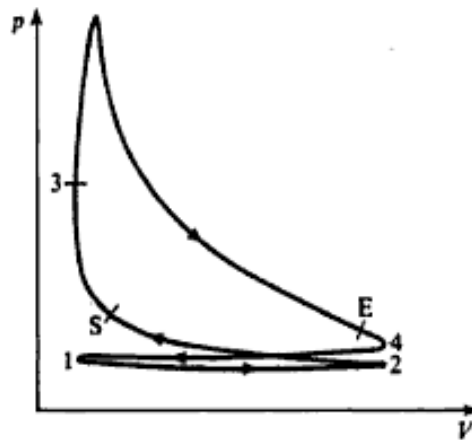
In all reciprocating internal-combustion (IC) engines the gases are induced into and exhausted from the cylinder through ports, the opening and closing of which are related to the piston position. In a two-stroke engine the ports can be opened or closed by the piston itself, but in the four-stroke engine a separate shaft, called the camshaft, is required; this is driven from the crankshaft through a 2 to 1 speed reduction. The cams on this shaft operate valves, called poppet valves, either directly or by means of push rods. Modern high-speed petrol engines have two camshafts, one operating the exhaust valves, and the other operating the inlet valves. The timing of the valves and the point of ignition are fundamental to the engine performance, and the specified timing

is a result of compromise between the many factors involved and is determined empirically. The beginning and end of each process does not coincide with the TDC and BDC positions, although nominally each process may be associated with a piston stroke. The timing of the valves can be indicated on a  $p$ - $V$  diagram, but is more conveniently represented by means of a timing diagram (see Fig. 13.3) in terms of crankshaft angle.

### 13.1 Four-stroke cycle

Figure 13.2 shows a typical  $p$ - $V$  diagram for a SI petrol engine. The individual strokes are given under the headings below.

**Fig. 13.2** Pressure-volume diagram for a SI engine



#### Induction stroke, 1-2

The air-plus-fuel charge is induced into the cylinder as the piston moves from TDC to BDC. Due to the movement of the piston the pressure in the cylinder is reduced to a value between the atmospheric pressure, and air flows through the induction system because of this pressure difference. On its way to the cylinder the air passes through the carburettor in which the metered amount of petrol is added to the air. Nominally the inlet valve closes at point 2, but in fact this does not occur until the piston has moved part of the way along the return stroke.

#### Compression stroke, 2-3

With both valves closed the charge is compressed by the piston. At the TDC position the charge occupies the volume above the piston, which is called the clearance volume, and consists mainly of the volume of the combustion chamber. The spark is timed to occur at a point such as S, which is before TDC. There is a time delay between S and the actual commencement of combustion. The combustion process occurs mainly at almost constant volume, and there is a large increase in pressure and temperature of the charge during this process.

### Working stroke, 3–4

The hot high-pressure gas expands, pushing the piston down the cylinder. It would appear that this expansion should proceed to completion at 4, but in order to assist in exhausting the gaseous products the exhaust valve opens at some point E which is before BDC. At this point the pressure is about 3.5 bar or higher and about 60% of the gas is exhausted between E and 4 as the pressure in the cylinder falls to nearly atmospheric pressure.

### Exhaust stroke, 4–1

The returning piston clears the swept volume of exhaust gas, and the pressure during this stroke is slightly higher than atmospheric pressure. In a normally aspirated engine as described, the clearance volume cannot be exhausted, and at the commencement of the next cycle this volume is full of exhaust gas at about atmospheric pressure. The mixture which is compressed thus consists of the fresh air plus fuel mixture, diluted by a quantity of exhaust gas from the previous cycle.

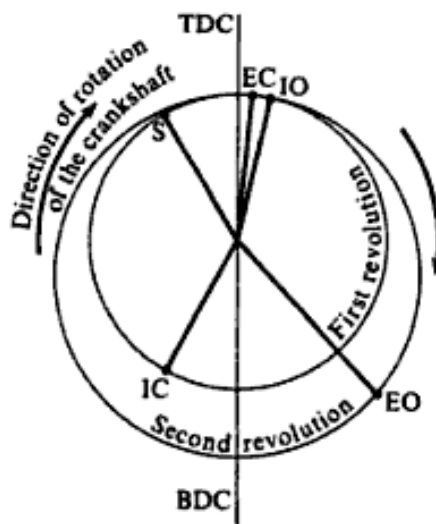
It should be remembered that the maximum volume of fresh charge which can be induced is equal to the swept volume,  $V_s$ , but the actual mass induced in practice is less than the maximum possible, for reasons which will be considered later.

### Timing diagrams

A typical timing diagram for a four-stroke petrol engine is shown in Fig. 13.3, and the angular positions in terms of crank angle position are quoted in relation to the TDC and BDC positions of the piston. The points on the diagram are as follows:

- IO Inlet valve opens. The actual position is between  $10^\circ$  before TDC and  $15^\circ$  after TDC.

Fig. 13.3 Timing diagram for a four-stroke SI engine





- IC Inlet valve closes. This occurs  $20\text{--}40^\circ$  after BDC to take advantage of the momentum of the rapidly moving gas.
- S Spark occurs. This is  $20\text{--}40^\circ$  before TDC when the ignition is fully advanced, and is at TDC when the ignition is fully retarded.
- EO Exhaust valve opens. The average value of this position is about  $50^\circ$  before BDC, but it is greater than this in racing-car engines.
- EC Exhaust valve closes. This occurs  $0^\circ$  to  $10^\circ$  after TDC.

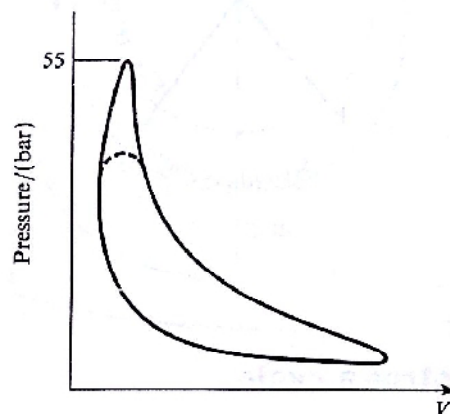
There may be an overlap between IO and EC such that both valves are open at the same instant.

It should be appreciated that the values of angular position quoted are average ones, and considerable differences occur between different engines. The points shown are the normal opening and closing positions for each valve. The time required to open and close the valves means that each valve will be fully open for a crank angle movement much less than that indicated by the timing diagram. Another point to be borne in mind is that at the dead centre positions a considerable amount of crank movement produces a small corresponding movement of the piston; this is shown by a consideration of crank-connecting rod geometry.

Crank angle displacements can be translated into time values if the engine rotational speed is known and can be assumed to be constant.

The  $p$ - $V$  diagram for a CI engine with mechanical or solid injection of the fuel is shown in Fig. 13.4. The ideal cycle for the engine is the dual combustion cycle (see section 5.8) in which the heat input is partly at approximately constant volume, and partly at approximately constant pressure. With modern high-speed engines the fuel injection is well advanced and the greater proportion of fuel is burnt at approximately constant volume. A diagram from the same engine with a later point of injection would be flatter at the top, as indicated by the dotted line in Fig. 13.4. The shape of the diagram is also influenced by the design of the combustion chamber.

Fig. 13.4 Pressure-volume diagram for a CI engine



With engines having the fuel injected by means of an air blast, the  $p$ - $V$  diagram is similar to that shown in Fig. 13.5. In this cycle the air which enters the cylinder with the fuel on injection helps to maintain a constant pressure over the early part of the return stroke, and this increases the area of the



**Fig. 13.5** Pressure-volume diagram for a blast-injection CI engine

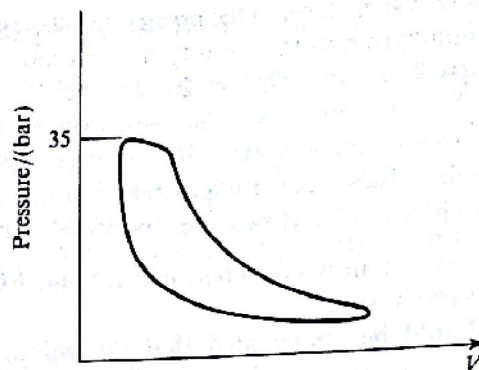
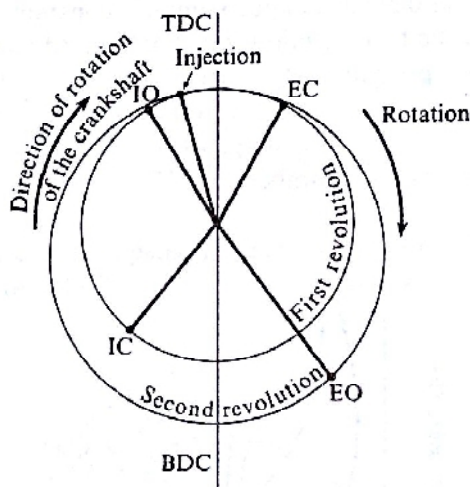


diagram. This method of injection is now practically obsolete due to the difficulties and cost of supplying the high-pressure air required.

Figure 13.6 shows a typical timing diagram for a four-stroke oil engine which has average values for the valve positions as follows:

|           |                       |
|-----------|-----------------------|
| IO        | Up to 30° before TDC. |
| IC        | Up to 50° after BDC.  |
| EO        | About 45° before BDC. |
| EC        | About 30° after TDC.  |
| Injection | About 15° before TDC. |

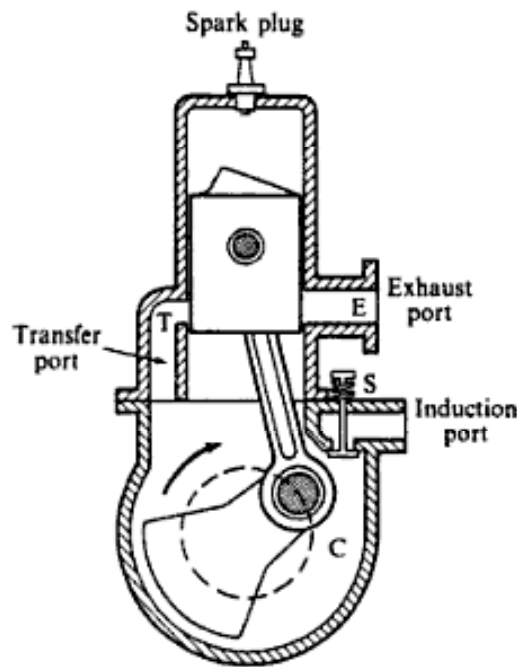
**Fig. 13.6** Timing diagram for a four-stroke CI engine



## 13.2 Two-stroke cycle

Figure 13.7 represents the cylinder of a two-stroke petrol engine with crankcase compression. As the piston ascends on the compression stroke the next charge is drawn into the crankcase, C, through the spring-loaded automatic valve, S. Ignition occurs before TDC, and at TDC the working stroke begins. As the piston descends through about 80% of the working stroke, the exhaust port,

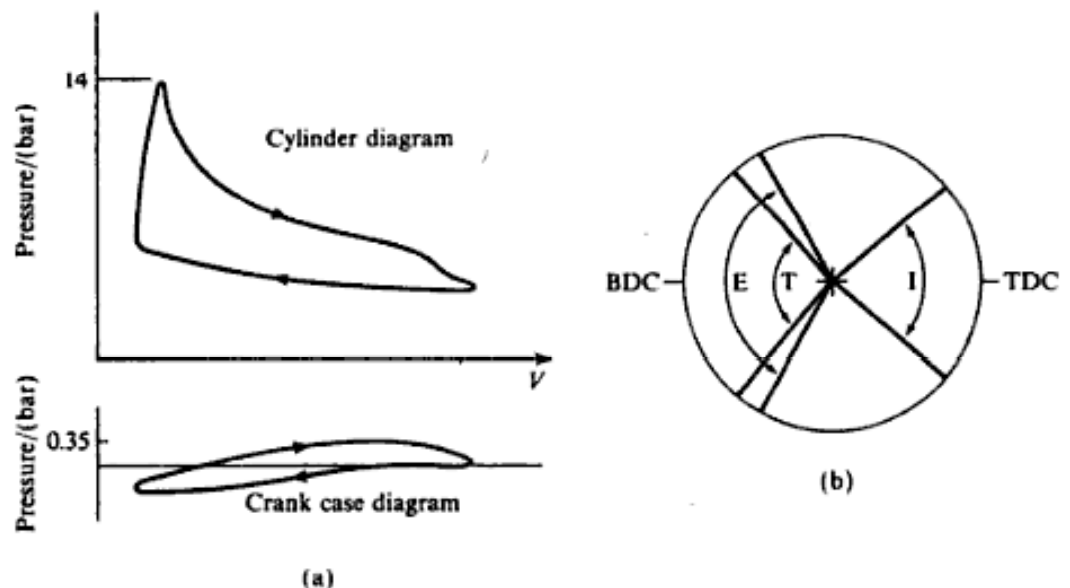
**Fig. 13.7** Two-stroke SI engine with crankcase compression



E, is uncovered by the piston and exhaust begins. The transfer port, T, is uncovered later in the stroke due to the shape of the piston or the position of the port in relation to the port E, and the charge in the crankcase, C, which has been compressed by the descending piston, enters the cylinder through the port T.

The piston can be shaped to deflect the fresh gas across the cylinder to assist the scavenging of the cylinder; this is called *cross-flow scavenging*. As the piston rises, the transfer port, T, is closed slightly before the exhaust port E, and after E is closed compression of the charge in the cylinder begins. The  $p$ - $V$  diagram and the timing diagram for a two-stroke petrol engine are shown in Fig. 13.8(a) and (b).

**Fig. 13.8** Pressure-volume (a) and timing (b) diagrams for a two-stroke SI engine. (I) inlet angle ( $80^\circ$  approx.); (E) exhaust angle ( $120^\circ$  approx.); (T) transfer angle ( $100^\circ$  approx.)



Instead of the spring-loaded valve, S, a design with a third port may be used. This is an induction port controlled by the piston, and through which the mixture is drawn into the crankcase.

The above description of the two-stroke cycle applies also to CI engines with the exception that air only is compressed, and the sparking plug is replaced by a fuel injector.

Crankcase compression has been described but the scavenging and charging of the cylinder may be achieved by other means. A separately phased pump cylinder with its piston driven from the crankshaft may be used. A positive displacement compressor or blower driven from the engine is a third way of charging the cylinder.

The deflector piston, which is unbalanced and can cause 'rattle', may be dispensed with and a flat piston used. The scavenging is then obtained by using two transfer ports which divert the incoming air up the cylinder. This is called 'reverse flow', or 'inverted flow', and the system is called *loop scavenge*.

In engines which have simple inlet ports and poppet or sleeve valve controlled exhaust ports, the inlet and exhaust ports are placed at opposite ends of the cylinder and the fresh charge sweeps along the cylinder towards the exhaust port. This is called *uni-flow scavenge* and is applied with great mechanical simplicity in opposed piston engines.

For several reasons the two-stroke cycle has more application in the CI field than in the SI field, especially for stationary constant-speed engines. In such engines a number of ingenious arrangements have been patented in an attempt to dispense with the scavenge blower. Some designs have used the Kadenacy effect, which employs the high vacuum created by suddenly releasing the exhaust gas through large-area, sharp-edged ports. With constant-speed engines it is possible to 'tune' the exhaust system such that the HP exhaust gas leaving one cylinder can be used to 'pack' another cylinder which is on the early part of its induction stroke.

### 13.3 Other types of engine

In the early days of development of the four-stroke engine one of the difficulties was the noisy poppet valve mechanism. As an alternative the *sleeve valve* became popular. A sliding sleeve is fitted in between the piston and the cylinder, the movement of the sleeve being controlled by an overhung crank-pin driven from a shaft at half crankshaft speed. The movement of this valve controls the inlet and exhaust ports in the cylinder.

Engines have been developed which are called *multi-fuel engines*; such engines will run on any petroleum fuel from diesel fuel to premium petrol. The main application of such engines is for military purposes and it is unlikely that they will have extensive commercial application.

The *dual-fuel engine* is of considerable industrial interest. Some diesel engines, naturally aspirated or turbocharged (see section 13.12) can be used as dual-fuel engines. They are started on diesel oil and then run on an available gaseous fuel such as methane, natural gas, sewage gas, coal gas, etc. The combustion process requires a pilot injection of oil which amounts to 7–10% of the full

power supply when running as a diesel engine. The changeover from diesel fuel to gas can be done automatically or manually. To supply the pilot injection of oil a second set of pumps is required which delivers fuel to the standard injectors.

A considerable amount of effort has been put into the development of the *free piston engine*. In this type of engine the crankshaft and connecting-rod are dispensed with, and two opposed but connected pistons operating in the same cylinder are used. The free piston engine is described more fully in section 13.14. One of the main applications of the engine is that instead of having a separate air compressor driven by an IC engine the compressor and the engine can be combined with the result that the intermediate rotating shafts are eliminated and a more compact unit is obtained. This is especially important for portable air compressors which are used extensively. Another use for the free piston engine is as a 'gasifier' from which the gaseous products of combustion are exhausted at a suitable pressure and allowed to expand through a gas turbine. No power output is taken from the free piston engine, the power output of the unit being that obtained from the turbine. The potential field for the smaller size free piston engine is regarded as being with road, rail, and tracked vehicles, earth-moving equipment, high-speed marine craft, cargo ships, and in generating stations.

Some classification of IC engines has been given, but this is not exhaustive. The applications for such engines are wide, both in the type of duty to be performed and in the power required. Modern developments promise the application of IC engines not only as individual units but as part of a complete plant to suit some specialized purpose.

## 13.4 Criteria of performance

An engine is selected to suit a particular application, the main consideration being its power/speed characteristics. Important additional factors are initial capital cost and running cost. In order that different types of engines or different engines of the same type may be compared, certain performance criteria must be defined. These are obtained by measurement of the quantities concerned during bench tests, and calculation is by standard procedures. The results are plotted graphically in the form of performance curves.

### Indicated power (ip)

This is defined as the rate of work done by the gas on the piston as evaluated from an indicator diagram obtained from the engine. An indicator diagram has the form shown in Fig. 13.9. Figure 13.9 shows both the power and the pumping loops.

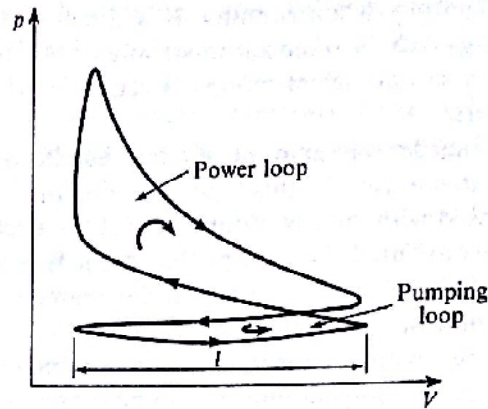
The mean effective pressure has been defined in section 5.9 and may be applied here.

Net work done per cycle  $\propto$  (area of power loop — area of pumping loop)



## Reciprocating Internal-combustion Engines

Fig. 13.9 Pressure-volume diagram for a reciprocating engine



Therefore indicated mean effective pressure,  $p_i$ , is given by

$$p_i = \frac{\text{net area of diagram}}{\text{length of diagram}} \times \text{constant} \quad (13.1)$$

(the constant depends on the scales of the recorder).

Considering one engine cylinder

$$\text{Work done per cycle} = p_i \times A \times L$$

where  $A$  is the area of piston and  $L$  the length of stroke.

$$\text{Power output/unit time} = \text{work done per cycle} \times \text{cycles per minute}$$

$$\text{or} \quad ip = p_i AL \times (\text{cycles/unit time})$$

The number of cycles per unit time depends on the type of engine; for four-stroke engines the number of cycles per unit time is  $N/2$ , and for two strokes the number of cycles per unit time is  $N$ , where  $N$  is the engine speed. The formula for  $ip$  then becomes for four-stroke engines

$$ip = \frac{p_i ALNn}{2} \quad (13.2)$$

for two-stroke engines

$$ip = p_i ALNn \quad (13.3)$$

where  $n$  is the number of cylinders.

The main recorder in use is the electronic engine indicator, which uses a cathode-ray oscilloscope (CRO) to display the pressure trace obtained (see Fig. 13.10).

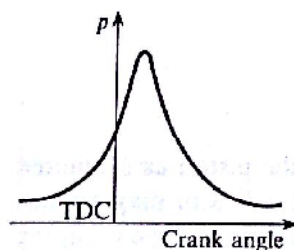


Fig. 13.10 Pressure against crank angle for a typical reciprocating IC engine

Modern methods of instrumentation, data acquisition, and analysis, based on recent advances in electronics, digital methods, and the application of microprocessors, have become applicable to engine testing, development, and control. Engine test and control systems have become increasingly 'computerized' and it is regretted that space requirements do not allow a comprehensive treatment of this very important aspect of engineering in this book. Readers are advised to obtain copies of the excellent publications by engine and equipment supplies which will give a state-of-the-art description of the latest equipment and techniques.

### Brake power (bp)

This is the measured output of the engine. The engine is connected to a brake or dynamometer which can be loaded in such a way that the torque exerted by the engine can be measured. The dynamometer may be of the absorption or the transmission type. Absorption dynamometers are the more usual and can be classified as: (i) friction type, used for the smaller powered, lower-speed engines; (ii) hydraulic; (iii) electrical; (iv) air-fan type. With types (i), (ii), and (iii), the torque is obtained by reading off a net load,  $W$ , at a known radius,  $R$ , from the axis of rotation, and hence the torque,  $T$ , is given by

$$T = WR \quad (13.4)$$

The brake power is then given by

$$bp = 2\pi NT \quad (13.5)$$

In the transmission type of dynamometer the torque,  $T$ , transmitted by the driving shaft is measured directly, and the bp is obtained by substitution in equation (13.5). With air fans the torque is obtained from a calibration curve for the fan.

### Friction power (fp) and mechanical efficiency, $\eta_M$

The difference between the ip and the bp is the friction power (fp), and is that power required to overcome the frictional resistance of the engine parts,

$$\text{i.e.} \quad fp = ip - bp \quad (13.6)$$

The mechanical efficiency of the engine is defined as

$$\text{Mechanical efficiency, } \eta_M = \frac{bp}{ip} \quad (13.7)$$

where  $\eta_M$  usually lies between 80 and 90%.

The fp is very nearly constant at a given engine speed; if the load is decreased giving lower values of bp, then the variation in  $\eta_M$  with bp is as shown in Fig. 13.11. At zero bp at the same speed the engine is developing just sufficient power to overcome the frictional resistance.

Mechanical efficiency depends on the ip and bp, and is therefore found by evaluating these experimentally. A considerable amount of literature has been published dealing with mechanical efficiency and the analysis of engine power losses. A useful report on this subject is that of ref. 13.1, and the conclusion is that no single method is satisfactory in every respect for the evaluation of mechanical efficiency. The four main methods are as follows:

- (i) Measurement of the ip and the bp by the means already described in the preceding subheadings.
- (ii) Measurement of the bp at a given speed followed by 'motoring' of the engine with the fuel supply cut off. This method can only be used on engines

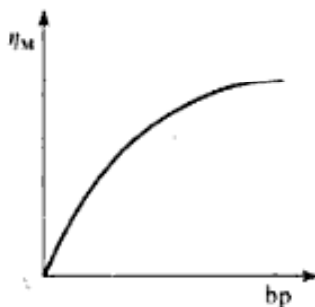


Fig. 13.11 Variation of mechanical efficiency with brake power

with an electrical dynamometer, the dynamometer being used as a motor instead of as a generator in order to motor the engine at the firing speed. The torque can be measured under firing and under motoring conditions and the mechanical efficiency evaluated. The fact that an electrical dynamometer can be used to find the mechanical efficiency in this way is one of the main advantages of this type of dynamometer.

- (iii) The Morse test: this is only applicable to multicylinder engines. The engine is run at the required speed and the torque is measured. One cylinder is cut out, by shorting the plug if an SI engine is under test, or by disconnecting an injector if a CI engine is under test. The speed falls because of the loss of power with one cylinder cut out, but is restored by reducing the load. The torque is measured again when the speed has reached its original value. If the values of  $ip$  of the cylinders are denoted by  $I_1, I_2, I_3$ , and  $I_4$  (considering a four-cylinder engine), and the power losses in each cylinder are denoted by  $L_1, L_2, L_3$ , and  $L_4$ , then the value of  $bp$ ,  $B$ , at the test speed with all cylinders firing is given by

$$B = (I_1 - L_1) + (I_2 - L_2) + (I_3 - L_3) + (I_4 - L_4)$$

If number 1 cylinder is cut out, then the contribution  $I_1$  is lost; and if the losses due to that cylinder remain the same as when it is firing, then the  $bp$ ,  $B_1$ , now obtained at the same speed is

$$B_1 = (0 - L_1) + (I_2 - L_2) + (I_3 - L_3) + (I_4 - L_4)$$

Subtracting the second equation from the first gives

$$B - B_1 = I_1 \quad (13.8)$$

By cutting out each cylinder in turn the values  $I_2, I_3$ , and  $I_4$  can be obtained from equations similar to (13.8). Then for the engine

$$I = I_1 + I_2 + I_3 + I_4 \quad (13.9)$$

- (iv) 'Willan's line': this method is applicable to CI engines only. At a constant engine speed the load is reduced in increments and the corresponding  $bp$  and gross fuel consumption readings are taken. A graph is then drawn of fuel consumption against  $bp$ , as in Fig. 13.12. The graph drawn is called the 'Willan's line', and is extrapolated back to cut the  $bp$  axis at the point A. The reading  $OA$  is taken as the power loss of the engine at that speed. The fuel consumption at zero  $bp$  is given by  $OB$ ; if the relationship between fuel consumption and  $bp$  is assumed to be linear, then a fuel consumption  $OB$  is equivalent to a power loss of  $OA$ .

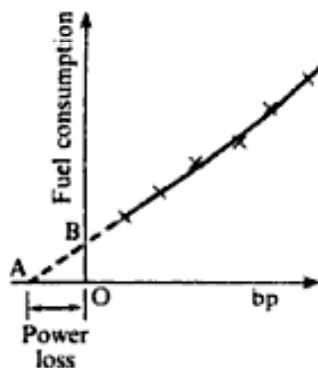


Fig. 13.12 Fuel consumption against brake power for a CI engine

### Brake mean effective pressure (bmep), thermal efficiency, and fuel consumption

The  $bp$  of an engine can be obtained accurately and conveniently using a dynamometer.

From equation (13.7)

$$bp = \eta_M \times ip$$



Therefore, substituting for  $i_p$  from equation (13.2) for a four-stroke engine, we have

$$bp = \frac{\eta_M \times p_i ALNn}{2}$$

Since  $\eta_M$  and  $p_i$  are difficult to obtain they may be combined and replaced by a brake mean effective pressure,  $p_b$ ,

$$\text{i.e. } bp = \frac{p_b ALNn}{2} \quad (13.10)$$

where  $p_b = \eta_M \times p_i$ .

The bmep may be thought of as that mean effective pressure acting on the pistons which would give the measured bp if the engine were frictionless. The bmep is a useful criterion for comparing engine performance. Taking the two equations for bp, equations (13.5) and (13.10), and putting them together we have

$$\frac{p_b ALNn}{2} = 2\pi NT$$

therefore

$$p_b = K \times T$$

where  $K$  is a constant.

Therefore bmep is directly proportional to the engine torque and is independent of the engine speed.

The power output of the engine is obtained from the chemical energy of the fuel supplied. The overall efficiency of the engine is given by the *brake thermal efficiency*,  $\eta_{BT}$ ,

$$\text{i.e. } \eta_{BT} = \frac{\text{brake work}}{\text{energy supplied}}$$

therefore

$$\eta_{BT} = \frac{bp}{\dot{m}_f \times Q_{net,v}} \quad (13.11)$$

where  $\dot{m}_f$  is the mass of fuel consumed per unit time, and  $Q_{net,v}$  the net calorific value of the fuel.

The *specific fuel consumption* (sfc) is the mass flow rate of fuel consumed per unit power output, and is a criterion of economical power production,

$$\text{i.e. } \text{sfc} = \frac{\dot{m}_f}{bp} \quad (13.12)$$

The *indicated thermal efficiency*,  $\eta_{IT}$ , is defined in a similar way to  $\eta_{BT}$ ,

$$\text{i.e. } \eta_{IT} = \frac{i_p}{\dot{m}_f \times Q_{net,v}} \quad (13.13)$$

Dividing equation (13.11) by equation (13.13) gives

$$\frac{\eta_{BT}}{\eta_{IT}} = \frac{bp}{ip} = \eta_M$$

therefore

$$\eta_{BT} = \eta_M \times \eta_{IT} \quad (13.14)$$

### Example 13.1

A four-cylinder petrol engine has a bore of 57 mm and a stroke of 90 mm. Its rated speed is 2800 rev/min and it is tested at this speed against a brake which has a torque arm of 0.356 m. The net brake load is 155 N and the fuel consumption is 6.74 l/h. The specific gravity of the petrol used is 0.735 and it has a lower calorific value,  $Q_{net,v}$  of 44 200 kJ/kg. A Morse test is carried out and the cylinders are cut out in the order 1, 2, 3, 4, with corresponding brake loads of 111, 106.5, 104.2, and 111 N, respectively. Calculate for this speed, the engine torque, the bmep, the brake thermal efficiency, the specific fuel consumption, the mechanical efficiency and the imep.

**Solution** Using equation (13.4)

$$\text{Torque, } T = WR = 155 \times 0.356 = 55.2 \text{ N m}$$

Using equation (13.5),

$$bp = 2\pi NT = \frac{2\pi \times 2800 \times 55.2}{60 \times 10^3} \text{ kN m/s} = 16.2 \text{ kW}$$

From equation (13.10),

$$\begin{aligned} \text{bmep} &= \frac{bp \times 2}{ALNn} = \frac{16.2 \times 2 \times 4 \times 60 \times 10^3}{\pi \times 0.057^2 \times 0.09 \times 2800 \times 4 \times 10^5} \\ &= 7.55 \text{ bar} \end{aligned}$$

Using equation (13.11)

$$\eta_{BT} = \frac{bp}{\dot{m}_f \times Q_{net,v}} = \frac{16.2}{0.001376 \times 44200} = 0.266 \text{ or } 26.6\%$$

where  $\dot{m}_f = (6.74/3600) \times 1 \times 0.735 = 0.001376 \text{ kg/s}$ .

Using equation (13.12),

$$\text{sfc} = \frac{\dot{m}_f}{bp} = \frac{0.001376}{16.2} = 0.000085 \text{ kg/kJ}$$

It is more convenient to express sfc in terms of fuel consumption rate per unit power and to express the fuel consumption rate in kilograms per hour rather than kg/s,

i.e.  $\text{sfc} = 0.000085 \times 3600 = 0.306 \text{ kg/kW h}$

Using equation (13.8) for each cylinder in turn, and substituting brake loads

instead of the values of  $b_p$  since the speed is constant, we have

$$I_1 = B - B_1 = 155 - 111 = 44 \text{ N}$$

$$I_2 = B - B_2 = 155 - 106.5 = 48.5 \text{ N}$$

$$I_3 = B - B_3 = 155 - 104.2 = 50.8 \text{ N}$$

$$I_4 = B - B_4 = 155 - 111 = 44 \text{ N}$$

Hence for the engine, the indicated load,  $I$ , is given by

$$I = I_1 + I_2 + I_3 + I_4 = 44 + 48.5 + 50.8 + 44 = 187.3 \text{ N}$$

Therefore from equation (13.7)

$$\eta_M = \frac{b_p}{i_p} = \frac{155}{187.3} = 0.828 \text{ or } 82.8\%$$

From the definition of  $b_{mep}$  given by equation 13.10, we have

$$b_{mep} = \eta_M \times i_{mep}$$

$$\text{i.e. } i_{mep} = \frac{7.55}{0.828} = 9.12 \text{ bar}$$

### Volumetric efficiency, $\eta_v$

The power output of an IC engine depends directly upon the amount of charge which can be induced into the cylinder. This is referred to as the *breathing capacity* of the engine and is expressed quantitatively by the *volumetric efficiency*, which is defined as for reciprocating compressors by equations (12.14) and (12.15). For IC engines, the volumetric efficiency is the ratio of the volume of air induced, measured at the free air conditions to the swept volume of the cylinder,

$$\text{i.e. } \eta_v = \frac{V}{V_s} \quad (13.15)$$

The power output of an engine depends on its capacity to breathe, and if a particular engine had a constant thermal efficiency then its output would be in proportion to the amount of air induced. The volumetric efficiency with normal aspiration is seldom above 80%, and to improve on this figure, *supercharging* is used. Air is forced into the cylinder by a blower or fan which is driven by the engine. More will be said about supercharging in section 13.12.

The volumetric efficiency of an engine is affected by many variables such as compression ratio, valve timing, induction and port design, mixture strength, specific enthalpy of vaporization of the fuel, heating of the induced charge, cylinder temperature, and the atmospheric conditions.

#### Example 13.2

In Example 13.1 an analysis of the dry exhaust showed no oxygen and negligible carbon monoxide. The engine was tested in an atmosphere at 1.013 bar and 15°C. Estimate the volumetric efficiency of the engine.



**Solution** The condition of the exhaust implies a stoichiometric air–fuel ratio which for petrols can be taken to be 14.5/1.

From Example 13.1

$$\dot{m}_f = 0.001\,376 \text{ kg/s}$$

therefore

$$\text{Air mass flow rate} = 14.5 \times 0.001\,376 = 0.019\,95 \text{ kg/s}$$

therefore

$$\begin{aligned} \text{Volume drawn in per unit time, } \dot{V} &= \frac{0.019\,95 \times 287 \times 288}{10^5 \times 1.013} \\ &= 0.0163 \text{ m}^3/\text{s} \end{aligned}$$

Now

$$\text{Swept volume of engine} = ALn \text{ m}^3/\text{cycle} = \frac{ALnN}{2} \text{ m}^3/\text{min}$$

$$\text{i.e. } \dot{V}_s = \frac{\pi \times 0.057^2 \times 0.09 \times 4 \times 2800}{4 \times 2 \times 60} = 0.0214 \text{ m}^3/\text{s}$$

Then using equation (13.15)

$$\eta_v = \frac{\dot{V}}{\dot{V}_s} = \frac{0.0163}{0.0214} = 0.76 \text{ or } 76\%$$

### 13.5 Engine output and efficiency

The power output of an engine depends on the conditions under which it is tested. In order to make reported performances acceptable and comparable, standard procedures are established which define the quantities to be measured, the methods of measurement to be used, and the procedure to be adopted for reporting. The standards which apply in the UK are defined in British Standard (BS) 5514 (ref. 13.2) which is equivalent to the corresponding standards ISO 3046 of the International Organization for Standardization (ISO). In the United States the standards are those of the Society of Automotive Engineers (SAE) and are described in the *SAE Handbook, 1984*, volume 3 or its replacement. In Germany the standards are those of the Deutsche Industrie Norm (DIN) and in the UK also the power and torque outputs of engines are often quoted to the DIN procedures. Standards are withdrawn or updated from time to time and the different standards may become fewer as a single international standard emerges.

The power output changes with the atmospheric conditions of pressure, temperature, and humidity and if the test conditions are not those of the relevant standard the readings of power and fuel consumption taken must be corrected to the defined conditions and procedures of the standard. The power output



also depends on the form in which the engine is tested with regard to the auxiliaries with which it is fitted, i.e. fully or partly equipped. Diesel engine power, torque and fuel consumption values or complete characteristics are readily available; for petrol engines maximum values of power (kilowatts) and torque (Newton metres) are quoted, but fuel consumption figures are not so readily come by. For private vehicle use the overall vehicle performance figures are quoted as power and torque to the specified standard, usually DIN, and the fuel consumption figures to official tests under the Passenger Car Fuel Consumption Order 1983 in litres/100 km at the specified test speeds of 90 and 120 km/h (56 and 75 mph).

For example, for one particular engine the power (in kilowatts) and torque (in Newton metres) outputs quoted to different test standards were respectively 69 and 359 to BS 5514, 72 and 377 to DIN 70020, 79 and 369 to BS AU 1141a, and 72.6 and 366 to BS 649, showing a considerable variation. This range of variation of the main performance parameters indicates the necessity for a rationalization of engine test standards.

Table 13.1 details the power and torque performances of engines (mainly petrol) to DIN ratings and the fuel consumptions of vehicles in which they are fitted. Some basic engines appear in different forms with different carburation or fuel injection systems, have alternative forms of ignition and may even be supercharged by means of a turbocharger (see section 13.12). Similarly, the same engine may be available in different forms of vehicle, e.g. saloon, estate or cabriolet which have different aerodynamic characteristics with corresponding vehicle performance figures. The comparison of engine power outputs by the criterion power/volume should be done with the understanding that for a given total engine capacity the small capacity multicylinder engine will give a higher

**Table 13.1** Some private vehicle engine and road performances

| Engine size/(litres)<br>[cylinders] | Fuel type/<br>[compression ratio] | Maximum output                  |                                   |                                    | Vehicle fuel consumption<br>(litre/100 m) [mpg] |                      |                             |
|-------------------------------------|-----------------------------------|---------------------------------|-----------------------------------|------------------------------------|-------------------------------------------------|----------------------|-----------------------------|
|                                     |                                   | Power/(kW)<br>[speed/(rev/min)] | Torque/(N m)<br>[speed/(rev/min)] | Power-capacity<br>ratio/(kW/litre) | 56 mph<br>(90 km/h)                             | 75 mph<br>(120 km/h) | Simulated<br>urban<br>drive |
| 0.96 [4]                            | Petrol<br>[8.5]                   | 33 [5750]                       | 68 [3700]                         | 34.5                               | 5.2<br>[54.3]                                   | 7.1<br>[39.8]        | 7.0<br>[40.4]               |
| 1.60 [4]                            | Petrol<br>[8.5]                   | 58 [5800]                       | 125 [3300]                        | 36.3                               | 5.6<br>[50]                                     | 7.6<br>[37]          | 8.9<br>[32]                 |
| 1.60 [4]                            | Petrol<br>FI [8.5]                | 77 [6000]                       | 138 [4800]                        | 48.1                               | 6.2<br>[45.6]                                   | 7.9<br>[37.5]        | 10.3<br>[27.7]              |
| 1.76 [4]<br>(16 valve)              | Petrol<br>[10.0]                  | 99 [6500]                       | 158 [4250]                        | 56.1                               | 6.3<br>[44.8]                                   | 7.6<br>[37.2]        | 10.6<br>[26.6]              |
| 2.30 [V6]                           | Petrol<br>[9.0]                   | 84 [5300]                       | 176 [3000]                        | 36.5                               | 6.8<br>[41.5]                                   | 8.7<br>[32.5]        | 12.6<br>[22.4]              |
| 2.30 [4]                            | Diesel<br>[22.2]                  | 49 [4200]                       | 139 [2000]                        | 21.2                               | 5.05<br>[56]                                    | 6.7<br>[42.2]        | 8.35<br>[33.8]              |
| 2.80 [V6]                           | Petrol<br>FI [9.2]                | 110 [5700]                      | 216 [4000]                        | 39.3                               | 7.75<br>[36.5]                                  | 10.2<br>[27.7]       | 14.9<br>[19]                |
| 2.80 [V6]                           | Petrol<br>TC [9.2]                | 151 [5000]                      | 353 [3500]                        | 54.0                               | —                                               | —                    | —                           |

FI—fuel injection; TC—turbocharged.

value than an engine with fewer, larger cylinders because the volume is swept more frequently for equal piston speeds.

The last two engines listed are different versions of the same engine and are the power ratings for intermittent running. The continuous running characteristics for the same engines are shown in Fig. 13.21 (p. 443).

The specific fuel consumptions at maximum power vary between 0.20 and 0.25 kg/kWh with somewhat lower values (by 0.005 to 0.010 kg/kWh at the maximum torque condition).

The details quoted in Table 13.2 are for the smaller size diesel engines for industrial and commercial vehicle use. Engines in the higher power range, 1000–3000 kW, are in service in fast patrol boats, for electrical power generation on warships, on offshore oil rigs, in rail traction including high-speed trains and in submarine applications.

**Table 13.2** Some diesel engine performances to BS 5514/DIN 6270

| Capacity<br>(litre) | Cylinders | Compression<br>ratio | Breathing            | Max. power/(kW)<br>at speed/(rev/min) | Max. torque/(N m)<br>at speed/(rev/min) | Power<br>(kW/litre) |
|---------------------|-----------|----------------------|----------------------|---------------------------------------|-----------------------------------------|---------------------|
| 2.9                 | 3         | 17.3                 | Normal               | 29 at 2000                            | 158 at 1600                             | 9.9                 |
| 3.3                 | 3         | 16.3                 | Normal               | 43 at 2200                            | 211 at 1400                             | 13.0                |
| 4.4                 | 4         | 16.3                 | Normal               | 57.5 at 2100                          | 284 at 1600                             | 13.0                |
| 4.4                 | 4         | 15.6                 | Turbocharged         | 68.7 at 2100                          | 359 at 1600                             | 15.6                |
| 6.6                 | 6         | 16.3                 | Normal               | 90 at 2300                            | 436 at 1400                             | 13.6                |
| 6.6                 | 6         | 15.6                 | Turbocharged         | 103 at 2200                           | 500 at 1600                             | 15.6                |
| 6.6                 | 6         | 15.6                 | Turbo + intercooling | 130 at 2200                           | 665 at 1600                             | 19.7                |

Diesel engine competition with the petrol engine for the private car market is an interesting study exercise. The principles of the diesel engine were well known at the start of the century and prior to the Second World War a few diesel-engined cars had been built as conversions from petrol engines. The advantages of the diesel of better fuel consumption and longer engine life were well known, particularly for use in taxis which are used for short journeys and have high annual mileages. For such applications the savings in fuel cost favour the diesel engine but its relative progress has been slow. In the late 1970s only about 69 000 diesel private cars and vans were in use in the UK, but the number rose to about 519 000 in 1986, and currently is in excess of 1 300 000 of which over 650 000 are private cars. The performance of an early (1974) diesel-engined vehicle compared favourably with that of a smaller petrol engine, e.g. a 1.8 litre diesel compared with a 1.3 litre petrol engine gave similar journey times and overall fuel costs of about half those of the petrol engine. The noise level of the diesel engine was not acceptable and the diesel was regarded as being sluggish in comparison with the petrol engine. Modern diesel engine design has narrowed the gap between the two with engines becoming lighter, quieter, and more responsive. The better fuel consumption of the diesel makes it an attractive power unit for the taxi and vehicle fleet owner, provided the price of diesel fuel

remains sufficiently lower than that of petrol. Another important factor is the frequency and cost of service work required by the diesel and petrol equivalents. Diesel engines required more frequent servicing, but the intervals between servicing are approaching equality.

## 13.6 Performance characteristics

The testing of IC engines consists of running them at different loads and speeds and taking sufficient measurements for the performance criteria to be calculated. As well as the measurements required for the criteria of section 13.4 the airflow is required to give the air-fuel ratio and the combustion products can be analysed (see section 7.6).

An energy balance is sometimes presented for an engine, and the heat taken by the cooling water is obtained by measuring the rate of flow of the water and its temperature rise. The outlet temperature of the cooling water is usually limited to about 80°C to prevent the formation of steam pockets. To estimate the energy of the exhaust gas, an exhaust calorimeter can be fitted; this is simply a heat exchanger in which the exhaust gas is cooled by circulating water, the rate of flow and temperature rise of which are measured. In order to avoid condensation of the steam in the gas, the gas is not usually cooled below about 50°C.

The items usually included in an energy balance and expressed as percentages of the energy supplied by the fuel (i.e.  $\dot{m}_f \times Q_{\text{net},v}$ ) are as follows: (a) bp; (b) the heat to cooling water; (c) the energy of the exhaust referred to inlet conditions, or as obtained by an exhaust calorimeter; (d) unaccounted losses obtained by difference and which include radiation and convection losses, etc.

The energy balance usually presented is not an accurate account of the energy distribution, but it is a useful one. The bp is conveniently and accurately measured and the percentage of the input energy to the bp is the most important item in the balance. The heat transferred to the cooling water may be used as an indication of how much heat could be usefully obtained if this water were used for other heating purposes in a combined plant.

The energy to exhaust is best obtained by means of the exhaust calorimeter as described. Ideally the exhaust gas should be cooled to the temperature of the inlet air, and the heat taken by the cooling water in the calorimeter per minute would give item (c) of the balance. The temperature at which the exhaust gas enters the calorimeter is most likely not that at which it passes through the exhaust valve, and some of the energy to the exhaust will have been taken by the cooling water or lost to the atmosphere. To obtain item (c) by calculation is more speculative since the gas is chemically different from the inlet air and the mass flow has increased due to the addition of the fuel. The error involved in choosing the datum is likely to be less than that produced by using an inaccurate value for the exhaust temperature, which is not easy to obtain with accuracy. It is sufficient to write

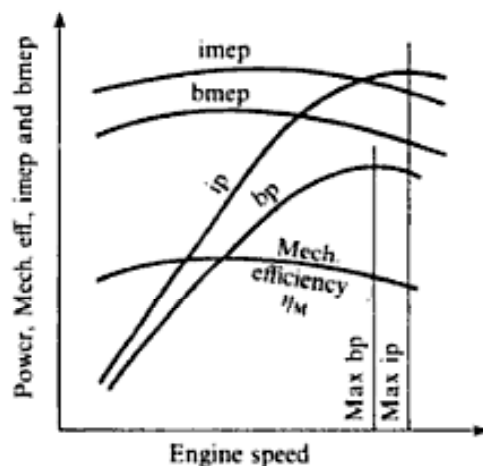
$$\text{Energy to exhaust} = (\dot{m}_a + \dot{m}_f)h_e - \dot{m}_a h_a$$

where  $\dot{m}_a$  and  $\dot{m}_f$  are the air and fuel mass flow rates,  $h_e$  is the specific enthalpy of the exhaust gas (dry exhaust + steam), reckoned from  $0^\circ\text{C}$ , and  $h_a$  is the specific enthalpy of the air at inlet reckoned from  $0^\circ\text{C}$ . A suitable value for  $c_p$  for the dry exhaust gas must be calculated or assumed.

For a diesel engine at full load typical values would be: to bp 35%; to cooling water 20%; to exhaust 35%; to radiation, etc. 10%. The heat to the jacket cooling water and exhaust can be utilized in industries which have heating loads such as space and heating and hot-water systems, and which require either steam or hot water for process work. The heat to the jacket water is recoverable and about 18% of the total energy supplied can be recovered from the exhaust gas.

The most elementary power test is that which gives the power-speed and the torque-speed characteristics, as shown in Fig. 13.13. The test is carried out at constant throttle setting in the petrol engine, and at constant fuel pump setting in the CI engine.

**Fig. 13.13** Engine characteristics of power, imep, bmep, and mechanical efficiency against engine speed

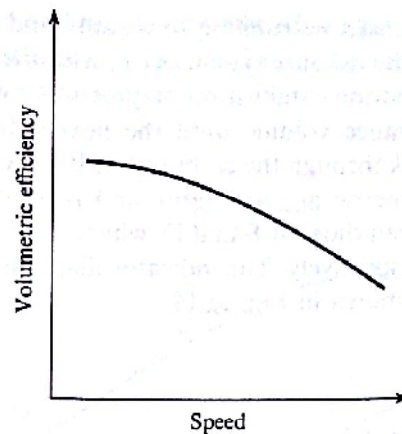


In Fig. 13.13 are shown the engine power characteristics for both ip and bp. As the speed increases from the lower values the two curves are similar, the difference between the ip and the bp at any speed being the fp, which increases with speed. Both curves show maximum values, but they occur at different speeds. The ip falls after the maximum because of a reduction in volumetric efficiency with increased speed. This is influenced by gas temperatures, valve timing, valve mechanism dynamics, and the pressure pulsation patterns in the induction and exhaust manifolds. This fall in volumetric efficiency affects also the bp curve, but this is further decreased by an increase in the fp. This latter effect is predominant since the bp reaches its maximum at a lower speed than the ip.

The variation of volumetric efficiency with speed is indicated in Fig. 13.14, and that of mechanical efficiency with speed in Fig. 13.13. The various methods used for the determination of mechanical efficiency are discussed in section 13.4.



Fig. 13.14 Variation of volumetric efficiency with engine speed

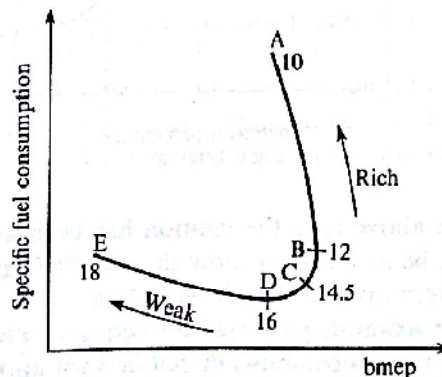


### SI engines

These engines are *quantity governed* by the opening or closing of a throttle valve which regulates the mass flow of charge to the cylinders. Some gas engines are throttled by alteration of the lift of the admission valve, and this can be controlled from the engine governor. The governed speed can be adjusted to select any value in its range.

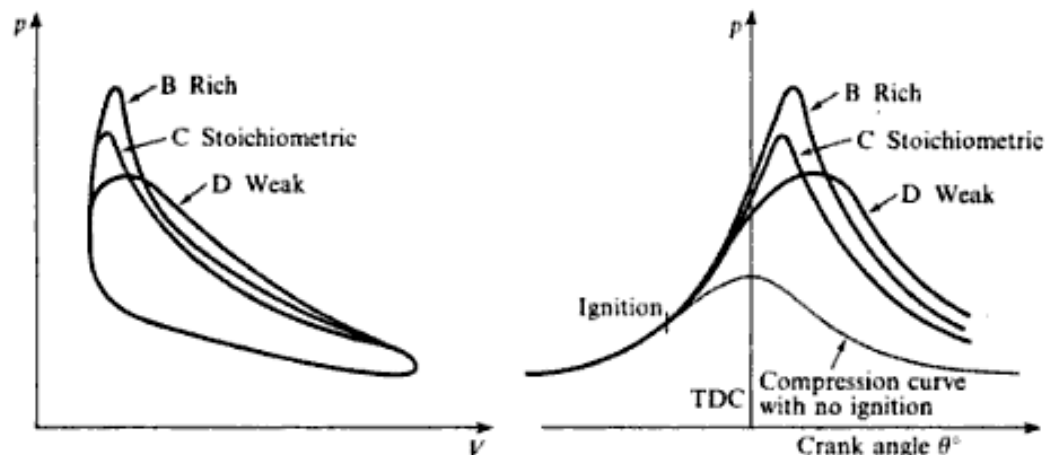
The petrol engine will operate on air-fuel ratios in the range 10/1–22/1, but not necessarily satisfactorily at the extremes. There is some variation between engines. An important test is to run the engine with the air-fuel ratio as the only variable. This is carried out at constant speed, constant throttle opening, and constant ignition setting. The specific fuel consumption is plotted to a base of bmep and a 'hook curve' or consumption 'loop' is obtained. For a single-cylinder engine at full throttle the curve is sharply defined as in Fig. 13.15. The air-fuel ratio is a minimum at A (i.e. the richest mixture). As the air-fuel ratio is increased the bmep increases until a maximum is reached at B (usually for an air-fuel ratio between 10/1 and 13/1). Further increase in the air-fuel ratio produces a decrease in bmep with increasing economy until the position of maximum economy is reached at D. Beyond D, for increasing air-fuel ratios, both bmep and consumption values are adversely affected. Near the point A

Fig. 13.15 Specific fuel consumption against bmep for a spark-ignition engine



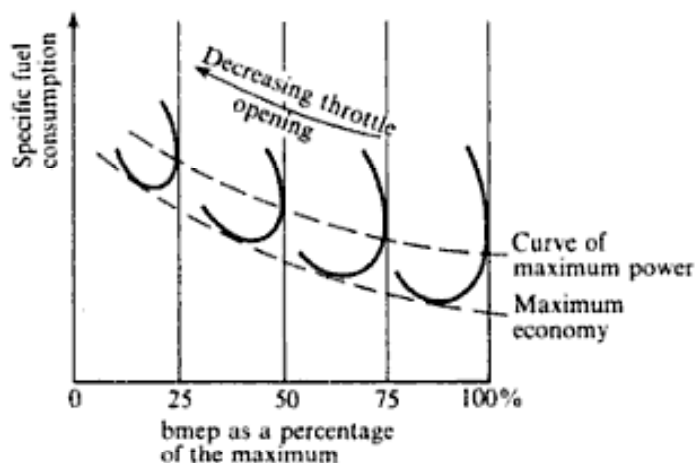
the engine could be running unsteadily and there may be combustion of the mixture in the exhaust system. At E, with the weakest mixture, running will be unsteady and the combustion may be so slow that the gases continue burning in the clearance volume until the next induction stroke begins; this causes *popping back* through the carburettor. Point C is the point of chemically correct or stoichiometric air-fuel ratio, and is about 14.5/1. The mixture strengths range between those at B and D, which are for maximum power and maximum economy respectively. The indicator diagrams corresponding to mixtures B, C, and D are shown in Fig. 13.16.

Fig. 13.16 Pressure-volume and timing diagrams for rich, weak, and stoichiometric mixtures for a SI engine



For multicylinder engines the consumption loops are less distinct, but are generally similar in shape to that for the single-cylinder engine. This is also true for tests made at part throttle openings. A series of readings obtained at different throttle positions at constant speed is shown in Fig. 13.17.

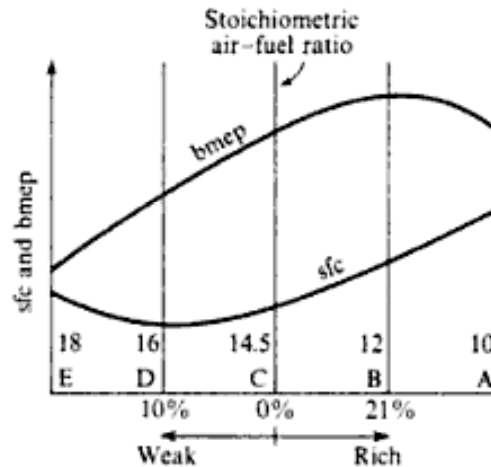
Fig. 13.17 SI engine consumption loops at different throttle settings



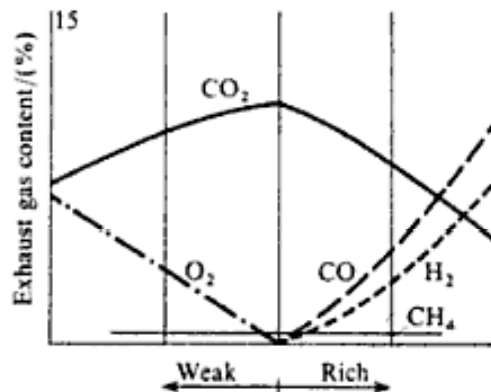
In the above tests the ignition has been assumed to be constant, but other tests can be included to show the effect of ignition timing on the consumption loop. Alternatively the ignition setting can be adjusted at each mixture strength to give maximum power at the speed of the test; by this means the rate of pressure rise on combustion can be kept approximately constant.

Bmep and sfc may be plotted against air-fuel ratio as shown in Fig. 13.18. To the same base of air-fuel ratio the variation of carbon dioxide, oxygen, and carbon monoxide contents of the dry exhaust can be plotted as shown in Fig. 13.19.

**Fig. 13.18** Specific fuel consumption and bmep against air-fuel ratio for a SI engine



**Fig. 13.19** Exhaust gas analysis against air-fuel ratio for a SI engine



Energy balances can be drawn up for the principal points taken from the consumption loop.

Testing the SI engine at part load shows the deficiency of the method of governing by throttling the charge since the efficiency falls with decreasing load. The induction pressure is reduced and the pumping losses increase. The dilution of the fresh charge by exhaust gas increases at lower loads, the clearance volume containing practically the same amount of exhaust gas since the back pressure in the exhaust process remains constant. For this type of mixture a greater amount of fuel is required for combustion to be possible.

### CI engines

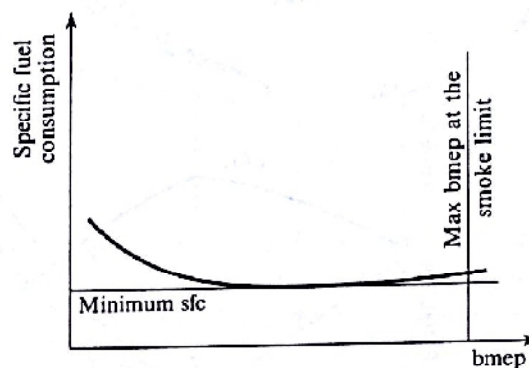
In the main CI engines are not controlled by throttling but by adjusting the amount of fuel supplied to the engine, and hence are *quality governed*. When



adjusting the fuel supplied to a CI engine the limiting condition is given by the *smoke limit*, which is the appearance of black smoke in the exhaust. Engines should not be operated with mixtures rich enough to produce smoke, although such a mixture may give a greater power output. The efficiency under these conditions is low, and the engine soon becomes dirty. The smoke limit occurs at air-fuel ratios of about 16/1. The engine is tested at different speeds to the smoke limit, which can be observed visually or measured by a smoke meter. The values of torque, bp, fuel consumption, and specific fuel consumption are then plotted against engine speed in revolutions per minute.

A consumption loop for a CI engine has the form shown in Fig. 13.20. This shows a minimum sfc and therefore a maximum brake thermal efficiency at part load (i.e. less than the maximum bmep). The curve is reasonably flat over a wide range of values of bmep, which shows the virtue of the CI engine compared with the SI engine for part load operation, a condition often prevailing in road vehicle engines. The reduction in the thermal efficiency at part load is less for the CI engine than for the SI engine.

**Fig. 13.20** Specific fuel consumption against bmep for a CI engine



Mechanical efficiencies and minimum values of sfc are plotted against speed to provide further characteristics.

For a given engine the individual characteristics described in this section are usually sufficient for normal use, but if comparisons are to be made with other engines, perhaps of different types, then the more comprehensive presentation by means of a performance map is an advantage. Figure 13.21 shows the form of such a map for a petrol engine and Fig. 13.22 shows the form for a diesel engine. In the maps bmep is plotted against engine (or piston) speed with curves of constant specific fuel consumption and power per unit piston area. If the engine is to be used in a vehicle the road requirement can be included as a bmep-speed characteristic.

## 13.7 Factors influencing performance

### SI engines

The analysis of the Otto cycle given in section 5.6 shows the dependence of thermal efficiency on compression ratio. A graph of air standard thermal



Fig. 13.21 Performance map for a SI engine

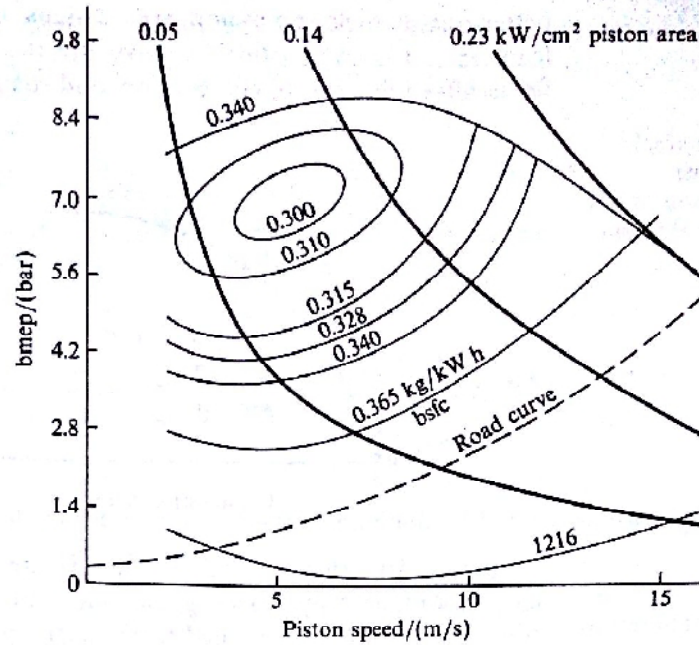
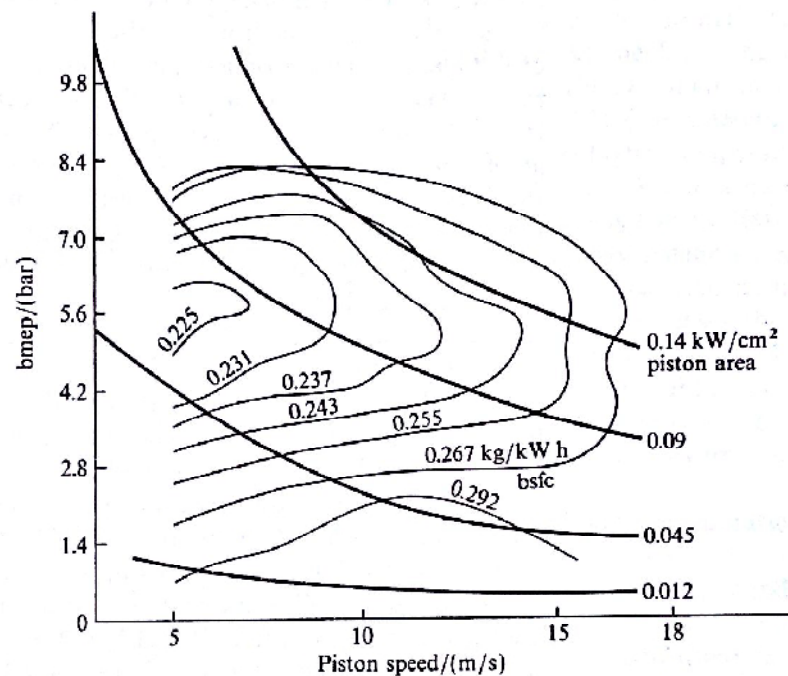


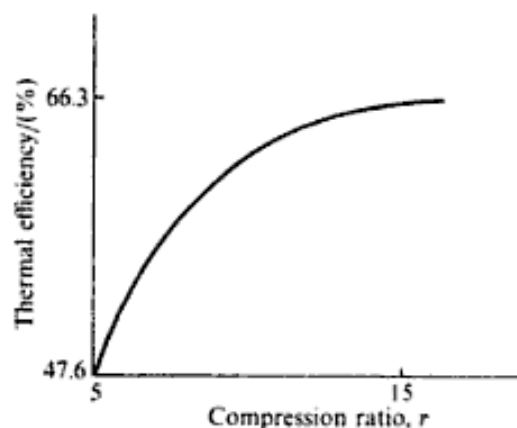
Fig. 13.22 Performance map for a CI engine



efficiency against compression ratio is shown in Fig. 13.23. This graph indicates the form engine development should take, and over the early years increases in compression ratio were made. However, since 1960 the compression ratios have not increased greatly and are in the range 9–10 for production vehicles in the UK. The ability to use higher ratios has depended on the provision of

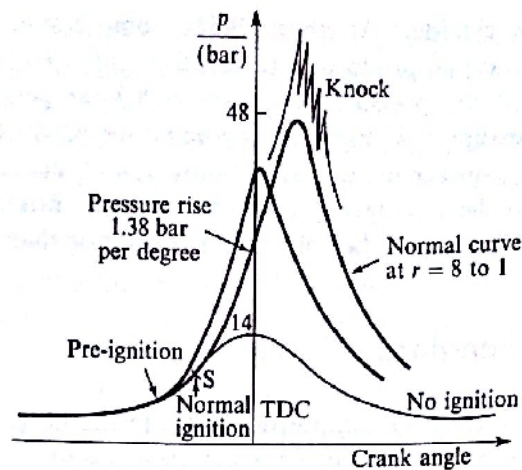
better-quality fuels and of improved designs of combustion chamber. The main features of the combustion chamber are the distances to be travelled by the flame after initiation of combustion, and the gas flow pattern established.

Fig. 13.23 Thermal efficiency against compression ratio for an ideal cycle for a SI engine



It is evident that if a petrol-air mixture is compressed sufficiently it will ignite spontaneously. This suggests one limit to compression ratio if controlled combustion is to be obtained from spark ignition. However, before this limit is reached for the whole charge, spontaneous ignition can occur in the unburnt charge after combustion has commenced normally. The unburnt gas, compressed by the advancing flame front, is raised in temperature and may reach the point of *self-ignition*. This produces an uncontrolled combustion and its occurrence may be heard as a knocking sound. A critical condition can be reached which is called *detonation*, or 'heavy knock'. The advancing flame front is suddenly accelerated by the occurrence of a high-pressure wave and the flame front and shock wave traverse the cylinder together. The *detonation wave* suffers successive reflections, and a high-frequency noise is created. This is an extreme condition which has been produced in test rigs, but such intense conditions are less likely to be produced in a normal engine. These combustion phenomena are usually referred to collectively as 'knock'. One of the results of knock is that local hot spots can be created which remain at a sufficiently high temperature to ignite the next charge before the spark occurs. This is called *pre-ignition*, and can help to promote further knocking. The result is a noisy, overheated, and inefficient engine, and perhaps eventual mechanical failure. The chemical behaviour during this type of combustion is still not fully understood, although a considerable amount of empirical data are available. The pressure/crank angle diagram for normal combustion is shown in Fig. 13.24 with maximum pressure occurring at 10–12° after TDC, and a rate of pressure rise of 1.38 bar per degree of crank angle, with a compression ratio of 8/1. The spark occurs at the point S on the normal compression curve, but there is a *delay period* between the occurrence of the spark and a noticeable departure of the pressure curve from that of normal compression. This is a time delay which is independent of engine speed so that as the engine speed is increased the point S must occur earlier in the cycle to obtain the best position of the peak pressure. This *ignition advance* can be accomplished manually, but can also be controlled automatically by a mechanism in the distributor which is sensitive to engine speed; an additional control is obtained at small throttle openings by a pressure connection from

**Fig. 13.24** Pressure against crank angle for various conditions for a SI engine



the distributor to the induction manifold. Modern petrol engines now have electronic ignition systems.

The compression ratio which can be utilized depends on the fuel to be used and a scale has been developed against which the knock tendency of a fuel can be rated. The rating is given as an *octane number*. The fuel under test is compared with a mixture of iso-octane (high rating) and normal heptane (low rating), by volume. The octane number of the fuel is the percentage of octane in the reference mixture which knocks under the same conditions as the fuel. The number obtained depends on the conditions of the test and the two main methods in use (the *research* and the *motor* methods; see ref. 13.3) give different ratings for the same fuel. The motor test is carried out at the higher temperature and gives the lower rating. The difference between the two is taken as a measure of the temperature sensitivity of the fuel. High-octane fuels (up to 100) can be produced by refining techniques, but it is done more cheaply and more frequently by the use of anti-knock *additives*, such as *tetraethyl lead*. (An addition of  $1.1 \text{ cm}^3$  of tetraethyl lead to 1 litre of 80-octane petrol increases the octane number to 90.) Unfortunately it has been shown that the levels of lead now in the atmosphere due to the SI engine are harmful to health. Public opinion has gradually caused governments to bring in financial incentives for the use of unleaded fuel and car manufacturers have adjusted engines for the use of unleaded fuel (see section 13.13).

Fuels have been developed which have a higher anti-knock rating than iso-octane and this has led to an extension of the octane scale. Aviation conditions of operation lead to another scale which gives a better indication of the detonation characteristics; this is the *performance number* (PN). The relationship between octane number (ON), above 100 and performance number is given by

$$\text{ON above 100} = 100 + \frac{\text{PN} - 100}{3}$$

With higher compression ratio engines other phenomena are observed. From compression ratios of 9.5/1 upwards there are high rates of pressure rise which have their origin in the additional flame fronts started from surface deposits in



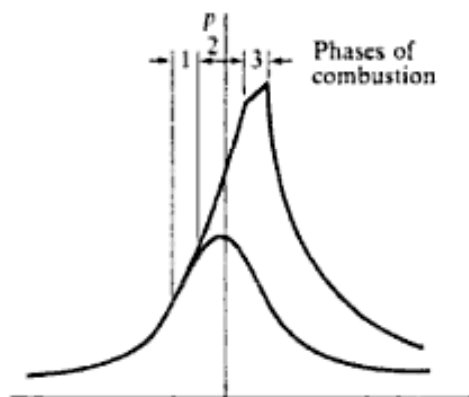
the cylinder. At about 9.5/1 compression ratio the low-frequency engine vibrations produced are called *rumble* or *pounding*. At compression ratios of 12/1 the pressure rise is about 8.3 bar per degree crank angle with a peak pressure of 83 bar. The engine noises produced are known as *thud* or *pressure rap*; surface ignition is not present and fuel characteristics have little influence. This field of development is one which brings many new problems which are more likely to be solved by the chemist than the engineer.

### CI engines

The effect of compression ratio in the CI engine is somewhat simpler than in the SI engine. For combustion to occur at the temperature produced by the compression of the air a compression ratio of 12/1 is required. The efficiency of the cycle increases with higher values of compression ratio and the limit is a mechanical one imposed by the high pressures developed in the cylinder, a factor which adversely affects the power-weight ratio. The normal range of compression ratios is 13/1 to 17/1, but may be anything up to 25/1.

The combustible mixture in the SI engine is formed before compression, but with the CI engine this mixture has to be formed after compression and after injection begins. This leads to delay periods in the CI engine which are greater than those in the SI engine (see Fig. 13.25). The fuel droplets injected have to evaporate and mix with oxygen to give a combustible mixture. The delay period forms the first phase of the combustion process, and is dependent on the nature of the fuel. The second phase consists of the spread of flame from the initial nucleus to the main body of the charge. There is a rapid increase in pressure during this phase and the rate of pressure rise depends to some extent on the availability of oxygen to the fuel spray, which in turn depends on the turbulence in the cylinder. The main factor, however, is that of the delay period. A long delay period means more combustible mixture has had time to form, and so more charge will be involved in the initial combustion. As the speed increases the rate of pressure rise in this phase also increases. This is because the delay period is a function of time if surrounding conditions remain constant, and at the higher engine speeds more mixture will be formed in the delay period. The initial rapid combustion can give rise to rough running and a characteristic

Fig. 13.25 Pressure against crank angle for a CI engine





noise called *diesel knock*. During the third phase of combustion the fuel burns as it is injected into the cylinder, and this phase gives more controlled combustion than that of phase two. One of the main factors in a controlled combustion is the swirl which is induced by the design of the combustion chamber. It has been stated that the delay period depends on the nature of the fuel, and a fuel with a short delay period, or high ignitability, is required. The *ignitability* of a fuel oil is indicated by its *cetane number*, and the procedure for obtaining it is similar to that for obtaining the octane number of petrols. Reference mixtures of cetane ( $C_{16}H_{34}$ ) (high ignitability), and  $\alpha$ -methyl-naphthalene ( $C_{11}H_{10}$ ) (low ignitability), are used. The mixture is made by volume and the ignitability of the test fuel is quoted as the percentage of cetane in the reference mixture which has the same ignitability. For higher-speed engines the cetane number required is about 50, for medium-speed engines about 40, and for slow-speed engines about 30. For details of the standard tests see reference 13.3.

The air-fuel ratios used in CI engines lie between 20/1 and 25/1. As these mixtures are much weaker than the stoichiometric proportion then the imep will be limited, and this also means that for a given fuel consumption the swept volume of the engine will be greater than that of the equivalent SI engine. For further reading on the combustion process in IC engines see refs. 13.4 and 13.9.

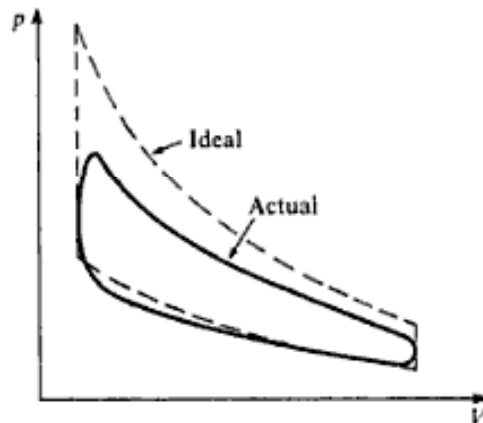
Engines are affected in performance by the atmosphere in which they operate and some allowance must be made in performance figures quoted for variations in pressure, temperature, and relative humidity. The variations in performance can be represented graphically, but the normal values quoted apply up to 30°C and 150 m altitude from sea-level, for normally aspirated engines. The reduction in output per 300 m of altitude above 150 m is about 3%, and for every 5 K above 30°C the reduction is also about 3%. The reduction in output for changes in relative humidity can be up to 6% depending on the conditions.

### 13.8 Real cycles and the air standard cycle

Some of the differences between practical and ideal cycles were discussed as an introduction to this chapter. Other differences are evident from a comparison of the indicator diagram, such as the rounding off of the corners of the diagram due to valve throttling and the fact that the combustion process is not truly at constant volume or at constant pressure. The working fluid is not air throughout; it starts off the combustion process as a mixture of air and fuel vapour, and this mixture changes as combustion proceeds. One of the properties of a perfect gas is that its specific heats remain constant, but in the IC engine there is a considerable change in these values due to the high temperatures reached. At the higher temperatures dissociation takes place (see section 7.7), and the work output is reduced. The maximum pressure and temperature reached in the practical cycle are much less than those obtained by calculation based on the theoretical cycle, assuming the same heat input. Heat losses from a heat engine must be avoided, but with a real engine some form of cooling is essential so that temperatures will not be reached such that there will be failure of the engine materials. The compression and expansion processes are thus not

adiabatic but are of the form  $pv^n = \text{constant}$ ; the assumption of internal reversibility is nevertheless a good approximation. A comparison of the real cycle and the ideal cycle is shown in Fig. 13.26 for a petrol engine; the pumping loop of the practical cycle has been omitted.

Fig. 13.26 Comparison of real and ideal cycles for a SI engine



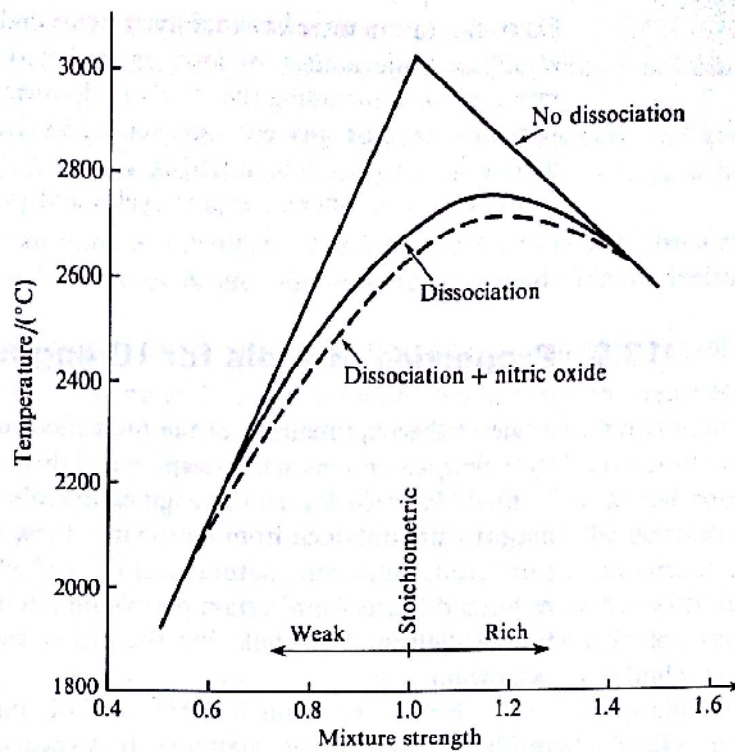
A most important criterion is the temperature (and hence pressure) which can be attained by the 'constant volume' combustion phase. Figure 13.27 shows the temperatures reached on the combustion of mixtures of fuel and air of different strengths between 50% weak and 50% rich, for a compression ratio of 5 to 1. The maximum temperature should theoretically be produced by a stoichiometric mixture strength, but in fact it occurs at about 20% rich. The shape of the temperature curve is due to the dissociation of  $\text{CO}_2$  and  $\text{H}_2\text{O}$  into  $\text{CO}$ ,  $\text{H}_2$ , and  $\text{O}_2$  which is slight at a temperature of  $2000^\circ\text{C}$ , but increases rapidly above that. In engine combustion the dissociation of  $\text{CO}_2$  affects engine behaviour more than that of  $\text{H}_2\text{O}$  which is relatively slight. Combustion theory, as described in Chapter 7, allows the dissociated proportions of  $\text{CO}$ ,  $\text{H}_2$ , and  $\text{O}_2$  to be calculated and represented in the combustion equations. Hence the energy released on combustion can be calculated. If the formation of nitric oxide which occurs at the higher temperatures is taken into account the temperatures attained are even lower, due to the absorption of energy to form  $\text{NO}$  as shown in Fig. 13.27. At the higher compression ratios of modern engines higher temperatures and pressures are attained and dissociation occurs, not only to a greater extent for  $\text{NO}$ , but also to some common radicals mainly  $\text{OH}$ ,  $\text{H}$ , and  $\text{O}$ , e.g. at  $3000^\circ\text{C}$  a stoichiometric mixture of octane and air at equilibrium will contain  $\text{OH}$  (1.4%),  $\text{H}$  (0.3%),  $\text{O}$  (0.3%), and  $\text{NO}$  (0.3%).

Mixture strength, particularly for a spark-ignited engine, can vary considerably due to the way in which the mixture is created and distributed. It is not a controlled process as it is in the diesel engine. One of the results of this is a considerable variation in the cycle to cycle and cylinder to cylinder performance of petrol engines.

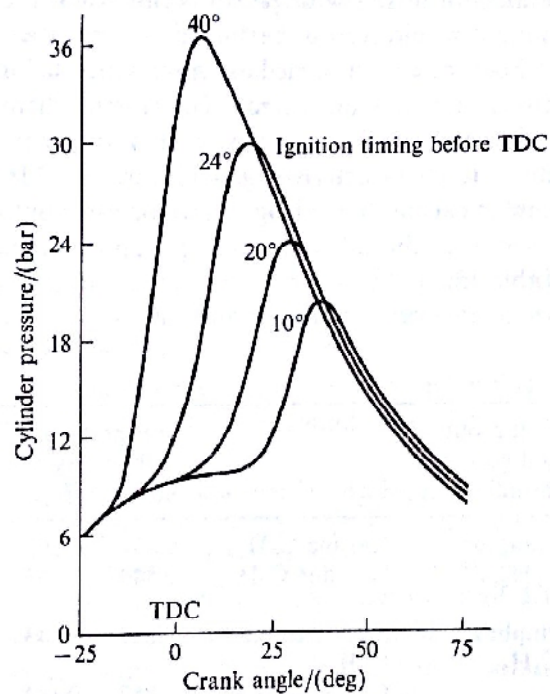
An important parameter which is open to selection is that of the spark timing as described in sections 13.6 and 13.7. Figure 13.28 shows the form of the variation of the maximum pressure reached in the cylinder with the advance of the ignition before the TDC position.



**Fig. 13.27** Combustion temperature against mixture strength for a SI engine with and without dissociation



**Fig. 13.28** Effect of ignition advance on the pressure–crank angle diagram



The factors dealt with above indicate the complexity of the processes of the IC engine that give the engine a character which has fascinated many distinguished engineers over lifetimes of service, and continues to do so. Calculation becomes more difficult as the information required is increased more and more, the relationship between data is complex, and the time taken

for computation increases rapidly. To this end charts have been prepared from established information to describe the behaviour of different fuels at different mixture ratios including the effects of dissociation and different proportions of recycling exhaust gas etc. (see ref. 13.5). In serious work this information will be included as data stored in a computer ready for use in the analysis or computation of different engine cycles and running conditions.

### 13.9 Properties of fuels for IC engines

Some of the requirements of the fuels used in IC engines have been indicated by the earlier sections of this chapter, and these will be added to in this section.

Fuels suitable for petrol engines are obtained from several sources. The majority are obtained from petroleum; these consist of straight petrol distilled from crude petroleum, natural gasoline (which is a light spirit), and cracked or re-formed spirits from certain petroleum fractions. Petrol can be obtained from the distillation of oil shale, but the product is more expensive than that from petroleum.

Alcohol in the main is ethyl alcohol obtained from the fermentation of residues from vegetable matter such as sugar-cane and sugar-beet. The amount of alcohol mixed with petrol is not usually greater than 20%, since a higher content would lead to carburation difficulties.

Fuels can be classified as those which belong to a chemical series and those which are manufactured. The main chemical groups are the paraffins, naphthenes, and aromatics, each group consisting of carbon and hydrogen atoms to a characteristic general formula. Members from two groups can have similar carbon to hydrogen proportions, but different chemical characteristics because of the different atom grouping in the molecule. This is illustrated in Table 13.3.

**Table 13.3** Chemical fuels

| Fuel group and general formula | Fuel                         | Proportions by mass |       | Relative density | Air/fuel ratio | NCV<br>(kJ/kg) |
|--------------------------------|------------------------------|---------------------|-------|------------------|----------------|----------------|
|                                |                              | C                   | H     |                  |                |                |
| Paraffins<br>$C_nH_{2n+2}$     | Hexane $C_6H_{14}$           | 0.837               | 0.163 | 0.67             | 15.3           | 45 124         |
|                                | Heptane $C_7H_{16}$          | 0.840               | 0.160 | 0.69             | 15.2           | 44 660         |
| Naphthenes<br>$C_nH_{2n}$      | Cyclo-hexane<br>$C_6H_{12}$  | 0.857               | 0.143 | 0.79             | 14.7           | 43 030         |
|                                | Cyclo-heptane<br>$C_7H_{14}$ | 0.857               | 0.143 | 0.78             | 14.7           | 43 960         |
|                                | Cyclo-octane<br>$C_8H_{16}$  | 0.857               | 0.143 | 0.74             | 14.7           | 43 960         |
|                                |                              |                     |       |                  |                |                |
| Aromatics<br>$C_nH_{2n-6}$     | Benzene $C_6H_6$             | 0.923               | 0.077 | 0.88             | 13.2           | 40 700         |
|                                | Toluene $C_7H_8$             | 0.913               | 0.087 | 0.87             | 13.4           | 41 054         |
|                                | Xylene $C_8H_{10}$           | 0.905               | 0.095 | 0.95             | 13.6           | 41 868         |



The common manufactured fuels are as follows:

**Petrol:** a refined distillate of petroleum; the boiling-point is not usually greater than  $200^{\circ}\text{C}$ .

**Kerosene:** a refined distillate of petroleum which boils between  $150$  and  $200^{\circ}\text{C}$ .

**Benzole:** a refined distillate of coal tar which consists mainly of benzene and toluene.

**Gas oil:** an unrefined distillate after kerosene fractions have been removed.

**Diesel fuel:** mixtures of gas oils from various crude oils or blends with fuel oils.

**Blends of alcohol:** petrol with about 15% alcohol and 15% benzole.

Kerosene is commonly called 'paraffin', but as this is the name of a chemical series as mentioned earlier, and as the commercial paraffin contains members of other series, the name kerosene is to be preferred. Petrol contains members from several different series and special additives. The actual content of the straight petrols depends on the geographical origin of the petroleum.

The specific enthalpy of vaporization of fuels is an important quantity in both CI and SI engines. With SI engines this factor together with the volatility of the fuel determines the density of the charge in the cylinder; the density is inversely proportional to the absolute temperature in the cylinder at the closing of the inlet valve. This temperature depends on the specific enthalpy of vaporization of the fuel and the amount of heating which takes place during induction. Most liquid fuels have similar specific enthalpies of vaporization, the exception being alcohol which has a high specific enthalpy of vaporization, a higher density of charge, and a marked increase in power compared with other fuels. In CI engines the specific enthalpy of vaporization has no bearing on volumetric efficiency due to the fact that the fuel is injected after compression; this is detrimental when the calorific value and delay period are considered since the energy necessary for evaporation of the fuel must come from the compressed air.

Fuel volatility is of importance in SI engines from the point of view of ease of starting and the heating required to provide an even distribution of mixture to different cylinders.

In CI engines the volatility of the fuel influences the time taken for a combustible envelope to form round the fuel droplets, and hence influences the delay period.

The calorific value of a fuel is important in determining the amount of fuel required, but does not indicate how much power can be obtained from it. Engines using gaseous fuels are affected more by variation in calorific value than petrol and diesel engines. The power output depends on the heating value of the mixture of fuel and air and the stoichiometric mixtures of all hydrocarbon fuels give the same heating value per unit volume at the same reference, temperature, and pressure, within close limits.

Tetraethyl lead has been mentioned as an anti-knock additive in petrols. Other additives are oxidation inhibitors, corrosion inhibitors, combustion control compounds, and anti-icing additives to counteract freezing in the carburettor.

The properties of fuels are determined by standard tests, the procedure for which may be obtained from ref. 13.3.

### 13.10 Fuel systems

The purpose of an engine fuel system is to provide the cylinder with a mixture of air and fuel in the correct proportions for the engine requirements at any particular instant. There are basically two methods available, one is called *carburation* and is used for petrol engines and the other is a type of *fuel injection* which is a characteristic method for diesel engines. There are many different designs of each and only the basic principles will be dealt with in this book.

The petrol engine for automotive purposes has been developed on the basis of the carburettor although petrol injection is becoming more common, the latter having been used for a long time for aircraft and special engines, such as military vehicles and racing cars. The carburettor is a simple, cheap device which has served its purpose for many years, but the trend to higher powered, multicylinder engines has shown the single carburettor system to be inadequate. As a consequence multichoke carburettors and twin or triple carburettor layouts have been used to meet increasingly sophisticated engine requirements, but several designs of petrol injection systems have also been introduced.

The fundamentally different methods of charge ignition by spark and by compression in the petrol and diesel engines respectively have dictated different means of fuel supply for the two engines. However, the features of the oil injection system from the point of view of control, accurate fuel metering, and good fuel consumption characteristics in comparison with the carbureted petrol engine with its poor fuel consumption, particularly at part load, have created the belief that ultimately petrol injection would be the preferred method if a sufficiently simple system could be produced at a cost which would make it competitive with the carburettor. In recent years the added requirement for engines to meet exhaust gas emission regulations (see section 13.13) has increased the demand for accurate fuel metering and engine control. These factors have increased the interest in petrol injection although the earlier regulations were more satisfactorily met by carbureted engines than by those with the first petrol injection systems. In general the fundamental problem is to measure, or compute accurately, the mass flow rate of air into the engine at any instant and to mix the correct amount of petrol into it in such a way that the air and fuel mixture produced is right for the engine running condition. If the airflow were steady in an engine and at constant temperature and pressure the problem would be relatively straightforward, but airflow rates, pressures and temperatures change; also the engine is in dynamic operation with phases of acceleration, deceleration, and overrun (when the throttle is closed and the vehicle wheels are turning the engine). Thus a fuel supply system, to be successful, must be able to cope with a wide range of running conditions and demands.

#### Carburation

The term 'carburation' covers the whole process of supplying continuously to a petrol engine a mixture of vaporized fuel and air which is suitable to each engine condition of load, speed, and temperature. In section 13.5 it is seen that the air-fuel ratio varies between maximum economy and maximum power

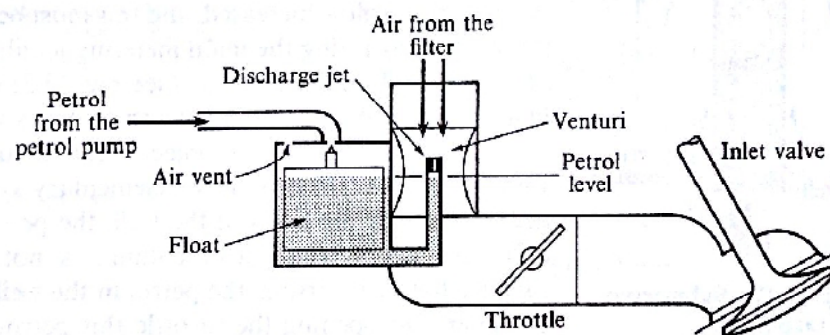


conditions, and the stoichiometric air–fuel ratio is not adequate to all demands. The function of the carburettor is to measure out the correct proportions of liquid fuel and air for the particular engine condition. The liquid fuel must be ‘atomized’ at the carburettor (i.e. broken up into a fine spray to assist in the evaporation of the fuel, so that the mixture entering the cylinders is homogeneous). The necessary energy for the evaporation of the fuel must be supplied to the mixture and this is not the function of the carburettor but of the whole induction tract. The metering process is carried out at the carburettor, but the actual mixture ratio, its condition and distribution between cylinders, depends also on the design of the complete induction system and the temperatures therein.

The perfect carburettor would supply the air–fuel ratio required at all speeds and throttle openings no matter what the climatic conditions or the rate at which the demand was changing. A consideration of all the factors which influence the final mixture burned in the cylinder (i.e. engine condition required, mechanical characteristics of the engine, the physical differences between the constituents of the mixture, the rapid fluctuation in demand, and the temperature and humidity variation) shows the difficulty in obtaining the ideal carburettor. The carburettor is a highly developed component which is produced in a number of different designs each having its own refinements, with the object of supplying to the engine the air–fuel mixture it needs. The elementary metering process will be considered, since this of fundamental importance to all types.

In Fig. 13.29 a simplified carburation system is shown. The petrol pump, either electrically driven or mechanical driven by the crankshaft, pumps petrol from the tank to the float chamber of the carburettor. The function of the float chamber is to maintain a constant level of petrol in the chamber by shutting off the supply from the pump when this level is about to be exceeded. The float chamber is vented to atmosphere through a small hole in the cover, hence the pressure on the surface of the petrol is constant and equal to that of the atmosphere. The air is induced by the depression created by the piston moving downwards in the engine cylinder, and, after passing through a filter, enters the carburettor at about atmospheric pressure. The petrol engine is quantity governed, which means that when less power is required at a particular speed the amount of charge delivered to the cylinders is reduced. This is achieved by means of a throttle valve of the butterfly type which is situated in the air inlet.

**Fig. 13.29** Simplified carburation system

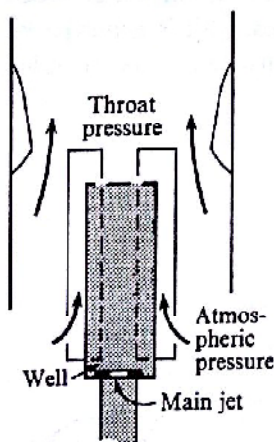
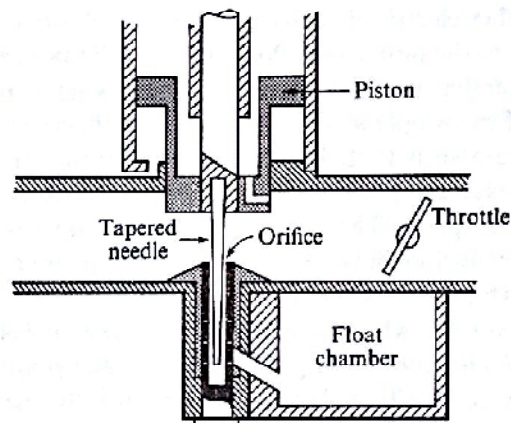




The air on induction enters the venturi or choke tube. This is a tube of decreasing cross-section which reaches a minimum at the throat or choke of the venturi, which is shaped to give the minimum resistance to the airflow. The petrol discharge jet is situated at the throat and is subject to the air pressure there. The pressure at the throat is below atmospheric since the air velocity has been increased from that at inlet to the carburettor to a maximum at the throat. Thus the two petrol surfaces, that in the float chamber and that at the discharge jet, are subject to different pressures. This pressure difference acting on the petrol column causes the petrol to flow into the airstream, and the rate of flow is controlled or metered by the size of the smallest section in the petrol passage. This is provided by the main jet and the size of this jet is chosen to give the required engine performance, and is an empirical selection. The pressure at the throat at the fully open throttle condition lies usually between 38 and 50 mm Hg below atmospheric, and seldom exceeds 76 mm Hg below atmospheric.

In the carburettor described the choke or throat has a constant area and the pressure changes with throttle opening and engine speed. It is referred to as the *fixed choke* type. Another type, of which the SU carburettor is the main example, is the *constant vacuum* type. In this type the pressure at the throat, and thus the air velocity, remains constant, but the area of the throat is varied. Similarly the area of the petrol orifice or jet must also vary, and this is achieved by means of a tapered needle, attached to a piston. The needle moves in the orifice thus forming a discharge annulus for the petrol (see Fig. 13.30).

**Fig. 13.30** Constant vacuum carburettor



**Fig. 13.31** Submerged jet carburettor

With the elementary form of carburettor described, the mixture would become richer as the airflow increased, and this must be 'corrected'. In some carburettors this is done by fitting the main metering jet about 2.5 cm below the petrol level, and it is called a submerged jet (see Fig. 13.31). The jet is situated at the bottom of a well, the sides of which have small holes which are in communication with the atmosphere. The second object of this arrangement is to achieve atomization which is not obtained with the elementary system described previously. Air is drawn through the holes in the well, the petrol is emulsified, and the pressure difference across the petrol column is not as great as in the elementary carburettor. On starting, the petrol in the well is at the level of that in the float chamber. On opening the throttle this petrol, being subject to the low throat

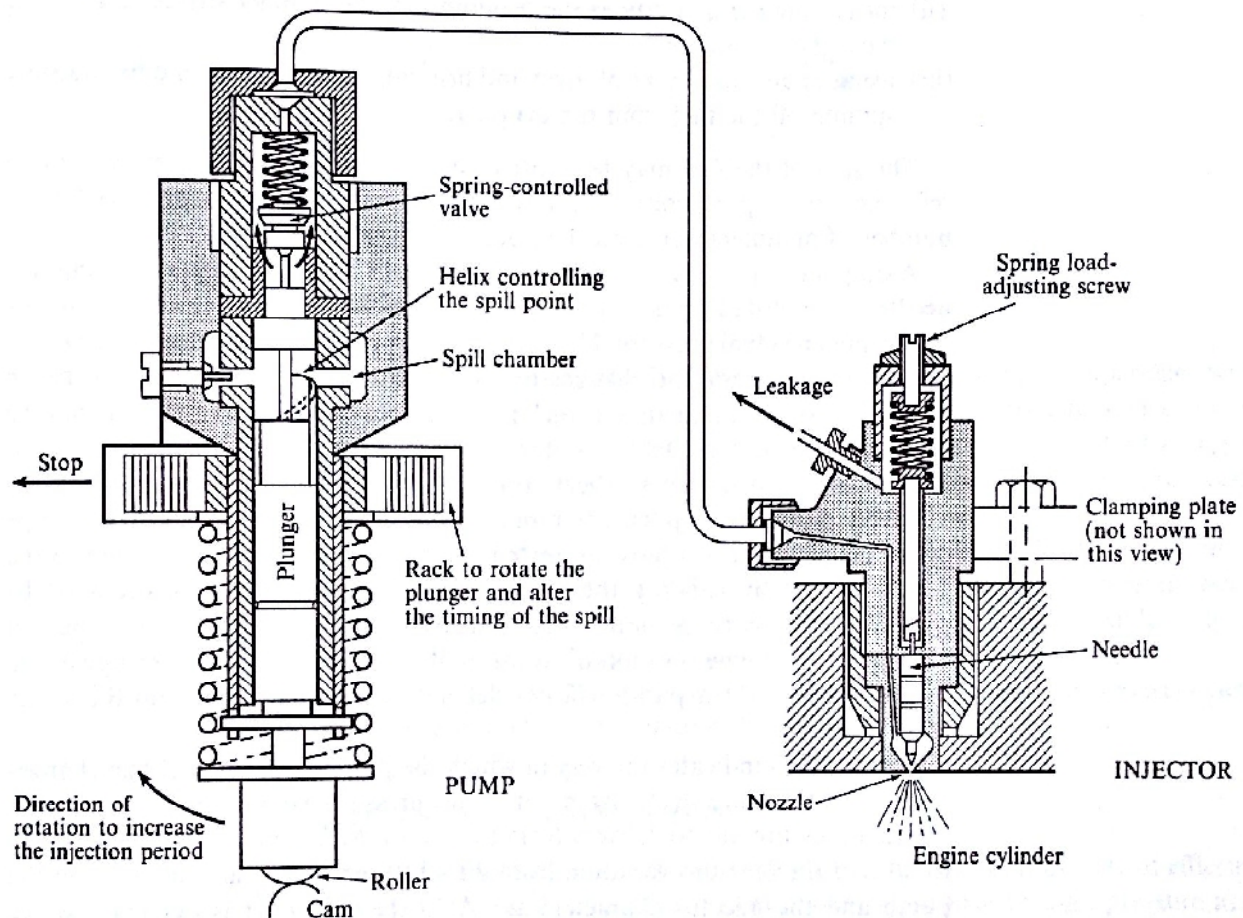


pressure, is drawn into the air. This continues with decreasing mixture richness as the holes in the central tube are progressively uncovered. Normal flow then takes place from the main jet.

A petrol engine operates in service mainly at part throttle and a carburettor which gives a correct full throttle delivery would not be able to meet part throttle requirements. As the throttle is closed the point of lowest pressure in the induction system moves from the venturi, at full throttle, to the engine side of the throttle when the engine is idling with the throttle practically closed. The position of minimum pressure changes between these limits. At the idling condition the pressure in the induction trace is 400–460 mm Hg below atmospheric. The main jet will not supply petrol when the engine is idling because there is no longer sufficient depression at the throat. Another petrol supply is provided which delivers to the engine side of the throttle. This is indicated diagrammatically in Fig. 13.32.

A rich mixture is required for starting and idling, but as the throttle is opened the demand is for a weaker mixture. There is a merging of the idling and main jet deliveries, but in order to avoid flat spots and be able to deliver the required mixture throughout the range of throttle positions, it is usual to provide a third 'progression' system. On acceleration the throttle is opened suddenly and it is necessary to provide a rich mixture. If the part throttle characteristics of the

Fig. 13.32 Injector pump and injector for a CI engine



carburettor are satisfactory the fuel for acceleration is best provided, in the interests of economy, by a separate accelerating device. The means of doing this vary and the types may be classified as using an *accelerating pump*, or an *accelerating well*.

### Fuel injection: diesel engines

The function of a fuel injection system is to meter the fuel accurately and uniformly to the engine cylinders under all operating conditions from idling to full load. The timing of the injection should be accurate enough to give the required combustion characteristics. The fuel from the tank is filtered before passing to the pump, and the metered fuel is then passed to the injection which is fitted in the engine cylinder. The 'jerk pump' system is the one which is used almost universally over the whole range of oil engines, and will be the one described here. The jerk pump is a piece of precision equipment, and consists of a barrel with plunger; the close fit required to prevent leakage at the high pressures reached is obtained by lapping. The plunger is driven from the camshaft and the control of the amount of fuel to be delivered can be made by the following:

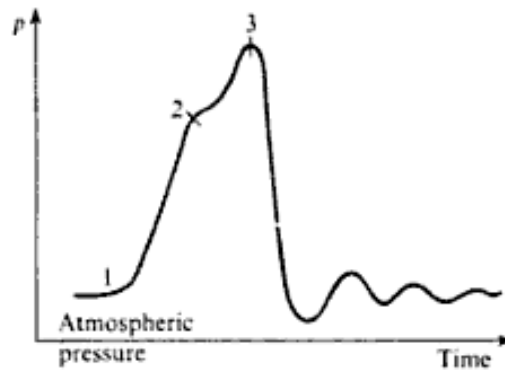
- (i) using a plunger with a variable stroke; or
- (ii) measuring the quantity at the beginning of the plunger stroke and spilling back the excess; or
- (iii) using a constant stroke plunger and bringing delivery to an end by suddenly spilling off the fuel from the cylinder.

The spill of the fuel may be controlled by a cam-controlled spill-valve (see ref. 13.6), or a port control. Pumps with port control are produced by a number of manufacturers and their action will be described.

A simplified injection system is shown in Fig. 13.32. The spring-loaded injector needle is set to lift at a predetermined pressure in the delivery line which contains oil at a high residual pressure. There are complex pressure and velocity variations set up in the system, the changes in the pressure being propagated in the oil at the speed of sound in the oil. The effect of such variations will not be considered here (see ref. 13.6 for a full treatment of the subject). The plunger is provided with a helical groove, the upper edge of which controls the uncovering of the spill port. The timing of the spill is thus decided by the shape of the helix, and this is most important. The part of the helix presented to the port is varied by rotating the plunger in the barrel, and a means must be provided for this to be done automatically or manually while the engine is running. An extreme position of rotation of the plunger in the barrel gives the position at which the pump will not deliver fuel to the engine, and this is the 'stop' position.

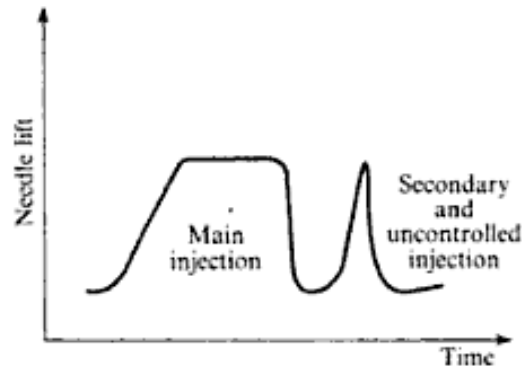
Figure 13.33 indicates the way in which the pressure in the fuel line changes during the injection cycle. At point 1 the pressure begins to rise above the residual pressure due to delivery from the pump. At 2 the injector needle begins to lift and the pressure variation from 2 to 3 depends on the delivery from the pump and the injector characteristics. At 3 the spill port is opened and the

**Fig. 13.33** Pressure variation in the fuel line during injection



pressure falls with a characteristic depending on the spill port action and that of the closing of the injector needle. The pressure variation after this is due to the reflection of pressure waves in the line which are damped out unless the next cycle commences before the damping is complete. The pressure in the line after spill may reach such a value that the needle opens for a second time in the cycle. This is undesirable and is known as *secondary injection* (see Fig. 13.34).

**Fig. 13.34** Needle lift against time showing undesirable secondary injection



### Fuel injection: petrol engines

At first sight it may appear that the application of diesel engine experience with injection systems to the petrol engine would produce a satisfactory mechanical petrol injection system. However, the different requirements of the two types of engine and the increased knowledge and use of electrical and electronically operated valves and transducers have made it evident that a petrol injection system would be wholly or mainly an electronic system. The development of electronic devices continues with increased component reliability, greater design sophistication, more miniaturization, and reduced cost giving flexible systems which are simple to locate, robust in operation, and easy to replace.

There are certain basic parameters to be decided upon for each system such as those given under the headings below.

#### Manifold or in-cylinder injection

Direct injection into the cylinder is attractive from the point of view of efficient fuel distribution, but the injector used would be subject to high pressure and



temperature conditions which it would not experience in the manifold. Injection into the manifold allows more time for the fuel and air to mix, giving better combustion, uses simpler injectors and requires easier access to the engine, particularly if it were not originally designed for injection.

### Continuous or pulsed injection

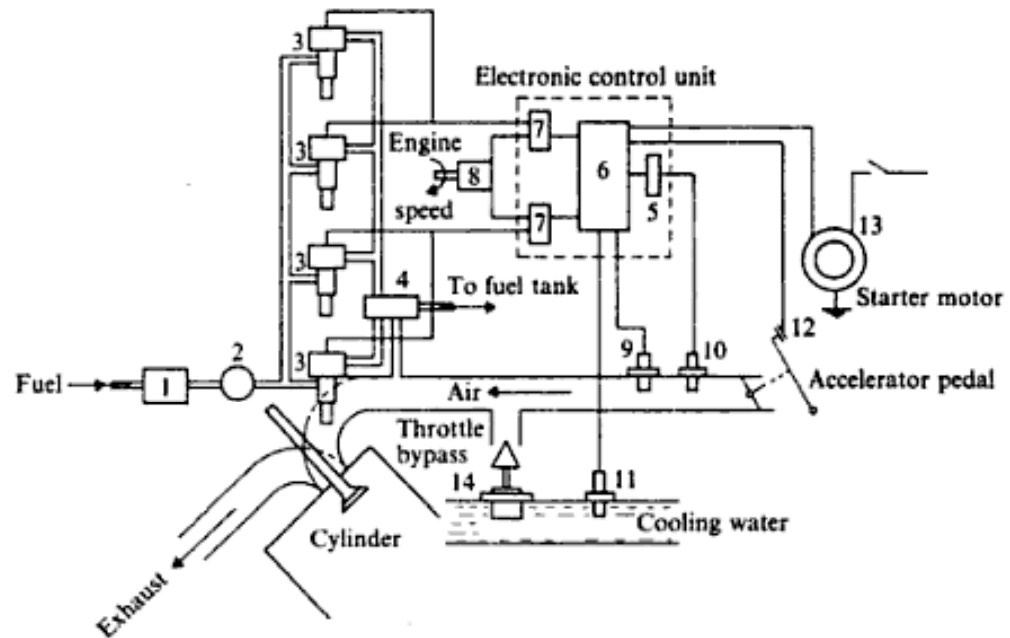
The injector can be used to spray fuel continuously into the manifold and this requires a fuel flow rate varying by 50 to 1 between idle and maximum speed running; accurate mixture preparation requires good control on airflow measuring and fuel metering. With pulsed injection the fuel is injected near each cylinder inlet valve in each cycle by an injector which is solenoid operated for a measured period of time. The range of pulse durations required is about 5 to 1 and 'time' is a simple quantity, conveniently and accurately measured electronically; the demands of several engine parameters can be computed and expressed as a single controlling value. The very small pulse durations required at idling speed, of the order of 1 ms, set the criterion of quality of design for the injector.

It would appear that the ability to time the injection pulse anywhere in the induction stroke would be a powerful facility, but experience shows that the effect of such timing on power output and hydrocarbon emissions is not great. This allows some relaxation on the injector design as it enables injectors to be grouped to receive the same pulse; over a longer duration, and so injection occurs into different cylinders at different points in the cycle or by 'non-time injection' procedure.

The layout of a typical injection system is illustrated in Fig. 13.35 and includes provision for the various engine parameters which it is believed it is necessary to take into account. Alternative, and possibly simpler, designs could employ fewer parameters and so eliminate the need for some of the transducers, etc. described in this illustration. For instance, in this case the steady airflow requirement is computed from measurements of manifold pressure, inlet air temperature, and engine speed; a single reading of air mass flow rate would simplify considerably the requirements of the system. The system shown takes into account in addition the parameters of cooling water temperature, starter motor action, a measurement of change of manifold pressure by electronic differentiation of the pressure signal, and a throttle-actuated switch which cuts off the fuel supply when the throttle is closed. Information from each of the transducers or switches involved is 'computed' electronically to give a single pulse width, or a main and a second pulse, and is communicated to the pulse generators.

Fuel is supplied to the injectors at a pressure of 1.72 bar above the manifold pressure and fuel not injected is returned through a pressure control valve to the supply tank. This circulation of fuel cools the injectors and purges them of air and vapour. The quantity of fuel injected depends on the pulse duration as decided by the electronic control unit, the orifice area and the pressure difference across it and the last two of these are constant, giving only the pulse duration as a variable. It would appear to be essential that the nozzles be matched very closely with respect to area and coefficient of discharge as small variations in

**Fig. 13.35** Typical direct fuel injection system for a SI engine. (1) fuel filter; (2) fuel pump (electrically driven); (3) injectors; (4) fuel pressure control valve; (5) pressure transducer; (6) computer; (7) pulse generator; (8) trigger unit (engine driven). Transducers: (9) manifold air temperature; (10) manifold air temperature; (11) cooling water temperature; (12) throttle switch; (13) starter motor switch; (14) idling air control valve (throttle bypass)



the fuel supply could affect the exhaust emissions of the engine quite considerably. A reduction in air-fuel ratio promotes the carbon monoxide content and an increase usually adversely affects the hydrocarbon content depending on the air-fuel ratio required, with a value of 16–16.5 to 1 being good average optimum values for both types of emission.

Information from the manifold temperature and pressure transducers and the engine speed is computed to give the basic pulse duration. If the engine is starting from cold a signal from the starter motor produces fuel enrichment, and a temperature signal from the cooling water controls the additional fuel during warm-up. Additional fuel for acceleration is provided by a control signal arising from the rate of change of pressure in the manifold as computed in the control unit. When the vehicle is in the overrun condition the throttle closes and switches off the fuel, thus eliminating the high output of hydrocarbons in the exhaust gas obtained with a carbureted engine during this phase of operation. A throttle bypass valve allows an increase in the air supply during idling when the throttle is in the closed position and the opening of this is controlled by cooling water temperature.

### Charge stratification

One of the developments that can utilize petrol injection is known as *charge stratification*. The objective of this is to initiate combustion in a region where the mixture is rich and to produce a flame that can then travel easily through the rest of the charge which is, on the whole, weak. One such system employs two injectors, one supplying the main combustion chamber and the other a small directly connected auxiliary chamber in which the spark plug is fitted. In the main chamber the air-fuel ratio is 2.18 to 1, in the auxiliary it is 0.66 to 1 and overall it is 1.64 to 1 relative to the stoichiometric ratio.

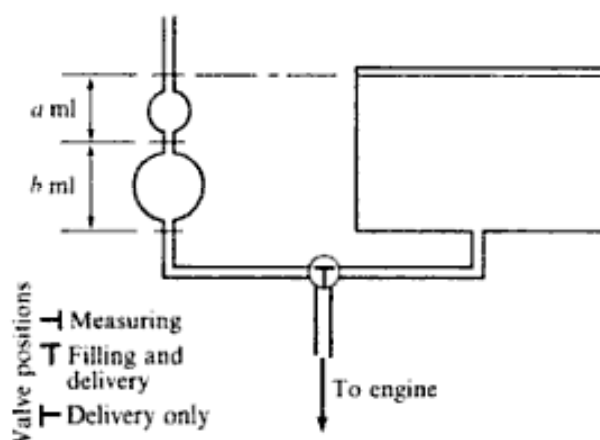
If the contents of this section are considered in relation to that of section 13.13 on engine emissions a reasonable picture of the petrol engine and its stage of development should be obtained.

### 13.11 Measurement of air and fuel flow rates

The practical determination of the air-fuel ratio consists of measuring the rate at which air and fuel are consumed by the engine.

Several means are available for the measurement of fuel consumption and one simple arrangement is shown in Fig. 13.36. The measuring vessel consists of two reservoirs of known capacity in series, the capacities being measured between marks on the connecting capillary tubes. The fuel level thus falls quickly past the marks on the small bore capillary tube. Fuel from the tank or measuring vessel passes through a three-way valve to the engine. The three positions of the valve are shown in Fig. 13.36. Provision must be made to allow the fuel to fill the measuring vessel up to the level in the tank. The time is taken for the consumption of a known volume of fuel, and thus the rate of fuel consumption can be determined. This arrangement has a number of possible variations which are suitable for particular purposes.

**Fig. 13.36** Simple test measurement of fuel consumption

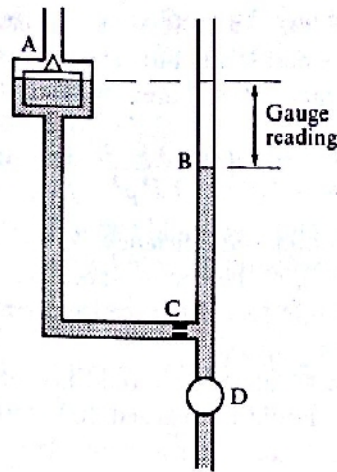


There are types of instruments available called flow meters, which give the rate of flow directly and the calibration can be made in litres per hour or kilograms per hour as required. The Amal flow meter is used widely in industry and is illustrated diagrammatically in Fig. 13.37. A float-controlled constant-level tank A is connected to a vertical glass tube B. The connecting tube carries an orifice at C, and a tap D is fitted between the instrument and the engine. When no fuel is being delivered to the engine the fuel is at the same level in A and B. As fuel is supplied to the engine the level in B falls until the fuel taken by the engine is equal to that passing through the orifice, and the level in B reaches an equilibrium position. The gauge reading is proportional to the fuel consumption and the calibration is an empirical one. Each instrument has two tubes such as B, each with its own range of flow.

Such instruments as those described which depend on volumetric measurement are satisfactory for carburettor engines, but may not be satisfactory for CI



Fig. 13.37 Amal flow meter for engine testing



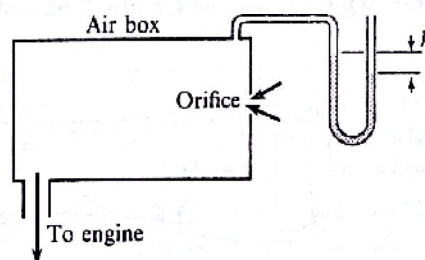
engines with fuel pumps. There may be variation in the volume of fuel between the inlet port of the pump and the measuring instrument, due to entrapped air. If the head of fuel under which the engine operates varies during the fuel measurement (as in the type in Fig. 13.36), the entrapped air in the system expands thus causing an error in the volume of fuel measured.

An instrument which overcomes this difficulty and gives an accurate measurement of the weight of fuel consumed is the BICERA flow meter. In this instrument the fuel during the measuring operation is drawn from a vessel which moves upwards as its weight decreases, thus keeping the fuel level constant. The vessel is attached to an arm pivoted on a block, and a counterweight on a rod also pivoted on the block provides the necessary balancing torque. The weight of fuel consumed is measured by observing the upward movement of the vessel; this is done by an optical system which magnifies the movement of the vessel approximately 30 times.

The satisfactory measurement of air consumption is a more difficult task. It is essentially the measurement of the rate of flow of a compressible fluid, complicated by the fact that the flow is pulsating due to the cyclic nature of the engine. This prevents the use of an orifice in the induction pipe since a steady and reliable reading would not be obtained. The usual method of damping out the pulsations is to fit an air box of suitable volume to the engine with an orifice in the side of the box remote from the engine. The pressure difference across the orifice is steady if the system is correctly designed, and is measured by means of a suitable manometer (see Fig. 13.38).

The criteria for the selection of the size of air-box have been established and

Fig. 13.38 Air box for air flow rate measurement



are described in ref. 13.7. It is a conclusion of this reference that the dimensions of a suitable air-box meter can be determined by the use of a non-dimensional factor  $U$ , defined by

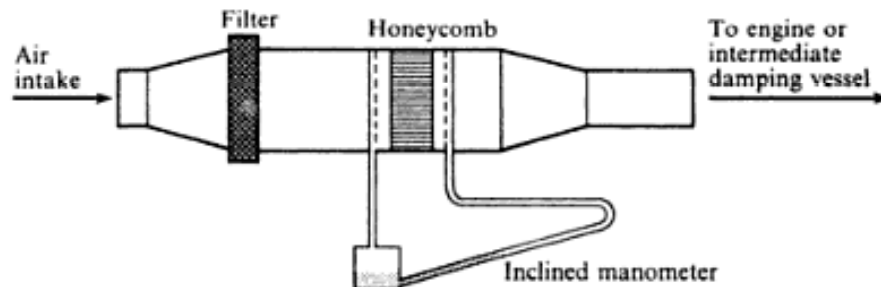
$$U = \frac{1}{40.94 \times 10^5} \left( \frac{\eta_v C V N^2 n^2}{T d^4 p^2} \right)$$

where  $\eta_v$  is the volumetric efficiency,  $V$  the engine swept volume ( $\text{m}^3$ ),  $C$  the airbox volume ( $\text{m}^3$ ),  $N$  the engine speed (rev/min),  $n$  the number of cylinders,  $T$  the air temperature (K),  $d$  the orifice diameter (m), and  $p$  the strokes of the piston per induction stroke.

For reasonable accuracy,  $U$  should not be less than 2.5, and the depression across the orifice should not exceed 100–150 mm of water.

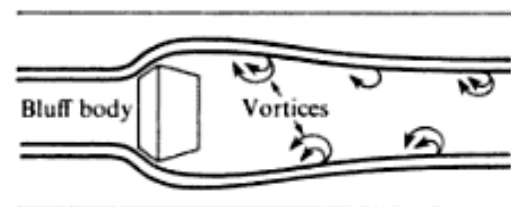
Another design of air meter is available for the measurement of airflow, this is the Alcock viscous-flow air meter and it is not subject to the errors of the simpler types of flow meters. With the air-box the flow is proportional to the square root of the pressure difference across the orifice. With the Alcock meter the air flows through a form of honeycomb so that the flow is viscous. The resistance of the element is directly proportional to the air velocity and is measured by means of an inclined manometer. Felt pads are fitted in the manometer connections to damp out fluctuations. The meter is shown diagrammatically in Fig. 13.39. The accuracy is improved by fitting a damping vessel between the meter and the engine to reduce the effect of pulsations.

Fig. 13.39 Alcock viscous flow air meter



An entirely different principle on which the velocity of an airflow can be measured is that of vortex shedding. As air flows over a bluff body, vortices or local regions of low pressure are created in the flow behind the body as shown in Fig. 13.40. A sensor of some kind, e.g. a pressure transducer, optical or ultrasonic sensor, built into the bluff body, registers the asymmetric conditions in the flow during vortex shedding and the frequency of the disturbance is measured. An alternative method is to record the frequency by a hot-wire anemometer placed in the airstream. The flow velocity  $C$  and the frequency  $f$

Fig. 13.40 Air flow measurement using vortex shedding



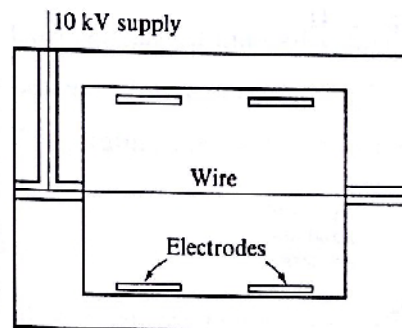


are calculated by the Reynolds number ( $Cd/v$ ) and the Strouhal number ( $fd/C$ ). The lower limit of  $Re$  is 1000 for vortex shedding to occur after which the Strouhal number is constant for a wide range of Reynolds numbers. Hence the volumetric flow rate is given by  $\dot{V} = K \times f$ , where  $K$  is the calibration coefficient for the instrument which can be refined in value by more elaborate calibration techniques.

The need to be able to measure accurately the instantaneous mass flow rate of air into an engine, or possibly into each engine cylinder, is an important requirement of modern transducer and instrument design and engine development work. One method of development is to use the pressure drop across an orifice or throttle plate, accurately measure and record the pressure difference, the air pressure and temperature, and then compute the air mass flow rate using modern microcomputing techniques. Another development is to seek new types of transducers or sensors which will respond, preferably in a linear manner, to the mass flow rate of air.

Such rapid response devices are particularly desirable when engine intakes suffer reversals in the direction of the airflow, and one principle on which a suitable unit may be devised is that of *corona* discharge, see Fig. 13.41. A thin wire maintained at a potential of 10 kV is fitted across the direction of the airflow and the discharge between the wire and the electrode collectors is disturbed by the airflow such that the discharge current to the electrodes is proportional to the mass flow rate of the air.

**Fig. 13.41**  
Measurement of air flow  
rate using a corona  
discharge method



## 13.12 Supercharging

Figure 13.14 (p. 439) shows how the power output of an engine is affected by the reduction in volumetric efficiency at increased engine speed. The purpose of supercharging is to raise the volumetric efficiency above that obtained with normal aspiration. The main attraction of supercharging engines is to obtain a high power output from a small engine hence giving a good power-weight ratio with a corresponding saving in space which is important in some applications such as road and rail vehicles. In this case the engine is designed as a supercharged engine to withstand the higher loads and temperatures reached in supercharging compared with normal aspiration.

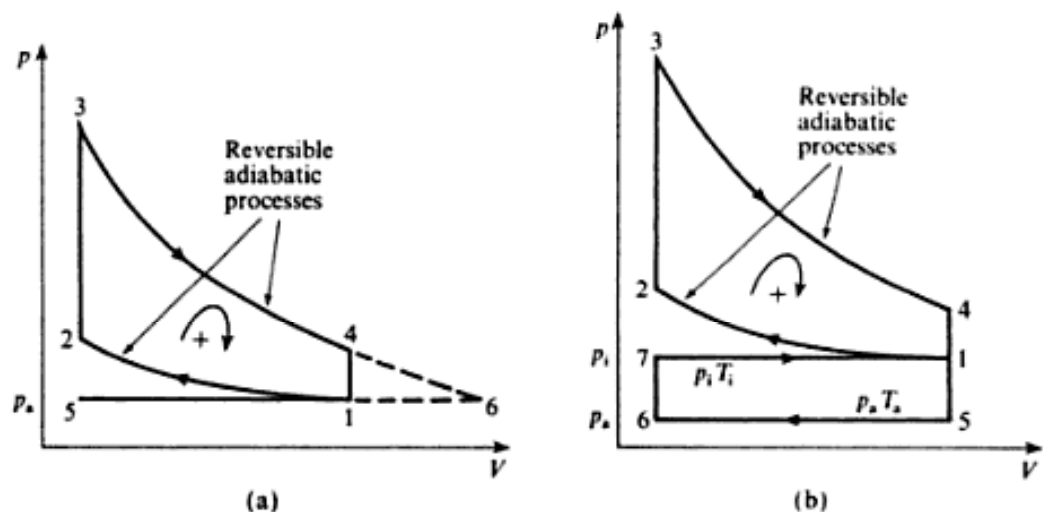
Greater benefits are to be expected from supercharging the diesel engine than from the petrol engine because of the different methods of charging the cylinders



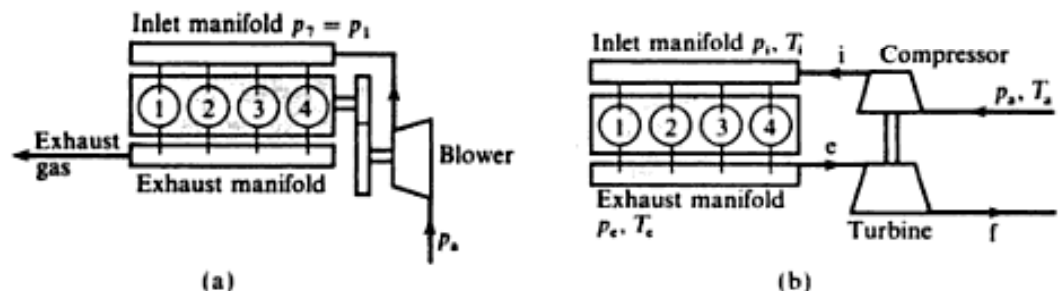
and the quite different combustion characteristics of the two types of engine. The diesel induces air only and the fuel is injected under pressure into the cylinder with self-ignition of the fuel in the air; the petrol engine induces a mixture of air and fuel which is spark ignited and burns as described in sections 13.6 and 13.7 giving fundamental combustion problems which do not occur with the diesel engine. To avoid charge detonation or 'knocking' in the petrol engine, giving uncontrolled combustion, the compression ratio may have to be reduced, an action which adversely affects the thermal efficiency of the engine. Alternatively a fuel of higher octane rating may be necessary. With the diesel engine higher boost pressures can give more satisfactory combustion conditions with a wider range of usable fuels, reduced delay periods, controlled pressure rise and an engine which is smoother and quieter in operation.

The main features of supercharging are illustrated in the  $p$ - $V$  diagrams for the idealized constant-volume four-stroke cycle in Fig. 13.42 and the plant line diagrams in Fig. 13.43. Figure 13.42(a) shows the normally aspirated cycle with line 1-5 representing both the induction and exhaust strokes at about the ambient air pressure  $p_a$ . The early applications of supercharging were for piston-engined aircraft in which the 'blower' was driven mechanically from the engine as shown in Fig. 13.43(a). The power output of the engine was increased by the higher flow of air, and hence the fuel consumed, but part of this increase in power was required to drive the blower. The effects on the  $p$ - $V$  diagram, as shown in Fig. 13.42(b), are to increase the pressures (and temperatures) reached during the cycle and to give a positive pumping loop, 15671, to add to the main working loop 12341.

**Fig. 13.42** Pressure-volume diagram for a four-stroke CI engine with supercharging (a) and without supercharging (b)



**Fig. 13.43** Diagrams of a four-stroke, four-cylinder CI engine with mechanical supercharging (a) and with turbocharging (b) (see p. 466)



The power required to drive a blower mechanically connected to the engine must be subtracted from the engine output to obtain the net bp of the supercharged engine. Then

$$\text{imep} = \left( \frac{\text{area 12341} + \text{area 15671}}{\text{length of diagram}} \right) \times \text{constant} \quad (13.16)$$

$$\text{and } \text{bp} = (\eta_M \times \text{ip}) - (\text{power to drive blower}) \quad (13.17)$$

(for mechanically driven blowers only).

### Example 13.3

The average ip developed in a CI engine is  $12.9 \text{ kW/m}^3$  of free air induced per minute. The engine is a 3-litre four-stroke engine running at 3500 rev/min, and has a volumetric efficiency of 80%, referred to free air conditions of 1.013 bar and  $15^\circ\text{C}$ . It is proposed to fit a blower, driven mechanically from the engine. The blower has an isentropic efficiency of 75% and works through a pressure ratio of 1.7. Assume that at the end of induction the cylinders contain a volume of charge equal to the swept volume, at the pressure and temperature of the delivery from the blower. Calculate the increase in bp to be expected from the engine. Take all mechanical efficiencies as 80%.

**Solution**

$$\text{Engine capacity} = 3 \text{ litres} = 0.003 \text{ m}^3$$

$$\text{Swept volume} = \frac{3500}{2} \times 0.003 = 5.25 \text{ m}^3/\text{min}$$

$$\text{Unsupercharged induced volume} = 0.8 \times 5.25 = 4.2 \text{ m}^3/\text{min}$$

$$\text{Blower delivery pressure} = 1.7 \times 1.013 = 1.72 \text{ bar}$$

$$\begin{aligned} \text{Temperature after isentropic compression} &= 288 \times 1.7^{(1.4-1)/1.4} \\ &= 335.2 \text{ K} \end{aligned}$$

Therefore

$$\text{Blower delivery temperature} = 288 + \left( \frac{335.2 - 288}{0.75} \right) = 351 \text{ K}$$

The blower delivers  $5.25 \text{ m}^3/\text{min}$  at 1.72 bar and 351 K.

$$\begin{aligned} \text{Equivalent volume at } 1.013 \text{ bar and } 15^\circ\text{C} &= \frac{5.25 \times 1.72 \times 288}{1.013 \times 351} \\ &= 7.32 \text{ m}^3/\text{min} \end{aligned}$$

therefore

$$\text{Increase in induced volume} = 7.32 - 4.2 = 3.12 \text{ m}^3/\text{min}$$

Increase in ip due to the increased induction pressure

$$= \frac{(1.72 - 1.013) \times 10^5 \times 5.25}{10^3 \times 60} = 6.2 \text{ kW}$$

$$\text{i.e. Total increase in ip} = 40.2 + 6.2 = 46.4 \text{ kW}$$

therefore

$$\text{Increase in engine bp} = 0.8 \times 46.4 = 37.1 \text{ kW}$$

From this must be deduced the power required to drive the blower.

$$\text{Mass of air delivered by blower} = \frac{1.72 \times 10^5 \times 5.25}{60 \times 287 \times 351} = 0.149 \text{ kg/s}$$

$$\text{Work input to blower} = \dot{m}c_p(351 - 288) = 0.149 \times 1.005 \times 63$$

therefore

$$\text{Power required} = \frac{0.149 \times 1.005 \times 63}{0.8} = 11.8 \text{ kW}$$

therefore

$$\text{Net increase in bp} = 37.1 - 11.8 = 25.3 \text{ kW}$$

Figure 13.42(a) shows the start of the exhaust process at 4 at a pressure substantially greater than the ambient,  $p_a$ . This means that over 60% of the cylinder charge is suddenly exhausted by a free expansion which constitutes a considerable loss of the energy released on combustion; of about 30–40% between diesel and petrol engines. The attraction of ‘turbocharging’ is evident as the energy lost in this way is used to drive a turbine wheel integral with a compressor wheel which delivers compressed air or charge to the cylinder. The additional work available from the gas is indicated, after continuing the reversible adiabatic expansion line 3–4 down to the pressure  $p_a$  at 6, by area 4614. The physical arrangement is shown in Fig. 13.43(b) and there is no mechanical connection to the engine. The turbocharger combination is a free-running unit with approximately equal mass flow rates over the turbine and compressor wheels reaching an equilibrium speed in the range 20 000–80 000 rev/min.

The simplest form of the supercharged cycle shows constant pressures created in the inlet manifold,  $p_i$ , and in the exhaust manifold,  $p_e$ , and it is essential that  $p_i > p_e$ , see Fig. 13.44(a). This pressure difference,  $p_i - p_e$ , can be utilized to scavenge residual gas from the combustion chamber if there is some overlap between the exhaust and inlet valve operation and particularly so for the diesel engine. This is called *constant pressure* supercharging and requires a large enough exhaust manifold to create a constant pressure supply to the turbine from a highly pulsating delivery from the engine cylinders. The  $T$ - $s$  diagram for the turbocharger is shown in Fig. 13.44(b) and, using the methods of Chapter 9, the energy balance for the unit is obtained as follows:

The compressor power input

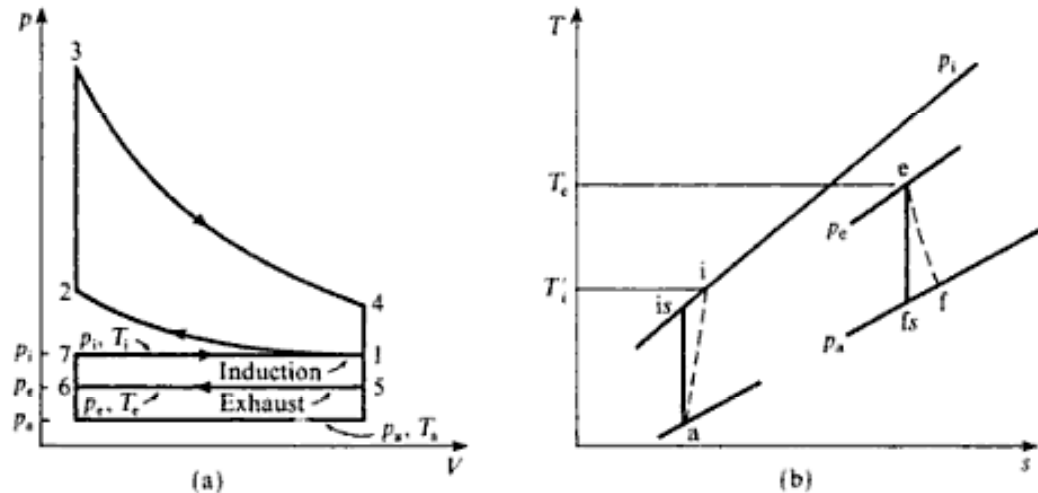
$$\dot{W}_c = \dot{m}_a c_{pa} T_a \left[ \left( \frac{p_i}{p_a} \right)^{(\gamma_a - 1)/\gamma_a} - 1 \right] / \eta_c$$

The turbine power output,

$$-\dot{W}_T = \dot{m}_e c_{pe} T_e \left[ 1 - \left( \frac{p_a}{p_e} \right)^{(\gamma_e - 1)/\gamma_e} \right] \times \eta_T$$



**Fig. 13.44** Pressure-volume diagram (a) and  $T$ - $s$  diagram (b) for the engine and turbocharger for 'constant-pressure' supercharging



where  $\eta_c$  and  $\eta_T$  are the isentropic efficiencies of the compressor and turbine respectively.

For a balance of mass flow rates

$$\dot{m}_c = \dot{m}_a + \dot{m}_f, \quad \frac{\dot{m}_c}{\dot{m}_a} = 1 + \frac{\dot{m}_f}{\dot{m}_a} = 1 + F/A$$

where  $F/A$  = fuel to air ratio =  $\dot{m}_f/\dot{m}_a$ .

Also  $\dot{W}_c = -\dot{W}_T \times \eta_M$ , where  $\eta_M$  = mechanical efficiency of the drive, therefore

$$\left[ \left( \frac{p_i}{p_a} \right)^{(\gamma_a - 1)/\gamma_a} - 1 \right] = \left[ 1 - \left( \frac{p_a}{p_e} \right)^{(\gamma_a - 1)/\gamma_a} \right] \frac{c_{pe}}{c_{pa}} \left( \frac{T_e}{T_a} \right) (1 + F/A) \times \eta_0$$

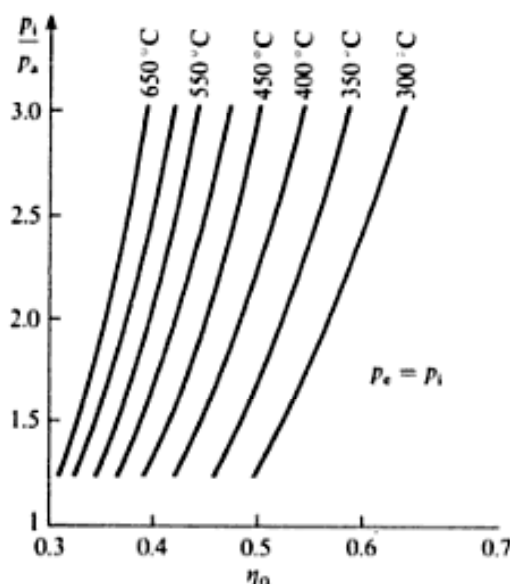
where  $\eta_0 = \eta_M \times \eta_T \times \eta_c$  = the overall efficiency of the turbocharger.

This expression shows how the manifold pressure  $p_i$  depends mainly upon  $\eta_0$  and  $T_e$  as the effect of the  $F/A$  ratio is small. A set of characteristics can be drawn of  $p_i/p_a$  against  $\eta_0$  for different values of  $T_e$ . A set is obtained for each value of  $p_e$  and the minimum requirement to sustain the limit is  $p_e = p_i$  as shown in Fig. 13.45.

For example for  $p_i/p_a = 2$ , and  $T_3 = 773$  K ( $500^\circ\text{C}$ )  $\eta_0 = 42\%$ . The overall efficiency is higher for lower values of  $p_e$  and higher values of  $p_i$  and  $T_3$ . The study of turbocharging should continue into the design of the turbocharger unit to meet its service requirements, but this is outside the scope of this book and specialist references such as 13.8 should be consulted.

The usual arrangement for a turbocharger is a single-stage centrifugal compressor driven by a single-stage axial-flow turbine for the medium and large-size engines for industrial, rail, and marine applications, and by a radial-flow turbine for the smaller engines used in automotive applications, transport vehicles and cars. It is somewhat against earlier expectations that most car manufacturers now include supercharged petrol-engined cars in their product range with apparent overlap with their normally aspirated engines of different capacities. This is in spite of the fact that cars spend a great deal of their time at part throttle and that additional control is necessary to restrict the boost pressure and prevent the onset of knock by retarding the ignition.

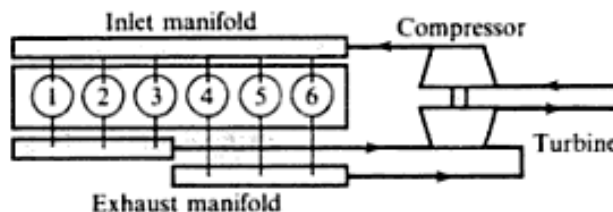
**Fig. 13.45**  
Turbocharger characteristics of pressure ratio against turbocharger overall efficiency for various exhaust gas temperatures



The above description has been confined to the constant-pressure charging of four-stroke cycle engines. The blowing of the two-stroke engine is attractive as the cycle does not include a separate exhaust stroke, and a means of improving the scavenging process would improve the breathing and hence the power output. The two-stroke is particularly sensitive to exhaust back pressure which is increased by turbocharging and its use would require additional care in port timing and the matching of characteristics.

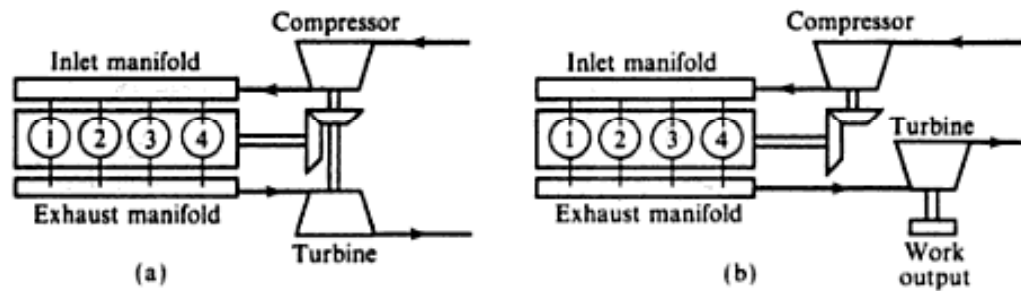
In any cycle the gas leaves the engine cylinders at high speed through the opening valve and possesses high kinetic energy which is dissipated in the large manifold of the constant-pressure system. To utilize the kinetic energy more fully a *pulsed* system of charging can be used provided the turbocharger is designed to cope with the conditions created. For multicylinder engines the cylinders are grouped, taking cylinders alternately from their firing order for entry into the turbine and this also groups them into the front and rear cylinders, as shown in Fig. 13.46. To improve the charging of the cylinders the compressed air can be cooled by passing it through a water- or air-cooled intercooler thereby increasing its density.

**Fig. 13.46** Grouping of cylinders in a turbocharged multicylinder CI engine



The turbine and compressor are complex devices and further study will show that the complexities of design, matching of the units to each other and to the engine, etc. are such that the performance may not be satisfactory over the whole speed range. Compound engines have been proposed, and some have been built which are turbocharged and have a mechanical connection between the engine crankshaft and the turbocharger as shown in Fig. 13.47(a). A fixed

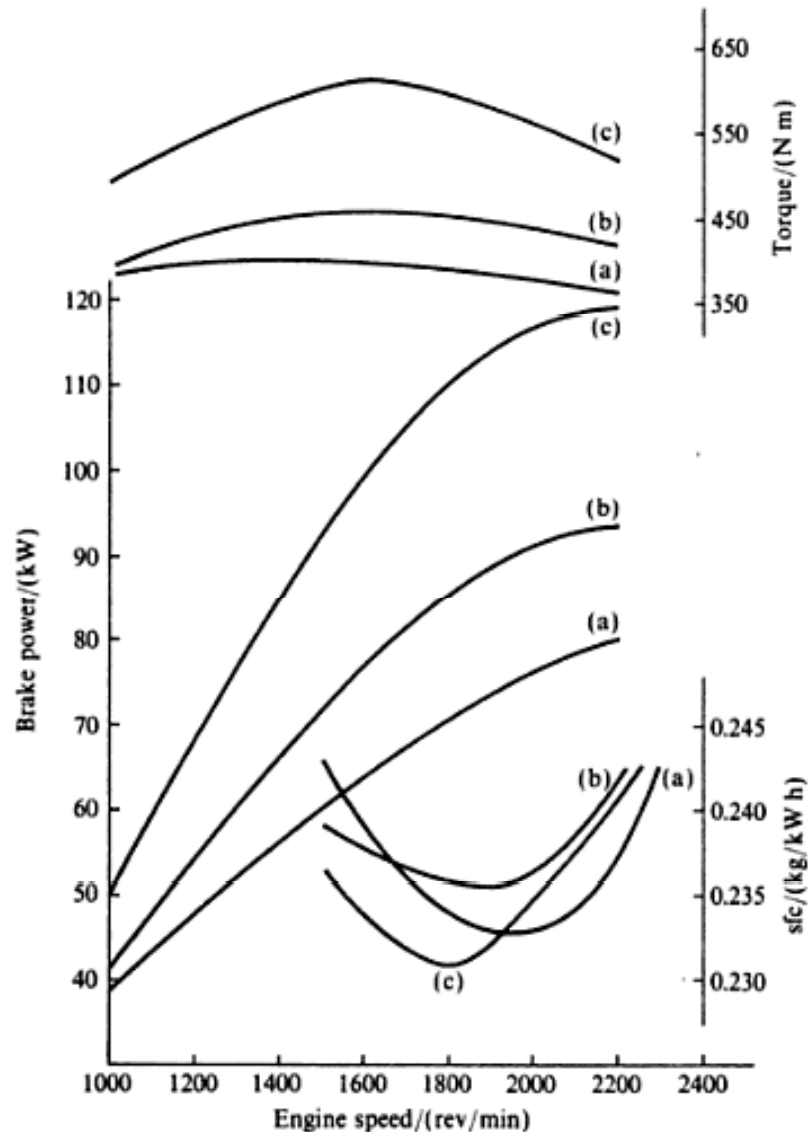
**Fig. 13.47** Two compound arrangements of CI engine and gas turbine



drive improves the low-speed performance but restrains the turbine at high speeds. A variable-speed drive is called for but this is an expensive addition. An extreme arrangement for a compound engine is shown in Fig. 13.47(b) in which the engine drives the compressor only and the exhaust gas drives an independent turbine from which the power output of the combination is taken.

Figure 13.48 shows the continuous running performance characteristics of a diesel engine in three modes: normally aspirated (a), turbocharged (b), and turbocharged with intercooling (c). The engines are of 6.61 capacity

**Fig. 13.48** Continuous running performance characteristics of a CI engine in three modes: (a) normally aspirated; (b) turbocharged; (c) turbocharged with intercooling





(bore = stroke = 112 mm) with a compression ratio of 16.3 for (a) and 15.6 for (b) and (c). The maximum cylinder pressures were 90, 121, and 138 bar respectively for (a), (b), and (c).

### 13.13 Engine emissions and legal requirements

An atmospheric phenomenon called 'smog' was experienced on an increasing scale in the State of California, USA, around the city of Los Angeles until it became a severe social problem with political and subsequently commercial and engineering consequences which have had a fundamental effect upon the design and operation of the IC engine.

Smog is a type of light fog which is unpleasant and a cause of irritation to the eyes and nasal passages, and although not affecting visibility greatly does affect vegetation and has caused serious economic losses in horticulture and agriculture. Smog is created by the action of sunlight on hydrocarbons (HC) in the atmosphere and the main source of HC is the exhaust gases of motor vehicles, the rapid increase in volume of town traffic causing the increase in smog. It was against a background of increasing public nuisance and distress that, in 1960, the State of California legislated against the motor vehicle and stated the procedures against which vehicles would be tested and the levels of pollutants which would have to be met by vehicles to be sold in that State. The procedures were extensive and covered the types of emissions to be controlled, HC and CO, the sources of emissions (exhaust, crankcase, carburettor, and fuel tank), the test procedures, the instrumentation and the test equipment to be used.

The initial aim of the Californian Motor Vehicle Pollution Control Board was to reduce the total exhaust emissions to pre-1940 levels by 1980. This was to be done by legislation against new vehicles, as no requirement was made for existing vehicles to be modified, and included the effects of a continuous increase in the number of vehicles on the road. Although mainly American manufacturers of vehicles were affected so were the manufacturers of imported vehicles and these were penalized by having to meet prescribed rates of emission output, and not total amounts, as their engines were generally of small capacity.

The legislation was introduced before the process of smog creation was understood and it was believed that CO and unburned HC were the cause. Mixture strengths were weakened to reduce the CO and HC output, but the smog levels were unaffected and it was later established that the oxides of nitrogen ( $\text{NO}_x$ ) and the unburned HC combined photochemically in the strong sunlight to create the smog. The Californian legislation was followed by that of the US Federal Government through the Environmental Protection Agency (EPA) although smog was not the primary concern in other parts of the USA where the atmospheric conditions for its creation did not occur. The Californian and federal authorities each have emission standards.

The introduction of emission controls for motor vehicles has spread in the wake of the legislation introduced in the USA to other countries including Japan, the EC, Canada, Sweden, the UK, Australia, and Finland. In Europe the most significant effort has been to reduce the use of leaded fuels as a means

of raising the knock rating of petrol. The reduction of the lead content in fuels is a response to the demand to reduce the quantity of airborne lead particles in the atmosphere, 90% of which are attributed to petrol consumption. Inhaled lead is a smaller contributor to the consumed lead in the human body than food or water but it is not negligible. The lead content of fuel can be reduced if either a lower knock rating or an increase in the cost of fuel processing is accepted. Premium petrol has a research octane number of 98.5 with a lead concentration of 0.4 g/litre. To reduce this to 0.15 g/litre would require an increase in crude oil consumption of 0.5% by mass at the refinery – a much more expensive operation than the proportion suggests – but which can be avoided by accepting the octane rating of 95.5 as a consequence of the reduction in the lead content to 0.15 g/litre. This in turn could lead to a reduction in usable compression ratio, and hence thermal efficiency of the engine, and an increase in fuel consumption. In Japan and North America lead has been removed from petrol because it poisoned the catalyst of the chemical converters fitted to engines to reduce the exhaust emissions and not for environmental reasons related to the lead itself.

The legislation of each country requires progressive reductions to be made in the HC, CO and NO<sub>x</sub> content of the exhaust gas from vehicles and this anticipates that the technology will be available to achieve the levels defined by the time stated. The requirement in the USA in 1970 was to obtain a 90% reduction in emissions from the 1970 levels within a stated time-scale. The aims of the different countries are not directly comparable as there are differences in the sampling and measuring techniques over test cycles which are selected to represent the driving patterns of the country concerned. The diesel engine is low on HC and CO emissions but is higher on NO<sub>x</sub> and particulate emission, the latter being 10 times that of a comparable petrol engine; the technical objective is to reduce the levels without losing on fuel economy.

Over the years during which emission control has applied the fuel consumption of the smaller-engined vehicles has improved, but the larger engines have given better emission figures at the expense of fuel economy as the engine has had to be 'detuned' by adjusting mixture strength and ignition timing to meet the emission requirements. The optimum settings were restored when catalytic reactors were fitted to the engines to reduce the emission levels chemically by units fitted external to the engine combustion systems.

Some approaches to emission control such as increasing the air-fuels ratio and recirculating part of the exhaust gas through the cylinders reduces the CO and HC output, controls the NO<sub>x</sub>, and does not increase the fuel consumption, but is comparatively cheap in the additional facilities it requires on the engine. The modification to basic engine parameters such as those affecting mixture preparation and spark ignition involve small costs, but others which require thermal reactors or catalytic converters to be added as additional units can involve large cost increases. Advanced aspects of engine design involving stratified charge, with torch ignition or open chamber systems, are expensive to produce but have considerable effects on emission levels, increasing the possibilities if taken together with exhaust gas recirculation and a catalytic converter.

A vehicle without emission control contributes to the pollution from several

sources with unburned HC coming from the fuel tank vent and the carburettor bowl vent to the atmosphere; from piston blow-by into the crankcase and leaks from fuel lines as well as from the exhaust gas itself which puts, in addition, CO, CO<sub>2</sub>, NO<sub>x</sub>, and particulates into the atmosphere. Crankcase emissions are dealt with by venting all crankcase fumes directly into the engine intake system to be burned in the engine cylinders.

The exhaust gas emissions are affected by many of the engine variables which also control the completeness of the combustion process, an appreciation of which is essential to an understanding of emission control. The way in which engines are used in the vehicles has a complex effect on emissions, and so it was necessary to devise representative test methods to reproduce on the road running as closely as possible on the test rig. As mentioned earlier, different countries have different test procedures and the following comments can only be regarded as being general to the problem.

A chassis dynamometer is used to load a vehicle to reproduce the road condition with regard to wind and inertia loads, the latter being experienced during acceleration and deceleration. This is not easy to achieve as the tyre/roller contact is quite different from that between tyre and road, and the engine operating parameters may not be the same as those experienced on the road for apparently corresponding conditions.

The drive cycle includes periods of acceleration, deceleration, steady-state cruising, and idling and the exhaust gases are continuously analysed for CO, CO<sub>2</sub>, NO, and HC (as a hexane equivalent). The analyses are recorded and may be interpreted by a weighting procedure to give an estimate of the total mass emitted over the test cycle. Acceleration periods have high weight factors and idle periods have low. More recent techniques described as 'constant-volume sampling' include the collection of quantities of gas in bags either during or over all of the test cycle for subsequent analysis to a mass basis.

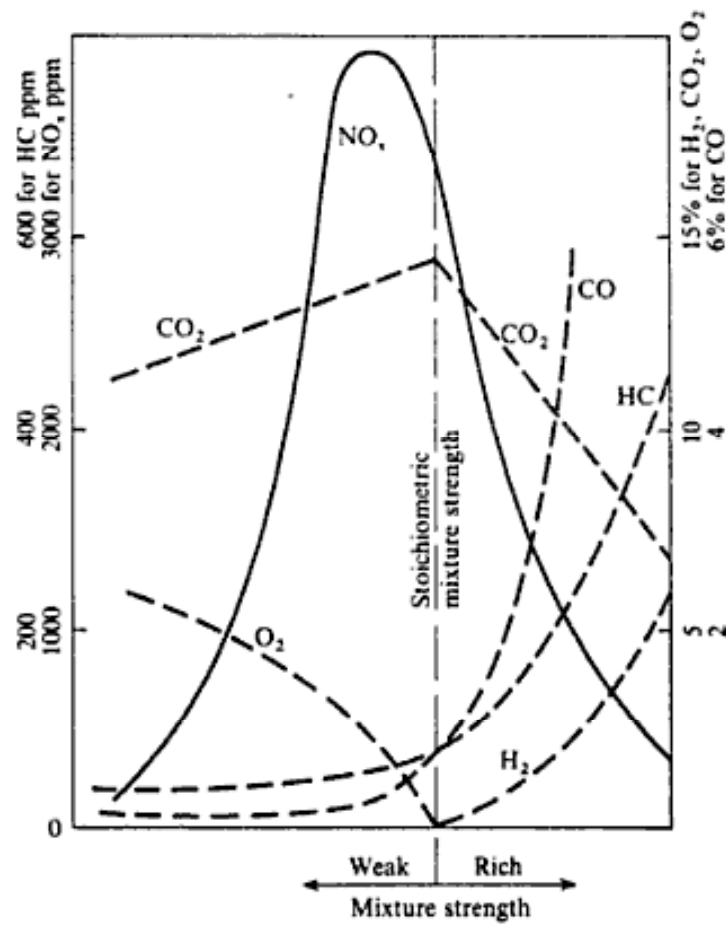
As the test procedures have developed and experience has been gained, the test techniques have become established in relation to specified instrumentation (see section 7.6).

Emission control is a complex matter which was brought into existence by legislation and has had to develop as a science since. Different countries have emission-control programmes to meet their own requirements albeit related to the experience and procedures pioneered in the USA. The EC countries have their set of tests which have been adopted by member countries to their own requirements to suit their road systems, vehicle population, and atmospheric conditions.

The number of official tests for all of the countries involved is high and the cover is comprehensive including the measurements of HC, CO, CO<sub>2</sub>, NO<sub>x</sub> and smoke (for diesel engines) with different classifications of tests for vehicles of different size, light- and heavy-duty passenger vehicles and trucks, petrol and diesel engines, the tests including different procedures and modes of testing.

Emissions vary with the engine parameters and tests can be carried out on research engines with the facilities of variable parameters to investigate the characteristics of emissions. Some of these are illustrated in Figs 13.49–53 inclusive, for petrol engines, and serve to show the complexity of the processes involved. It is probably true to say that the legal emission requirements have

**Fig. 13.49** Emission characteristics of a SI engine showing HC and  $\text{NO}_x$  against mixture strength



**Fig. 13.50** HC emissions for a SI engine against air-fuel ratio

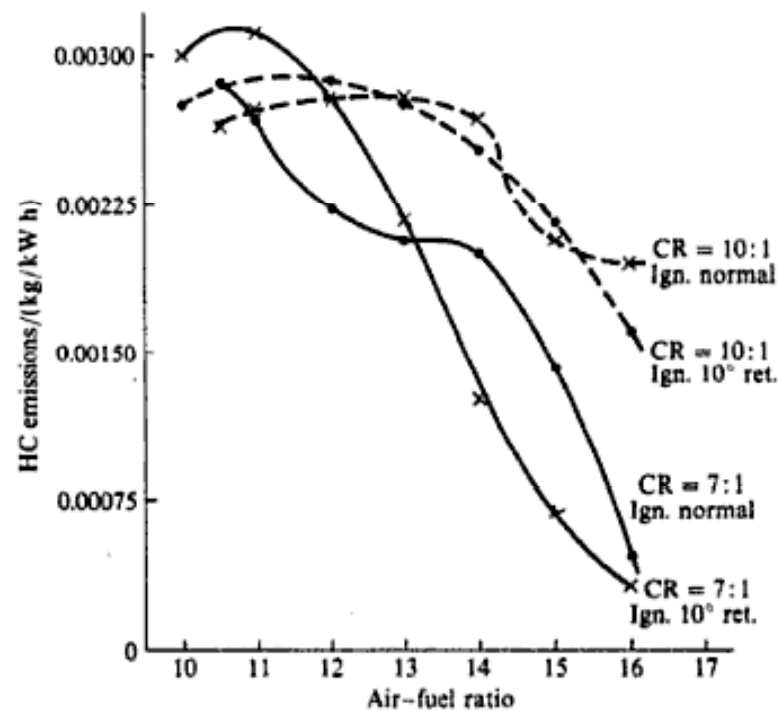




Fig. 13.51  $\text{NO}_x$  emissions of a SI engine against air-fuel ratio

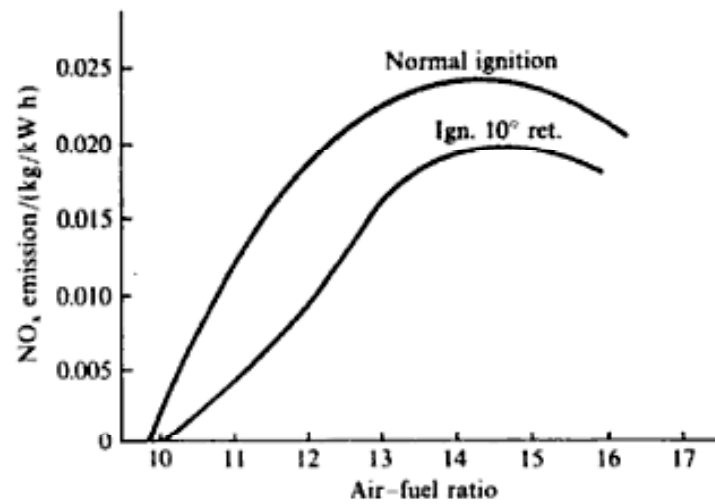


Fig. 13.52 CO emissions of a SI engine against air-fuel ratio

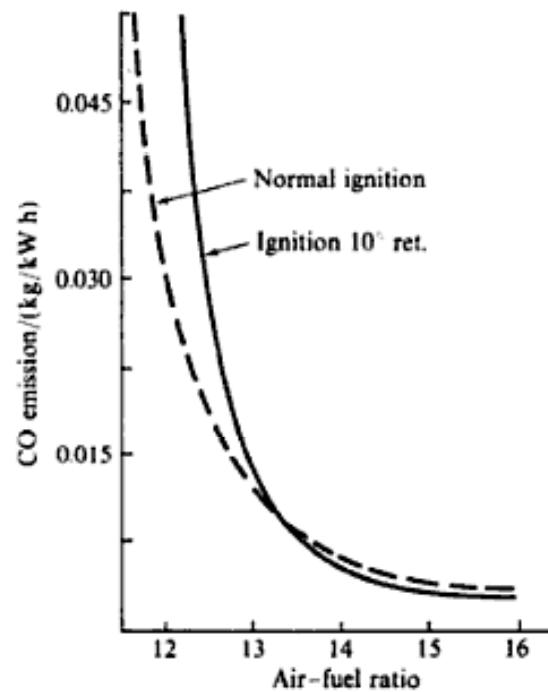
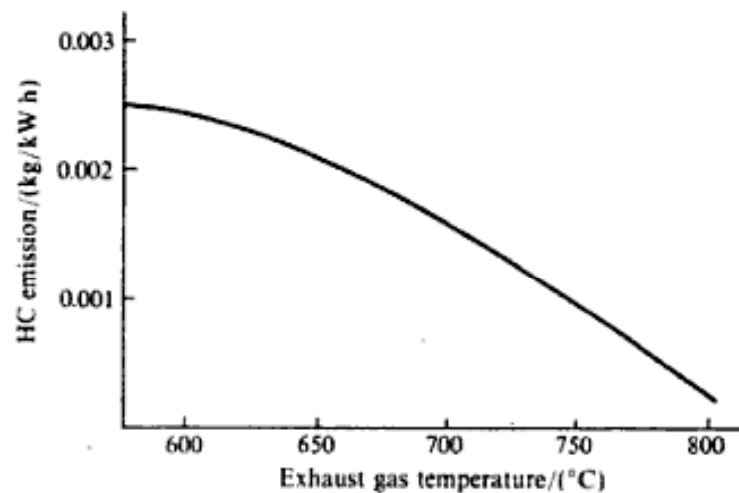


Fig. 13.53 HC emissions of a SI engine against exhaust temperature



added a new dimension to the development of the IC engine and have probably led to more fundamental research into the process of internal combustion since the onset of legislation than was done before, (see ref. 13.9).

The pollution problem is closely linked to that of fuel economy with the increase in cost of fuel and oil and the need to conserve natural resources. It has therefore become essential to rationalize the experience of generations of engine builders and integrate modern, analytical, computational, and experimental methods into the science of IC engineering. Section 13.15 describes some of the modern developments in IC engines which are a result of advanced understanding of the basic processes and modern techniques. Further progress can be made by studying research papers on individual topics.

### 13.14 Alternative forms of IC engines

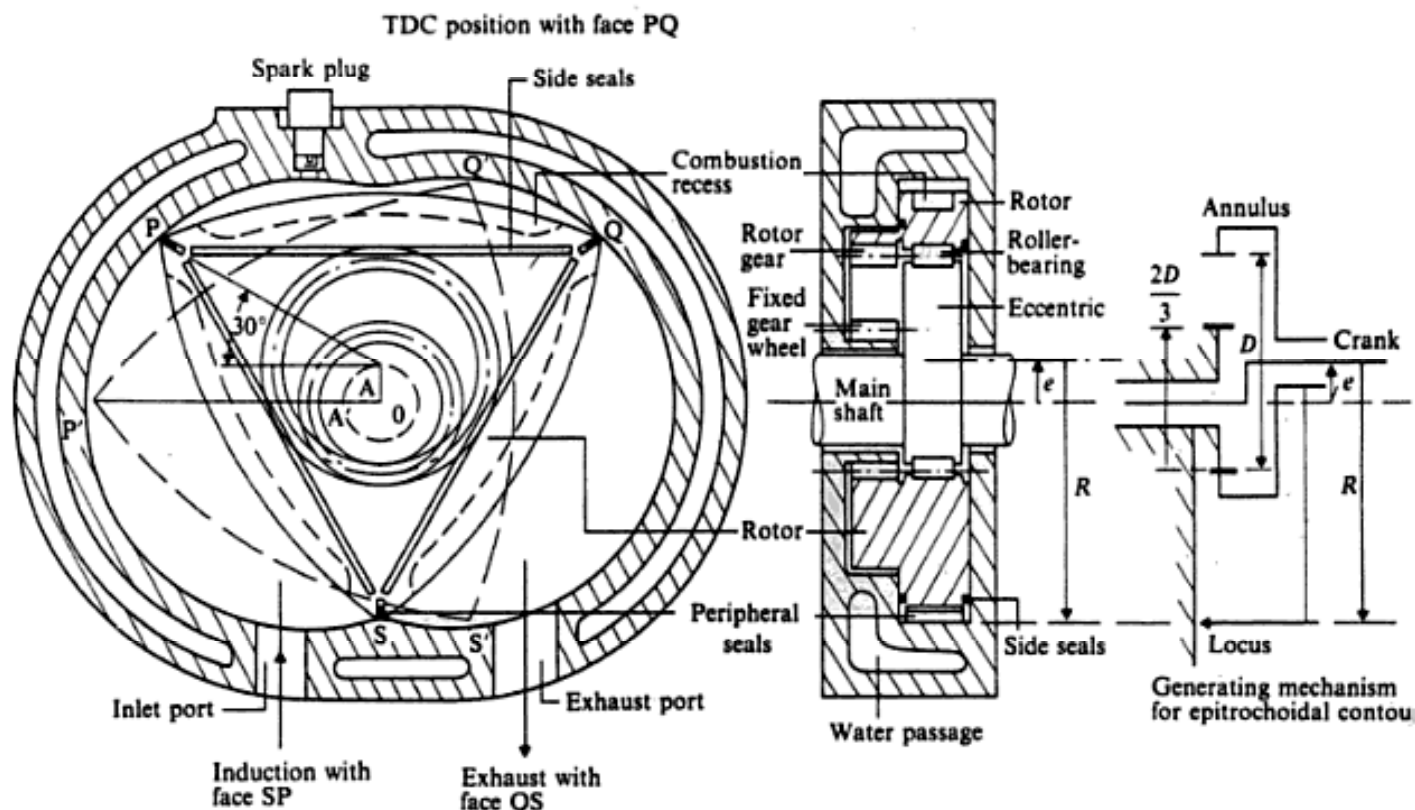
In recent years there have been several attempts to produce power units which would be superior to the IC engine in its conventional forms. Some attempts have been merely to improve the breathing of the reciprocating engine by alternative designs of the valve mechanism. The more ambitious projects have had the object of basic improvements, and have included engines which have a fundamentally different geometry.

#### The Wankel engine

One engine with this object is the Wankel rotary engine, the most successful of many proposed rotary engine designs. For many years inventors have worked to produce an engine which would fulfil all the promises of the rotary engine concept, with improved power-to-weight ratio, and compete successfully with the reciprocating engine. A simplified representation of the Wankel engine is shown in Fig. 13.54 and the engine is briefly described below.

The rotor, which has a profile defined by three circular arcs PQ, QS, and SP, is attached through a roller-bearing to an eccentric which is an integral part of the engine main shaft. The bearing (and rotor) centre is at an eccentricity  $e$  to the mainshaft centre. The rotor radius  $R$  to the apex points P, Q and S is the generator radius for the enclosing 'cylinder' or housing. The profile of the cylinder is of epitrochoidal form and is followed by the rotor by means of the epicyclic gear formed by the gear-wheel which is fixed to the casing and the internal gear of the rotor. The pitch circle diameter of the fixed gear-wheel is two-thirds that of the rotor gear-wheel. With this mechanism the rotor turns at one-third of the speed of the mainshaft (or eccentric). As each of the three faces of the rotor is concerned in turn with one power cycle, for each revolution of the rotor the mainshaft receives one working 'stroke' per revolution. Thus the single-rotor Wankel compares with a single-cylinder two-stroke cycle engine.

The rotor is shown in Fig. 13.54 in two positions, one in dotted outline and the other in full line; for the former position the eccentric  $OA'$  ( $= e$ ) and the generator radius  $A'P'$  ( $= R$ ) are in line to show  $OP'$  equal to half the length of the major axis ( $= R + e$ ). The rotor turns clockwise and to give the position



**Fig. 13.54** Simplified representation of a Wankel engine

*P* the mainshaft (or eccentric) has turned through  $90^\circ$  from *OA* and the rotor through  $30^\circ$  (i.e.  $90/3$ ). By continuing this construction for other mainshaft positions the epitrochoidal profile can be generated and is repeated for apex *S* which is displaced  $120^\circ$  with regard to *P* and similarly for *Q*. For the full line position *OS* forms half of the minor axis length ( $= R - e$ ). In the diagram the pitch circle of the bearing coincides with that of the rotor gear for simplicity of drawing.

The three parts of the cylinder are sealed by peripheral seals at *P*, *Q*, and *S* and side seals which are held in the flanks of the rotor on both sides and these constitute a sealing system corresponding to that of the piston rings and valves in the reciprocating engine. The peripheral seals take gas loads only and do not influence the movement of the rotor.

In the full line position shown *PQ* is at TDC and defines the minimum or clearance volume of the engine. This volume can be varied by forming a recess in the rotor. The maximum cylinder volume will be shown when *S* takes up a position just on completion of its movement across the inlet port. The difference between the two volumes gives the swept volume of the engine per rotor face. As *PQ* moves towards the position occupied by *QS* the working stroke is performed and exhaust begins when the cylinder space is uncovered by the seal as it passes over the exhaust port. Continued movement of the rotor opens the space to the inlet and the induction of the fresh charge and so there is a considerable overlap between exhaust and inlet phases and this can be considered to be a basic limitation of the design. When the apex seal passes over the inlet port, compression of the charge begins up to the TDC position after which the

cycle is repeated. To the right of Fig. 13.54 a 'skeleton' figure shows the essentials of the 'generating' mechanism.

The basic design is attractive and offers advantages in comparison with a reciprocating four-stroke engine with regard to compactness for a given power output, fewer working parts, lower mass, lack of vibrations and elimination of the poppet valve. There have been many development problems which are not yet solved to the extent that the automotive market shows a significant conversion to the rotary engine.

Some of the criticisms are basic and include the belief that the combustion space is the wrong shape (long and narrow) for good combustion and the limitations on porting are a disadvantage for good breathing, particularly at low speeds. These are fundamental to the epitrochoidal geometry which also limits the attainable compression ratio, thereby making a diesel version impossible without increased complication such as two cylinders working in series. Another outcome of the geometry is the re-entrant shape or 'cusp' on the minor axis which the peripheral seal finds difficult to follow at high speed because of the reversal of acceleration experienced. The effectiveness of the sealing is held to be in doubt in spite of the considerable development work which has taken place on these. It is also believed that the rotor bearing is subject to adverse conditions, being close to the hot rotor surface without adequate cooling. Solutions to these problems have been applied such as dual, phased ignition and side porting as well as peripheral ports and the use of different seal designs and materials. The assessment and development of the Wankel engine has been undertaken by companies in many countries of the world and for many applications other than for motor vehicle purposes, for much smaller engines and much larger power, lower speed units.

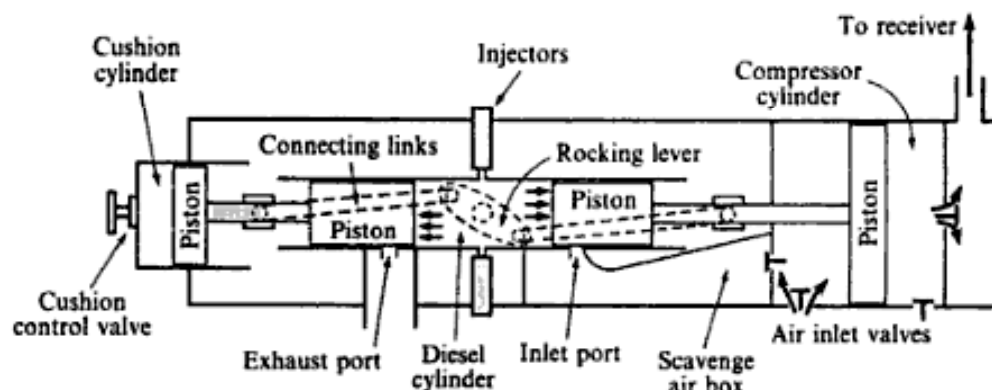
The SI petrol engine is predominant in the light power and high-speed field, and for automobile purposes, but is receiving some competition from the high-speed diesel engine. They have geometries which are basically identical and both are subject to continuous development; it is not likely that they will be replaced to any great extent in their particular field by any other unit for some time. Other power units may appear for particular purposes and may be modifications to the conventional layouts, or they may be combined units.

### Free piston engine

The *free piston* engine is a unit of interest and is referred to in section 13.3 in an introductory manner. Two versions are referred to: the free piston air compressor and the free piston gasifier. The free piston air compressor cycle will be described and the gasifier action can be deduced from it. The free piston engine is usually constructed as an opposed-piston, two-stroke diesel cycle with a conventional fuel injection system, but it could be a gas engine or have spark ignition. In Fig. 13.55 the engine/compressor unit is shown on its power stroke. The two diesel pistons are connected mechanically and externally by lightly loaded links through a centrally situated rocking lever. Alternatively the connection may be by a twin rack and pinion arrangement. On the power stroke the air in the compressor cylinder is compressed and then delivered to



**Fig. 13.55** Free piston engine operating on a two-stroke CI cycle



the receiver. At the same time scavenge air enters the compressor cylinder on the underside of the piston, and the air in the cushion cylinder is compressed. Towards the end of this stroke the exhaust port is uncovered and exhaust begins. Slightly later in the stroke air enters from the scavenge air box to complete the exhaust process and to charge the cylinder. The piston then returns on the compression stroke in the diesel cylinder due to the pressure of the air in the cushion cylinder assisted by the air pressure in the compressor cylinder. The air from the underside of the compressor piston is forced into the scavenging air box during this stroke and is ready for the next scavenge process. At the inner dead centre, ignition occurs and the cycle is then repeated.

When used as a gasifier the two pistons have identical cushion cylinder arrangements and no power is taken from the engine. The HP exhaust gas, which is at a moderate temperature ( $<480^{\circ}\text{C}$ ), is taken from the engine to an expansion turbine from which the whole of the mechanical power output is taken. This combined unit gives an overall thermal efficiency which is higher than that obtainable with a complete turbine unit fitted with a heat exchanger. The output of the gasifier is given as a *gas power* (gp), which is calculated on the assumption of isentropic expansion of the gas leaving the cylinder, and is the potential power.

The free piston engine can utilize a wider range of fuel oils than the conventional diesel engine and has a higher compression ratio, which indicates a higher thermal efficiency. The compression pressures in the diesel cylinder are in the order of 69 bar, and the thermal efficiency is 40–45%, which gives an overall thermal efficiency of about 35%.

### Gas turbines for vehicles

The gas turbine has been a competitor to the piston engine for use in vehicles apart from its natural field of application in the higher power range for aircraft, industrial turbines, power stations, and small ships. For vehicle use the gas turbine appeared as the power unit in a few prototype cars, but it is for the heavier vehicles, trains, buses, and the larger trucks that the gas turbine has more to offer. The small power single-shaft turbines cannot give competitive fuel consumptions to the petrol engine without a highly effective heat exchanger and even then, although the full load figures are good, the part load fuel

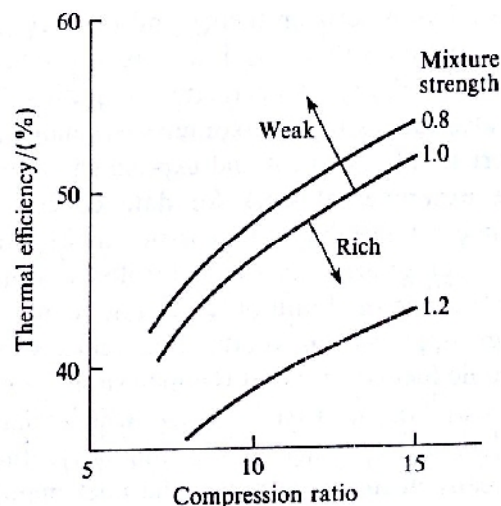
consumption is poor. The advantages offered by the gas turbine are small volume and mass for a given power output giving a good power-to-weight ratio, a light and relatively simple and robust construction, a potential for burning a wide range of fuels giving a running cost advantage if cheaper fuels are available, and low emission output. Power units can be scaled down to give the required power output, but the inherent source of losses in the engine, e.g. essential running clearances, cannot be scaled down and so some losses are proportionately increased.

In the search for an alternative power unit to the petrol and diesel engines interest in the Stirling engine, see section 5.10, has had some revival. It has external continuous combustion, high thermal efficiency, quiet operation, good torque at low speed, and low emissions, but requires a highly effective heat exchanger, as for the gas turbine, which must be cheap in construction and reliable in operation at high temperatures ( $> 1000^{\circ}\text{C}$ ) over the life of the engine.

### 13.15 Developments in IC engines

There is still no serious replacement for the SI and CI engines for the mass-produced power units required for cars and other road vehicles and this is likely to remain so as long as the existing fuels are available. Petrol and diesel engines have been subject to continuous development to improve their performance particularly with respect to fuel consumption and specific power output. The constant volume cycle analysis in section 5.6 shows the fundamental dependence of thermal efficiency on compression ratio. This study can be extended to deal with air-fuel cycles by the selection of appropriate values of  $\gamma$  for the mixture. As the fuel content is increased the value of  $\gamma$  falls and a plot of the thermal efficiency against compression ratio for mixture strengths 0.8, 1.0 and 1.2 is drawn in Fig. 13.56, showing the increase in efficiency obtained with the weaker mixtures. The development problem is to incorporate these principles into practical engine design; prototype engines, radically designed and shown to give improved performances, have appeared only very slowly as production

Fig. 13.56 Thermal efficiency of a SI engine against compression ratio for different mixture strengths



engines. The simple reciprocating piston-in-cylinder arrangement offers scope for development mainly in the combustion chamber, the induction and exhaust system, the preparation and movement of the combustible charge, and in the timing and quality of the ignition. The use of multiple valves, e.g. two inlet and two exhaust or two inlet and one exhaust, improves the charging and exhausting of the cylinder if the added complexity and cost are acceptable; the former arrangement requires two overhead camshafts and the latter can be done with one. In addition to the quantity of gas induced, the movement of the charge is important with the gas swirling as the piston descends on induction and through compression until the gas is forced by an increasing amount of 'squish' into the combustion chamber as it is squeezed between the rising piston and the flat face of the exposed cylinder head. The purpose of combustion chamber design is to obtain the necessary characteristics and the various shapes are described as bathtub, hemispherical, pentroof, etc. The fireball design is an advanced form of combustion chamber in which the main part of the volume is concentrated around the exhaust valve and charge movement is encouraged by the shape of path from the inlet valve side to the main part of the chamber round the exhaust valve. This design has shown good improvements in the average fuel consumption figures, with prospects of 20% being achieved with production engines, because of its ability to burn lean mixtures due to the high temperatures obtained with the high compression ratio and the highly turbulent movement of the charge produced by the geometry of the combustion chamber. There are other forms of stratified charge/combustion chambers, such as the divided chamber of the Honda CVCC engine and the Ford PROCO open chamber, which have shown improvements in mixture control and performance.

Modern engine development starts with an understanding of the characteristics of the combustion process arising from the theoretical modelling of this highly complex two-phase reaction which describes the initiation of the ignition and the propagation of the flame through a mixture of the fuel-air charge induced and the proportion of residual exhaust gas in the cylinder. This must describe the nature of the flame front, the rate of burn, the charge turbulence and combustion kinetics. The equations established are complex and require a great deal of computing time. The prediction by such programmes has to be compared with engine results and then the models are further refined to improve the correlation between theory and practice.

Such complex theoretical investigations have only become possible because of the availability of adequate computing facilities, but the impact of the computer and microprocessor have not ended there. The computer is a powerful support to all analytical and experimental work and can be involved directly in the experimental work for data collection, analysis, and control of the experiment. Once the characteristics are known, from theory or experiment, the combustion process must be controlled and increasingly the microprocessor is in evidence as the brain of the engine management system. In the early 1980s engines appeared in production vehicles with programmed ignition and electronic fuel control and the inclusion of knock sensors to warn the ignition control unit of the onset of 'knock' which then retards the ignition of individual cylinders to avoid detonation. Similarly, during warm-up the output of an electrically heated hot spot in the inlet manifold is automatically reduced as



the engine warms up and is switched off at the normal running temperature. Fuel supply is controlled by microchip logic as an automatic choke, low speed control, and overrun cut-off. Apart from the control of the basic engine parameters computer electronic control also leads to simpler ignition devices of more robust construction than the traditional contact-breaker type, requiring less maintenance and giving longer life.

The experimental investigation of engine performance has made increasing demands for refined techniques, improved transducers and instrumentation for the measurement of gas flow, turbulence, pressure, temperature, and charge composition throughout the process. It is necessary to be able to take the measurements in production engines under test-bed conditions and also, although it is more difficult, in the vehicle in which the engine is used if that is its application. A modern technique to allow such measurements to be made on the test bed is that which employs lasers such that no intrusion is made into the combustion process, a fault of the earlier techniques which by the presence of transducers or other measuring devices changed the combustion system.

One of the disadvantages of the SI engine is the cycle-to-cycle and cylinder-to-cylinder variation in performance which is readily illustrated by taking a continuous sample of pressure/crank angle diagrams. Investigation has shown that a significant factor in the cycle-to-cycle variation is the size of the eddies created by the induction system. The smaller the eddy size variation about a critical mean size the more consistent the combustion process, but the grids used to control the generation of eddies, although improving the efficiency of combustion, cause restrictions to gas flow and hence reduce the full throttle volumetric efficiency and power output. Greater knowledge of the induction and combustion processes supported by reliable experimental measurements should lead to further improvements in engine power output and fuel efficiency. For further reading see references 13.4 and 13.10.

## Problems

- 13.1** A quality governed four-stroke, single-cylinder gas engine has a bore of 146 mm and a stroke of 280 mm. At 475 rev/min and full load the net load on the friction brake is 433 N, and the torque arm is 0.45 m. The indicator diagram gives a net area of 578 mm<sup>2</sup> and a length of 70 mm with a spring rating of 0.815 bar per mm. Calculate the ip, bp, and mechanical efficiency.  
(12.49 kW; 9.69 kW; 77.6%)
- 13.2** A two-cylinder, four-stroke gas engine has a bore of 380 mm and a stroke of 585 mm. At 240 rev/min the torque developed is 11.86 kN m. Calculate: (i) the bp; (ii) the mean piston speed in m/s; (iii) the bmep.  
(298.1 kW; 4.68 m/s; 11.23 bar)
- 13.3** The engine of Problem 13.2 is supplied with a mixture of gas and air in the proportion of 1 to 7 by volume. The estimated volumetric efficiency is 85% and the  $Q_{\text{net},p}$  of the gas is 38.6 MJ/m<sup>3</sup>. Calculate the brake thermal efficiency of the engine.  
(27.4%)



- 13.4** A four-cylinder racing engine of capacity 2.495 litres has a bore of 94 mm and a compression ratio of 12/1. When tested against a dynamometer with a torque arm of 0.461 m a maximum load of 622 N was obtained at 5000 rev/min, and at the peak speed of 6750 rev/min the load was 547 N. The minimum fuel consumption was 17.2 ml/s at a speed of 5000 rev/min, the specific gravity of the fuel being 0.735, and  $Q_{\text{net},v} = 44\,200$  kJ/kg. Calculate the maximum bmep, the maximum bp, the minimum specific fuel consumption, and the maximum brake thermal efficiency at maximum torque, and compare this latter answer with the air standard efficiency.  
(14.44 bar; 178 kW; 0.303 kg/kW h; 26.9%; 63%)

- 13.5** A three-cylinder, direct-injection, water-cooled, two-stroke oil engine with two horizontally opposed pistons per cylinder has a bore of 82.6 mm and each piston has a stroke of 102 mm. The engine was tested against a brake with a torque arm of 0.381 m. The results taken on a variable speed test are as in Table 13.4. Plot curves of torque, bp, and specific fuel consumption against speed. Convert the torque curve to a bmep curve by calculation of the appropriate scale factor.

**Table 13.4** Data for Problem 13.5

| Speed<br>(rev/min) | Brake load<br>(N) | Fuel<br>(kg/min) |
|--------------------|-------------------|------------------|
| 1000               | 607.8             | 0.146            |
| 1100               | 614.6             | 0.157            |
| 1200               | 621.4             | 0.172            |
| 1300               | 621.4             | 0.185            |
| 1400               | 621.4             | 0.201            |
| 1500               | 621.4             | 0.216            |
| 1600               | 616.0             | 0.229            |
| 1700               | 609.0             | 0.241            |
| 1800               | 596.5             | 0.252            |

- 13.6** A four-cylinder, four-stroke diesel engine has a bore of 212 mm and a stroke of 292 mm. At full load at 720 rev/min the bmep is 5.93 bar and the specific fuel consumption is 0.226 kg/kW h. The air–fuel ratio as determined by exhaust gas analysis is 25/1. Calculate the brake thermal efficiency and the volumetric efficiency of the engine. Atmospheric conditions are 1.01 bar and 15 °C, and  $Q_{\text{net},v}$  for the fuel may be taken as 44 200 kJ/kg.  
(36%; 76.1%)

- 13.7** The engine of Problem 13.6 is to be used as a dual-fuel engine. It is to burn methane (calorific value 33 480 kJ/m<sup>3</sup> at 1.013 bar and 15 °C), and has a pilot injection of oil of 10% of the input when running as a diesel engine. The air–fuel ratio for the oil is 25/1 as before, and for the methane 8.5/1. If the volumetric efficiency and the power output remain the same, what is the brake thermal efficiency of the engine when running on the dual fuel?  
(23%)

- 13.8** A four-cylinder petrol engine with a bore of 63 mm and a stroke of 76 mm was tested at full throttle at 3000 rev/min over a range of mixture strengths. The following readings were taken during the test:

|                         |      |       |      |      |      |      |      |
|-------------------------|------|-------|------|------|------|------|------|
| Brake load/(N)          | 162  | 165.5 | 169  | 170  | 169  | 162  | 159  |
| Fuel consumption/(ml/s) | 2.08 | 2.04  | 2.17 | 2.50 | 2.84 | 3.40 | 3.56 |

- 13.4** A four-cylinder racing engine of capacity 2.495 litres has a bore of 94 mm and a compression ratio of 12/1. When tested against a dynamometer with a torque arm of 0.461 m a maximum load of 622 N was obtained at 5000 rev/min, and at the peak speed of 6750 rev/min the load was 547 N. The minimum fuel consumption was 17.2 ml/s at a speed of 5000 rev/min, the specific gravity of the fuel being 0.735, and  $Q_{\text{net},v} = 44\,200$  kJ/kg. Calculate the maximum bmep, the maximum bp, the minimum specific fuel consumption, and the maximum brake thermal efficiency at maximum torque, and compare this latter answer with the air standard efficiency.  
(14.44 bar; 178 kW; 0.303 kg/kW h; 26.9%; 63%)
- 13.5** A three-cylinder, direct-injection, water-cooled, two-stroke oil engine with two horizontally opposed pistons per cylinder has a bore of 82.6 mm and each piston has a stroke of 102 mm. The engine was tested against a brake with a torque arm of 0.381 m. The results taken on a variable speed test are as in Table 13.4. Plot curves of torque, bp, and specific fuel consumption against speed. Convert the torque curve to a bmep curve by calculation of the appropriate scale factor.

**Table 13.4** Data for Problem 13.5

| Speed     | Brake load | Fuel     |
|-----------|------------|----------|
| (rev/min) | (N)        | (kg/min) |
| 1000      | 607.8      | 0.146    |
| 1100      | 614.6      | 0.157    |
| 1200      | 621.4      | 0.172    |
| 1300      | 621.4      | 0.185    |
| 1400      | 621.4      | 0.201    |
| 1500      | 621.4      | 0.216    |
| 1600      | 616.0      | 0.229    |
| 1700      | 609.0      | 0.241    |
| 1800      | 596.5      | 0.252    |

- 13.6** A four-cylinder, four-stroke diesel engine has a bore of 212 mm and a stroke of 292 mm. At full load at 720 rev/min the bmep is 5.93 bar and the specific fuel consumption is 0.226 kg/kW h. The air–fuel ratio as determined by exhaust gas analysis is 25/1. Calculate the brake thermal efficiency and the volumetric efficiency of the engine. Atmospheric conditions are 1.01 bar and 15°C, and  $Q_{\text{net},v}$  for the fuel may be taken as 44 200 kJ/kg.  
(36%; 76.1%)
- 13.7** The engine of Problem 13.6 is to be used as a dual-fuel engine. It is to burn methane (calorific value 33 480 kJ/m<sup>3</sup> at 1.013 bar and 15°C), and has a pilot injection of oil of 10% of the input when running as a diesel engine. The air–fuel ratio for the oil is 25/1 as before, and for the methane 8.5/1. If the volumetric efficiency and the power output remain the same, what is the brake thermal efficiency of the engine when running on the dual fuel?  
(23%)
- 13.8** A four-cylinder petrol engine with a bore of 63 mm and a stroke of 76 mm was tested at full throttle at 3000 rev/min over a range of mixture strengths. The following readings were taken during the test:

|                         |      |       |      |      |      |      |      |
|-------------------------|------|-------|------|------|------|------|------|
| Brake load/(N)          | 162  | 165.5 | 169  | 170  | 169  | 162  | 159  |
| Fuel consumption/(ml/s) | 2.08 | 2.04  | 2.17 | 2.50 | 2.84 | 3.40 | 3.56 |

The relative density of the fuel is 0.724. Calculate the corresponding values of bmep and specific fuel consumption. Plot a consumption loop and obtain from it the corresponding values for maximum power and maximum economy. The bp in kilowatts is given by  $WN/26830$ , where  $W$  is the brake load in newtons and  $N$  is the engine speed in revolutions per minute.

(8.02 bar; 0.34 kg/kW h; 7.84 bar; 0.28 kg/kW h)

- 13.9** For the test outlined in Problem 13.8 an air box was fitted which had an orifice of 41.65 mm diameter with a discharge coefficient of 0.6. The corresponding manometer readings in millimetres of water were as follows:

|                      |       |       |       |       |       |       |       |
|----------------------|-------|-------|-------|-------|-------|-------|-------|
| Manometer/(mm water) | 33.50 | 33.50 | 33.50 | 33.80 | 33.80 | 34.25 | 34.80 |
|----------------------|-------|-------|-------|-------|-------|-------|-------|

Take the density of the air at inlet as  $1.215 \text{ kg/m}^3$ . Plot to a base of air-fuel ratio graphs of bp and specific fuel consumption. What are the air-fuel ratios at maximum power and maximum economy?

(12.8/1; 15.7/1)

- 13.10** A four-cylinder petrol engine has an output of 52 kW at 2000 rev/min. A Morse test is carried out and the brake torque readings are 177, 170, 168, and 174 N m respectively. For normal running at this speed the specific fuel consumption is 0.364 kg/kW h. The  $Q_{\text{net},v}$  of the fuel is 44 200 kJ/kg. Calculate the mechanical and brake thermal efficiencies of the engine.

(81.6%; 22.4%)

- 13.11** A V-8 four-stroke petrol engine is required to give 186.5 kW at 4400 rev/min. The brake thermal efficiency can be assumed to be 32% at the compression ratio of 9/1. The air-fuel ratio is 12/1 and the volumetric efficiency at this speed is 69%. If the stroke to bore ratio is 0.8, determine the engine displacement required and the dimensions of the bore and stroke. The  $Q_{\text{net},v}$  of the fuel is 44 200 kJ/kg, and the free air conditions are 1.013 bar and  $15^\circ\text{C}$ .

(5.1 l; 100.5 mm; 80.4 mm)

- 13.12** A four-cylinder, four-stroke diesel engine develops 83.5 kW at 1800 rev/min with a specific fuel consumption of 0.231 kg/kW h, and air-fuel ratio of 23/1. The analysis of the fuel is 87% carbon and 13% hydrogen, and the  $Q_{\text{net},v}$  is 43 500 kJ/kg. The jacket cooling water flows at 0.246 kg/s and its temperature rise is 50 K. The exhaust temperature is  $316^\circ\text{C}$ . Draw up an energy balance for the engine. Take  $R = 0.302 \text{ kJ/kg K}$  and  $c_p = 1.09 \text{ kJ/kg K}$  for the dry exhaust gas, and  $c_p = 1.86 \text{ kJ/kg K}$  for superheated steam. The temperature in the test house is  $17.8^\circ\text{C}$ , and the exhaust gas pressure is 1.013 bar.

(bhp 35.8%; cooling water 22.1%; exhaust 25.3%; radiation and unaccounted 16.9%)

- 13.13** An eight-cylinder, four-stroke diesel engine of 229 mm bore, 304 mm stroke, and compression ratio 14/1, has an output of 375 kW at 750 rev/min. The volumetric efficiency is 78%, the mechanical efficiency is 90%, and the air-fuel ratio is 25/1. If the imep for the pumping loop is 0.345 bar, calculate the imep for the working loop. The engine is now fitted with an exhaust-driven turbo-blower which delivers air to the cylinders at 1.43 bar. The compression ratio is reduced to 13/1 and the measured volumetric efficiency is 102%. It can be assumed that the exhaust pressure remains constant and equal to 1.013 bar, and that the imep of the main loop is directly proportional to the mass of air induced. Calculate the bp which can be expected if the speed and mechanical efficiency remain the same. Compare the specific fuel consumption for the two cases, if the air-fuel ratio for the supercharged engine is 26.8/1. The free air conditions are 1.013 bar and  $15^\circ\text{C}$ .

(7.0 bar; 514 kW; 0.23 kg/kW h; 0.205 kg/kW h)



- 13.14** A six-cylinder, four-stroke CI engine of 75 mm bore and 100 mm stroke has a brake power output of 110 kW at 3750 rev/min. The volumetric efficiency at this operating condition referred to ambient conditions of 1.013 bar and 20°C is 80%.
- The engine is now fitted with a mechanically driven supercharger which has an isentropic efficiency of 0.7 and a pressure ratio of 1.6. The supercharged version has a volumetric efficiency of 100% referred to the supercharger delivery pressure and temperature. If it is assumed that the indicated power developed per unit volume flow rate of induced air at ambient conditions is the same for normal aspiration and supercharging, calculate the net increase in brake power to be expected from the supercharged engine. Take the mechanical efficiency of the engine as 80% in both cases and the mechanical efficiency of the drive from engine to supercharger as 95%. (64.1 kW)

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