DIESEL ENGINE

REFERENCE BOOK

SECOND EDITION

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Part 1

Theory

1

The theory of compression ignition engines

Contents

1.1 Introduction

1.1.1 Historical

Although the history of the diesel engine extends back into the closing years of the 19th century when Dr Rudolf Diesel began his pioneering work on air blast injected stationary engines, and in spite of the dominant position it now holds in many applications, e.g. marine propulsion and land transport, both road and rail, it is today the subject of intensive development and capable of improvements. These will guarantee the diesel engine an assured place as the most efficient liquid fuel burning prime mover yet derived.

Before 1914, building on the work of Dr Rudolf Diesel in Germany and Hubert Akroyd Stuart in the UK, the diesel engine was used primarily in stationary and ship propulsion applications in the form of relatively low speed four-stroke normally aspirated engines.

The 1914–18 war gave considerable impetus to the development of the high speed diesel engine with its much higher specific output, with a view to extending its application to vehicles. Although the first generation of road transport engines were undoubtedly of the spark ignition variety, the somewhat later development of diesel engines operating on the self or compression ignition principle followed soon after so that by the mid 1930s the high speed normally aspirated diesel engine was firmly established as the most efficient prime mover for trucks and buses. At the same time with the increasing use of turbocharging it began to displace the highly inefficient steam engine in railway locomotives while the impending 1939–45 war gave a major impetus to the development of the highly supercharged diesel engine as a new aero engine, particularly in Germany.

Since the 1939–45 war every major industrial country has developed its own range of diesel engines. Its greatest market penetration has undoubtedly occurred in the field of heavy road transport where, at any rate in Europe, it is now dominant. It is particularly in this field where development, in the direction of turbocharging in its various forms, has been rapid during the last twenty years, and where much of the current research and development effort is concentrated. However, a continuous process of uprating and refinement has been applied in all its fields of application, from the very largest low speed marine two-stroke engines, through medium speed stationary engines to small single cylinder engines for operation in remote areas with minimum attendance. There is little doubt that it will continue to occupy a leading position in the spectrum of reciprocating prime movers, so long as fossil fuels continue to be available and, provided it can be made less sensitive to fuel quality, well into the era of synthetic or coal derived fuels.

1.1.2 Classifications

The major distinguishing characteristic of the diesel engine is, of course, the *compression-ignition* principle, i.e. the adoption of a special method of fuel preparation. Instead of relying on the passage of a spark at a predetermined point towards the end of the compression process to ignite a pre-mixed and wholly *gaseous* fuel–air mixture in approximately *stoichiometric* proportions as in the appropriately named category of *spark-ignition* (SI) engines, the *compression ignition* (CI) engine operates with a *heterogeneous* charge of previously compressed air and a finely divided spray of *liquid* fuel. The latter is injected into the engine cylinder towards the end of compression when, after a suitably intensive mixing process with the air already in the cylinder, the self ignition properties of the fuel cause combustion to be initiated from small nuclei. These spread rapidly so that complete combustion of all injected fuel, usually with air-fuel ratios well in excess of stoichiometric, is ensured. The mixing process is crucial to the operation of the Diesel engine and as such has received a great deal of attention which is reflected in a wide variety of *combustion systems* which may conveniently be grouped in two broad categories, viz.

(a) *Direct Injection (DI) Systems as used in DI engines,* in which the fuel is injected directly into a combustion chamber formed in the clylinder itself, i.e. between a suitably shaped non-stationary piston crown and a fixed cylinder head in which is mounted the fuel injector with its single or multiple spray orifices or nozzles. (See *Figures 1.1* and *1.2*.)

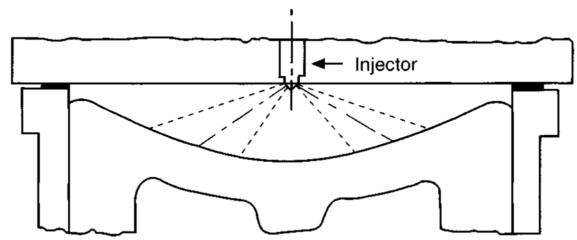

Figure 1.1 Quiescent combustion system. Application–Four-stroke and two-stroke engines mostly above 150 mm bore (*Benson and Whitehouse*)

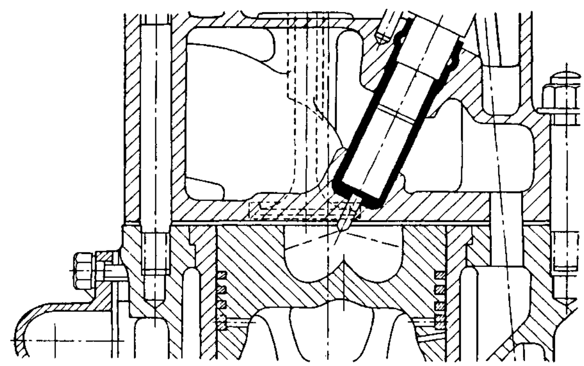

Figure 1.2 High swirl system. Application to virtually all truck and bus sized engines, but increasingly also to the high speed passenger car engine

(b) *Indirect Injection (IDI) Systems as used in IDI engines* in which fuel is injected into a prechamber which communicates with the cylinder through a narrow passage. The rapid transfer of air from the main cylinder into the prechamber towards top dead centre (TDC) of the firing stroke promotes a very high degree of air motion in the prechamber which is particularly conducive to rapid fuel–air mixing. (See *Figure 1.3*.)

Combustion systems are described in more detail in Chapter 4 and generally in Chapters 22 to 29 describing engine types. A further major subdivision of diesel engines is into *two-stroke* and *four-stroke* engines, according to the manner in which the gas exchange process is performed.

1.2 Two-stroke and four-stroke engines

An even more fundamental classification of diesel engines than that according to combustion system is into two-stroke or four-stroke engines, although this latter classification applies equally to spark ignition engines and characterizes the gas exchange process common to all air breathing reciprocating engines. The function of the gas exchange process, in both cases, is to effect

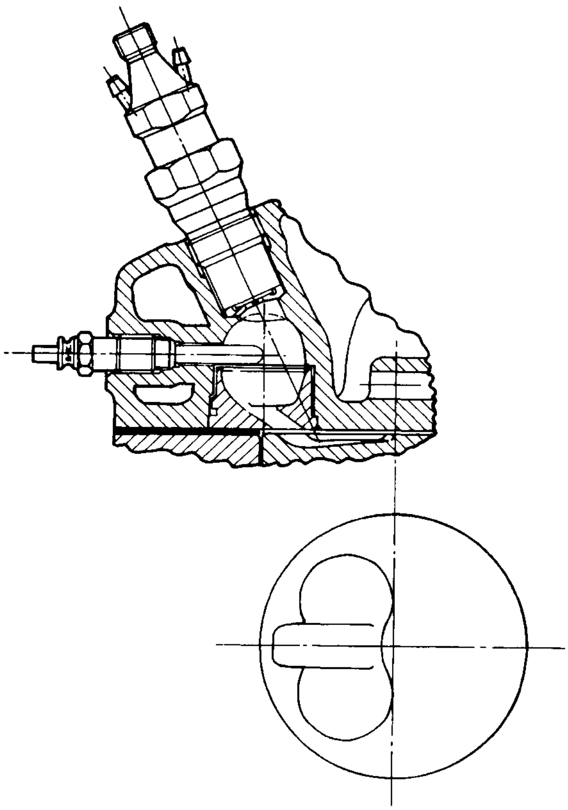

Figure 1.3 Prechamber system–compression swirl. Application–traditionally to high speed passenger car engines but now increasingly replaced by direct injection engine

expulsion of the products of combustion from the engine cylinder and their replacement by a fresh air charge in readiness for the next working cycle.

1.2.1 Two-stroke engines *(Figures 1.4a, b, c)*

In two-stroke engines combustion occurs in the region of top dead centre (TDC) of every *revolution.* Consequently gas exchange also has to be effected once per revolution in the region of bottom dead centre (BCD) and with minimum loss of expansion work of the cylinder gases following combustion.

This implies that escape of gas from the cylinder to exhaust and charging with fresh air from the inlet manifold must occur under the most favourable possible flow conditions over the shortest possible period. In practice the gas exhange or SCAVENGING process in two-stroke engines occupies between 100° and 150° of crank angle (CA) disposed approximately symmetrically about BDC.

Two-stroke engines may be subdivided according to the particular scavenging system used into the following sub-groups.

1.2.1.1 Loop scavenged engines (Figure 1.4a)

This is the simplest type of two-stroke engine in which both inlet and exhaust are controlled by ports in conjunction with a single piston. Inevitably this arrangement results in symmetrical timing which from the standpoint of scavenging is not ideal. In the first instance the 'loop' air motion in the cylinder is apt to produce a high degree of mixing of the incoming air with the products of combustion, instead of physical displacement through

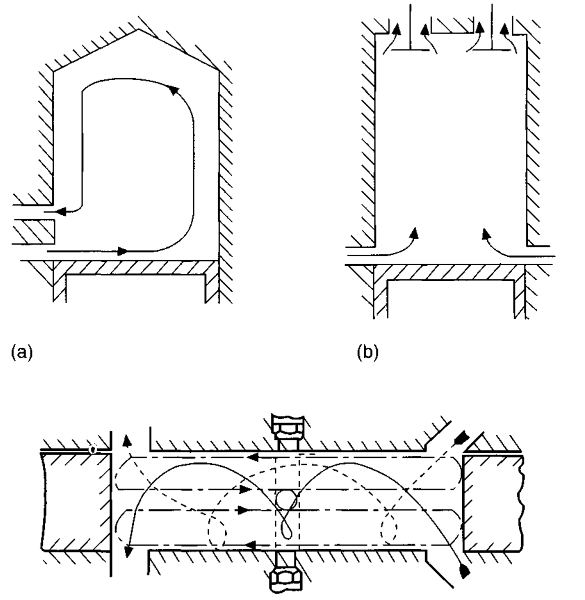

Figure 1.4 Two-stroke engines: (a) Loop scavenged engine; (b) Exhaust valve-in-head engine; (c) Opposed piston engine (*Benson and Whitehouse*)

the exhaust ports. As a result the degree of charge purity (i.e. the proportion of trapped air) at the end of the scavenging process tends to be low.

A second adverse feature resulting from symmetrical timing is loss of trapped charge between inlet and exhaust port closure and susceptibility to further pollution of the trapped charge with exhaust gas returned to the cylinder by exhaust manifold pressure wave effects. The great advantage of the system is its outstanding simplicity.

1.2.1.2 Uniflow scavenge single piston engines (Figure 1.4b)

In engines of this type admission of air to the cylinder is usually effected by piston controlled ports while the products of combustion are exhausted through a camshaft operated exhaust valve. Such systems are preferable from the standpoint of scavenging in that the 'uniflow' motion of the air from the inlet ports upwards through the cylinder tends to lead to physical displacement of, rather than mixing with, the products of combustion thus giving improved charge purity at the end of the scavenging process. At the same time it is now possible to adopt asymmetrical timing of the exhaust and inlet processes relative to bottom dead centre (BDC) so that, with exhaust closure preceding inlet closure the danger of escape of fresh charge into the exhaust manifold present in the loop scavenge system is completely eliminated. This system has been adopted in a number of stationary and marine two-stroke engines.

1.2.1.3 Uniflow scavenge opposed piston engines (Figure 1.4c)

In engines of this type admission of air is effected by 'air piston' controlled inlet ports, and rejection of products of combustion by 'exhaust piston' controlled exhaust ports. The motion of the two sets of pistons is controlled by either two crankshafts connected through gearing, or by a signle crankshaft with the 'top' bank of pistons transmitting their motion to the single crankshaft through a crosshead-siderod mechanism. By suitable offsetting of the cranks controlling the air and exhaust pistons asymmetrical timing can be achieved.

It is evident that this system displays the same favourable characteristics as the exhaust valve in head system, but at the expense of even greater mechanical complications. Its outstanding advantage is the high specific output per cylinder associated with two pistons. However, the system is now retained only in large low speed marine, and smaller medium speed stationary and marine engines. In high speed form it is still employed for naval purposes such as in some fast patrol vessels and mine searchers, although its use in road vehicles and locomotives is discontinued.

1.2.2 Four-stroke engines *(Figure 1.5)*

The vast majority of current diesel engines operate on the four-stroke principle in which combustion occurs only every other revolution, again in the region of top dead centre (TDC), and with the intermediate revolution and its associated piston strokes given over to the gas exchange process. In practice the exhaust valve(s) open well before bottom dead centre (BDC) following the expansion stroke and only close well after the following top dead centre (TDC) position is reached. The inlet valve(s) open before this latter TDC, giving a period of overlap between inlet valve opening (IVO) and exhaust valve closing (EVC) during which the comparatively small clearance volume is scavenged of most of the remaining products of combustion. Following completion of the inlet stroke, the inlet valve(s) close well after the following bottom dead centre (BDC), after which the 'closed' portion of the cycle, i.e. the sequence compression, combustion, expansion, leads to the next cycle, commencing again with exhaust valve opening (EVO).

The main advantages of the four-stroke cycle over its two-stroke counterpart are:

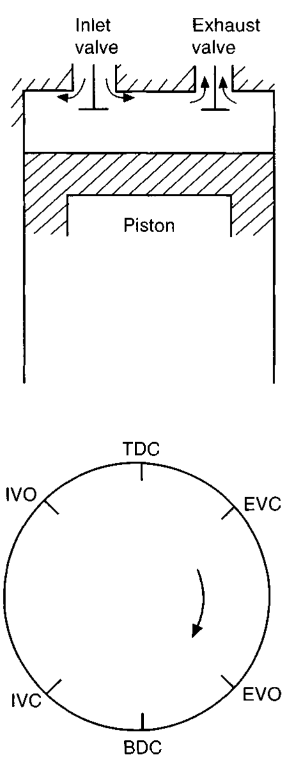

Figure 1.5 Four-stroke engine (turbocharged)

(a) the longer period available for the gas exhange process and the separation of the exhaust and inlet periods—apart from the comparatively short overlap—resulting in a purer trapped charge.
(b) the lower thermal loading associated with engines in which pistons, cylinder heads and liners are exposed to the most severe pressures and temperatures associated with combustion only every other revolution.
(c) Easier lubrication conditions for pistons, rings and liners due to the absence of ports, and the idle stroke renewing liner lubrication and giving inertia lift off to rings and small and large end bearings.

These factors make it possible for the four-stroke engine to achieve output levels of the order of 75% of equivalent two-stroke engines. In recent years attention has focused particularly on three-cylinder high speed passenger car two-stroke engines as a possible replacement for conventional four-cylinder, four-stroke engines with considerable potential savings in space and weight.

1.2.3 Evaluation of power output of two-stroke and four-stroke engines (*Figures 1.6a and b*)

In order to determine the power developed within the engine cylinder as a result of gas forces acting on the piston as opposed to shaft power from the output shaft, it is necessary to have a record of the variation of gas pressure (p) with stroke or cylinder volume (V) referred to as an Indicator Diagram (or *p-V Diagram*). This used to be obtained by mechanical means, but such crude instrumentation has now been completely replaced by electronic instruments known as pressure transducers. It is also generally more convenient to combine the pressure measurement with a crank angle (CA) measurement, using a position transducer in conjunction with a suitable crank angle marker disc, and subsequently convert crank angle to stroke values by a simple geometric transformation.

The sequence of events for the two cycles may be summarized as follows:

(a) *Two-stroke cycle* (asymmetrical timing)

1–2	compression	Closed Period	
2–3	heat release associated with combustion		
3–4	expansion		360°CA
4–5	blowdown	Open Period	
5–6	scavenging		
6–1	supercharge		

(b) *Four-stroke cycle*

1–2	compression	Closed Period	
2–3	heat release associated with combustion		
3–4	expansion		720°CA
4–5	blowdown	Open Period	
5–6	exhaust		
6–7	overlap		
7–8	induction		
8–1	recompression		

In both cases the cycle divides itself into the *closed period* during which power is being produced, and the *open* or *gas exchange* period which may make a small positive contribution to power production or, in the case of the four-stroke engine, under conditions of adverse pressure differences between inlet and exhaust manifold, a negative contribution. In the case of the four-stroke engine the area enclosed by the p-V diagram for the gas exchange process, i.e. 5–6–7–8, is known as the *pumping*

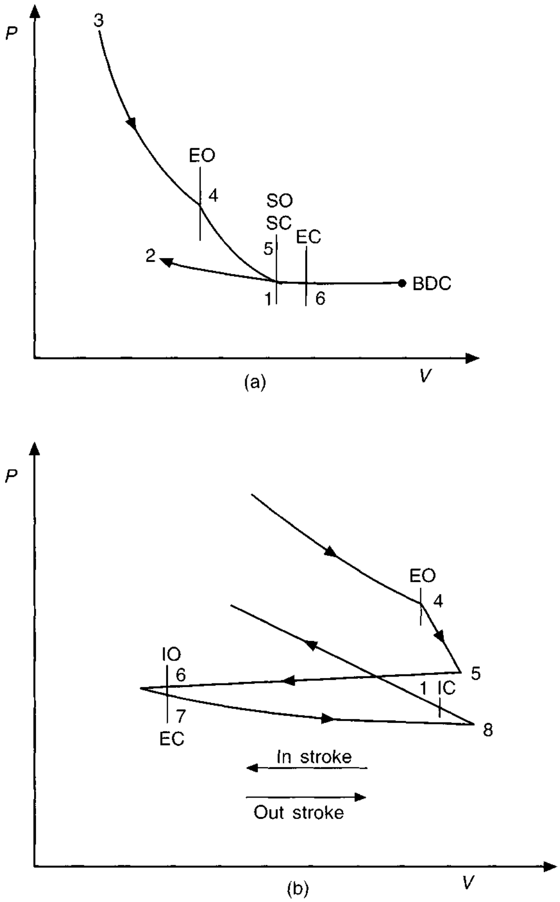

Figure 1.6 Gas exchange period. *p-V* diagrams: (a) Two-stroke (asymmetrical timing); (b) Four-stroke. (*Benson and Whitehouse*)

loop which may contribute positive or negative loop work to the work associated with the power loop. *Figures 1.6a and b* are typical p-V diagrams of the open or gas exchange period for two-stroke and four-stroke engines.

In both types of engine the cyclic integral expression leads to the so-called indicated *mean effective pressure*

$$p_{\mathrm{ind}} = \frac{\int p\,dV}{V_{\mathrm{swept}}} = \frac{\int dW}{V_{\mathrm{swept}}} \qquad (1.1)$$

where $\int dW$ represents the cyclic work with the distinction that the cycle occupies 360° for two-stroke and 720° for four-stroke engines.

The power may then be evaluated from the following expressions:

$$\dot{W}_{\text{two-stroke}}\,(\mathrm{kW}) = \frac{p_{\mathrm{ind}}\,(\mathrm{bar})\;V_{\mathrm{swept}}\,(\mathrm{m}^3)\,N_{\mathrm{e}}\,(\mathrm{rev/s})\,n}{10^{-2}} \qquad (1.2a)$$

or

$$\dot{W}_{\text{four-stroke}}\,(\mathrm{kW}) = \frac{p_{\mathrm{ind}}\,(\mathrm{bar})\;V_{\mathrm{swept}}\,(\mathrm{m}^3)\,N_{\mathrm{e}}\,(\mathrm{rev/s})\,n}{2\times 10^{-2}} \qquad (1.2b)$$

For purposes of comparison with air standard cycles (see section 1.3), it is appropriate to use the *effective* swept volume $(V_{\mathrm{swept}})_{\mathrm{eff}}$, i.e. that associated with the closed period only rather than the geometric swept volume V_{swept}. In the case of two-stroke engine, with the gas exchange period occupying up to 150°CA, $(V_{\mathrm{swept}})_{\mathrm{eff}}/V_{\mathrm{swept}}$ may be considerably less than unity while for four-stroke engines it varies between close to unity and 0.8 (approx.).

Similarly the volumetric compression ratio (CR), which again is crucial in air standard cycle calculations, is usually based on the effective swept volume

$$\text{i.e.}\quad (CR)_{\mathrm{eff}} = \frac{(V_{\mathrm{swept}})_{\mathrm{eff}} + V_{\mathrm{clearance}}}{V_{\mathrm{clearance}}} \qquad (1.3a)$$

rather than the geometric value

$$(CR)_{\mathrm{geom}} = \frac{V_{\mathrm{swept}} + V_{\mathrm{clearance}}}{V_{\mathrm{clearance}}} \qquad (1.3b)$$

Finally, indicated thermal efficiency η_{i} or indicated specific fuel consumption i.s.f.c. are evaluated from the expression

$$\eta_{\mathrm{i}} = \frac{\dot{W}(\text{eqn (1.2a) or (1.2b)})}{\dot{m}_{\mathrm{f}}\,(\mathrm{kg/sec})\;CV(\mathrm{kJ/kg})} \qquad (1.4a)$$

where $\dot{m}_{\mathrm{f}}$ is the rate of fuel flow to the engine and CV is the lower calorific value of the fuel and

$$\text{i.s.f.c.} = \frac{\dot{m}_{\mathrm{f}} \times 3600}{\dot{W}}\ \mathrm{kg/kW\ hr} \qquad (1.4b)$$

1.2.4 Other operating parameters

(a) *Air-fuel ratio*

The combustion process is governed in large measure by the air fuel ratio in the cylinder, expressed either in actual terms

$$\text{i.e.}\quad (A/F) = \frac{\dot{m}_{\mathrm{a_t}}\,(\mathrm{kg/sec})}{(\dot{m}_{\mathrm{f}})\ \mathrm{kg/sec}} \qquad (1.5a)$$

where $\dot{m}_{\mathrm{a_t}}$ is the rate of trapped airflow to the engine *or* relative to the chemically correct or stoichiometric air fuel ratio for the particular fuel, i.e. excess air factor

$$\varepsilon = \frac{(A/F)_{\mathrm{actual}}}{(A/F)_{\mathrm{stoichiometric}}} \qquad (1.5b)$$

In practice, for most hydrocarbon fuels

$$(A/F)_{\mathrm{stoichiometric}} \doteqdot 14.9 \qquad (1.5c)$$

and, depending on the combustion system used, the limiting relative air fuel ratio for smokefree combustion at full load is in the range

$1.2 < \varepsilon < 1.6$

being lower for IDI than for DI engines.

(b) *Gas exchange parameters*

For two-stroke engines, in particular, it is vitally important to make a distinction between the trapped rate of airflow $\dot{m}_{\mathrm{at}}$ and the total rate of airflow supplied to the engine $\dot{m}_{\mathrm{a}}$. This arises from the fact that the scavenging process in two-stroke engines is accompanied by substantial loss of air to exhaust, partly through mixing with products of combustion and partly through short-circuiting (see section 1.3.1) and leads to the definition of *trapping efficiency* as

$$\eta_{\mathrm{tr}} = \frac{(\dot{m}_{\mathrm{a}})_{\mathrm{t}}}{(\dot{m}_{\mathrm{a}})} \qquad (1.6a)$$

or its reciprocal, the scavenge ratio

$$R_{SC} = \frac{\dot{m}_a}{(\dot{m}_a)_t} \tag{1.6b}$$

In practice $1.1 < R_{SC} < 1.6$.

For four-stroke engines, particularly those with small valve overlap, e.g. in road traction, it is safe to assume that all the air delivered to the engine is trapped in the cylinder, i.e. $\eta_{tr} \doteqdot 1$. However, due to charge heating during the gas exchange process and adverse pressure conditions in the cylinder, it is likely that the volumetric efficiency η_{vol} defined as

$$\eta_{vol} = \frac{\text{volume of air trapped under inlet manifold conditions}}{\text{swept cylinder volume}}$$

$$= \dot{m}_a \frac{RT}{10^2\, p} \Big/ V_{swept} \tag{1.7}$$

(where T and p are respectively the inlet manifold temperature (°K) and pressure (bar)) is considerably less than unity. Clearly for the highest specific output, both the relative air fuel ratio ε and the volumetric efficiency should be as close to unity as possible.

1.3 Air standard cycles

It will be clear from the foregoing sections that the real processes in the diesel engine cylinder, particularly those of fuel preparation, combustion and gas exchange are extremely complex and require sophisticated computational techniques which are discussed in a number of specialist texts.[1,2,3]

Air standard cycles which are discussed in most elementary textbooks, provide a useful basis for comparing actual engine performance expressed in terms of indicated mean effective pressure (p_{ind}, eqn (1.1) and indicated thermal efficiency (η_i, eqn (1.4a) with corresponding values for highly idealized cycles, based on certain drastic simplifying assumptions as follows:

(a) the mass of working fluid remains constant throughout the cycle, i.e. gas exchange and fuel addition are ignored;
(b) the working fluid throughout the cycle is pure air treated as a perfect gas;
(c) the combustion and gas exchange processes are replaced by external heat transfer to or from the working fluid under idealized, e.g. constant volume or constant pressure conditions;
(d) compression and expansion processes are treated as adiabatic and reversible, i.e. heat transfer and friction effects are completely neglected;
(e) at any point of the working cycle, cylinder charge pressure and temperature are completely uniform, i.e. spatial variations in their values as for instance during combustion or scavenging, are completely neglected.

The most commonly used air-standard cycles are as follows (*Figures 1.7a, b and c*):

(a) *The constant pressure or diesel cycle (Figure 1.7a)*
Here combustion is simulated by constant pressure heat addition (2–3), and blowdown, followed by scavenge, by constant volume heat rejection 4–1. Compression 1–2 and expansion 3–4 follow the isentropic state relationships for a perfect gas. This particular cycle has, in the past, been used as a reference cycle for the 'classical' Diesel engine with air blast injection giving a rather long injection and hence heat release period, corresponding to 2–3. It has, however, little relevance to the modern diesel cycle.

(b) *The constant volume or Otto cycle (Figure 1.7b)*
Here combustion is simulated by constant volume heat release

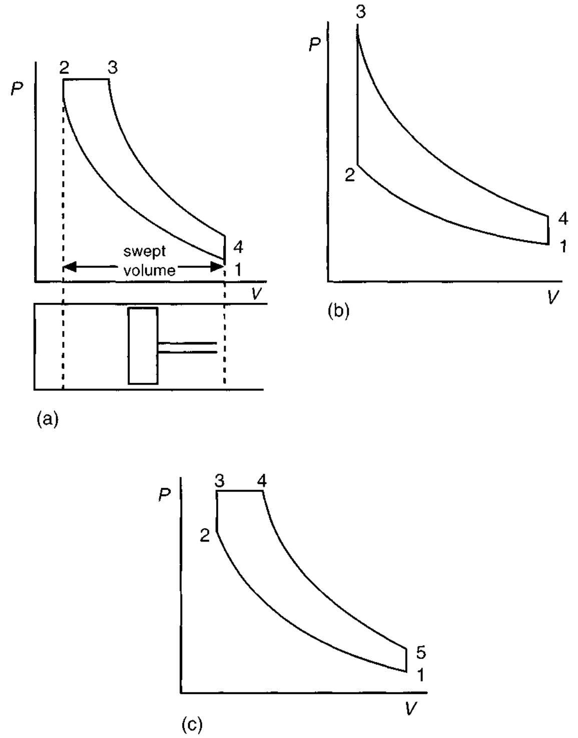

Figure 1.7 Air standard cycles: (a) Constant pressure cycle; (b) Constant volume cycle; (c) Dual combustion or composite cycle

2–3, and the blowdown-gas exchange sequence once again by constant volume heat rejection 4–1. Again compression 1–2 and expansion 3–4 are isentropic.

Traditionally this is the reference cycle for spark ignition (SI) engines, but it has distinct validity as a reference cycle for diesel engines, particularly under light load conditions when the heat release period is short so that the assumptions of zero heat release duration implied by the constant volume process 2–3 does not introduce excessive errors.

(c) *The 'dual combustion' or composite cycle (Figure 1.7c)*
This represents a combination of the constant pressure and constant volume cycles and is intended to provide a closer approximation to actual diesel cycles than either of the above ideal cycles. It is particularly appropriate where comparisons are to be made with actual diesel cycles on the basis of the maximum cylinder pressure p_{max} obtained during the heat release period, i.e. for engines operating in the mid-to full load range.

1.3.1 Theoretical expressions for air standard cycles

In the following derivations it will be assumed that the compression ratio CR corresponds to the effective compression ratio $(CR)_{eff}$ of the engine, eqn (1.3a), and that the isentropic index γ, i.e. the specifc heat ratio for air as a perfect gas, has the constant value $\gamma = 1.4$.

1.3.1.1 The constant pressure or diesel cycle (Figure 1.7a)

From basic engineering thermodynamics:

$$\text{compression work } W_{12} = \frac{+p_1 V_1 - p_2 V_2}{\gamma - 1} \tag{i}$$

(note this is negative)

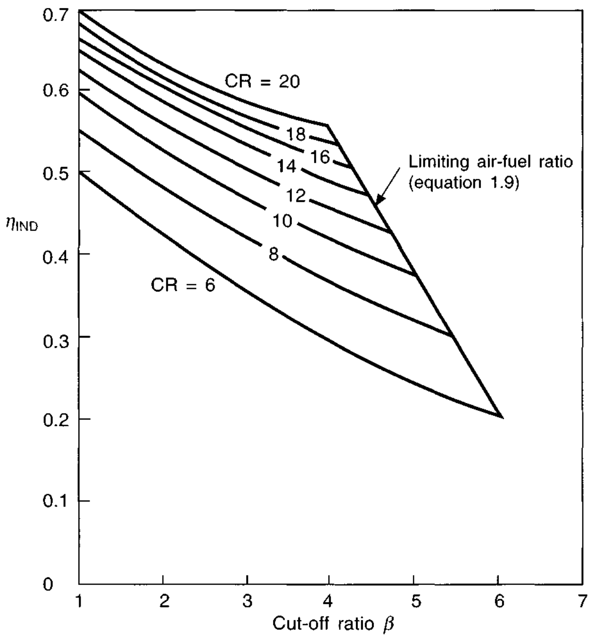

Figure 1.8 Constant pressure cycle. Indicated efficiency *vs* cut-off ratio (eqn 1.8)

constant pressure work

$$W_{23} = p_2\,(V_3 - V_2) = p_2V_2(\beta - 1) \qquad \text{(ii)}$$

where β = volume ratio $\dfrac{V_3}{V_2}$

constant pressure heat transfer

$$Q_{23} = mC_p(T_3 - T_2)$$

$$= \frac{p_2V_2}{RT_2}\left(\frac{\gamma}{\gamma-1}R\right)\left(\frac{V_3}{V_2} - 1\right)T_2$$

$$= p_2V_2\left(\frac{\gamma}{\gamma-1}\right)(\beta - 1) \qquad \text{(iii)}$$

$$\text{expansion work } W_{34} = \frac{p_3V_3 - p_4V_4}{\gamma-1} = \frac{p_2V_2\beta - p_4V_1}{\gamma-1} \qquad \text{(iv)}$$

$$\text{nett work } W_{\text{nett}} = W_{12} + W_{23} + W_{34}$$

$$= \frac{p_1V_1 - p_2V_2}{\gamma-1} + \frac{p_2V_2\beta - p_4V_1}{\gamma-1} + p_2V_2(\beta - 1) \qquad \text{(v)}$$

but

$$p_2 = p_3 = p_1\left(\frac{V_1}{V_2}\right)^{\gamma} = p_1(CR)^{\gamma},\ V_2 = \left(\frac{V_1}{CR}\right)$$

$$p_4 = p_3\left(\frac{V_3}{V_4}\right)^{\gamma} = p_2\left(\frac{\beta V_2}{V_1}\right)^{\gamma} = p_2\left(\frac{\beta}{CR}\right)^{\gamma} \qquad \text{(vi)}$$

Substituting from (vi) for p_2, V_2 and p_4 in (v) and (iii) and writing for the ideal efficiency of the constant pressure (CP) cycle:

$$(\eta_i)_{CP} = \frac{W_{\text{nett}}}{Q_{23}} \qquad \text{(vii)}$$

(vii) eventually reduces to

$$(\eta_i)_{CP} = 1 - \left(\frac{1}{(CR)^{\gamma-1}}\right)\frac{\beta^{\gamma} - 1}{\gamma(\beta - 1)} \qquad (1.8)$$

The volume ratio β is an indication of the air–fuel ratio A/F at which the engine is operating, since to a first approximation

$$Q_{23} = mC_p(T_3 - T_2) = p_2V_2\frac{\gamma}{\gamma-1}(\beta - 1)$$

$$= m_f(CV) \qquad \text{(viii)}$$

where m_f is the mass of fuel burnt

$$\text{But } m_{\text{air}} = m_1 = \frac{p_1V_1}{RT_1} \qquad \text{(ix)}$$

$$\text{whence } A/F = \frac{p_1V_1}{RT_1} \bigg/ \left[\frac{p_2V_2\dfrac{\gamma}{\gamma-1}(\beta - 1)}{(CV)}\right]$$

$$= \left(\frac{1}{CR}\right)^{\gamma-1}\frac{CV}{RT_1}\frac{1}{\dfrac{\gamma}{\gamma-1}(\beta - 1)} \qquad (1.9)$$

Assuming that the limiting air–fuel ratio is the stoichiometric ratio $(A/F)_{\text{stoich}}$ eqn (1.5c), it is possible to find a limiting value of the volume ratio β for any given compression ratio *CR* from eqn (1.9). This is shown in *Figure 1.8* indicating the behaviour of eqn (1.8) with different values of compression ratio *CR* and 'cut off' ratio β, including the position of the 'limiting line' for stoichiometric combustion.

Indicated efficiency $(\eta_i)_{CP}$ is seen to increase rapidly with volumetric compression ratio *CR* and to decrease with increasing values of the cut off ratio β, i.e. with decreasing air-fuel ratio, being a minimum, for any value of *CR* on the limit line, and a maximum for a cut off ratio $\beta = 1$.

Efficiency is not the only consideration appertaining to cycles. Specific output also has to be taken into account so that the relationship between indicated efficiency, specific output and compression ratio is equally important. The specific output is best measured in terms of the mean effective pressure defined by eqn (1.1) relative to the trapped pressure p_1. The calculation is as follows:

For any assumed value of the cut-off ratio β the equivalent air–fuel ratio A/F may be calculated from eqn (1.9), giving the heat input to the cycle as

$$Q_{\text{in}} = m\frac{CV}{A/F} = \frac{p_1V_1}{RT_1}\frac{CV}{A/F} \qquad \text{(i)}$$

With indicated efficiency $(\eta_i)_{CP}$ from eqn (1.8), the indicated work output of the cycle is given by

$$\int dW = Q_{\text{in}}(\eta_i)_{CP} \qquad \text{(ii)}$$

and the mean effective pressure, from eqn (1.1) becomes

$$p_{\text{ind}} = \frac{\int dW}{V_{\text{swept}}} = \frac{\int dW}{V_1\left(1 - \dfrac{1}{CR}\right)} = \frac{\dfrac{p_1V_1}{RT_1}\dfrac{CV}{A/F}(\eta_i)_{CP}}{V_1\left(1 - \dfrac{1}{CR}\right)}$$

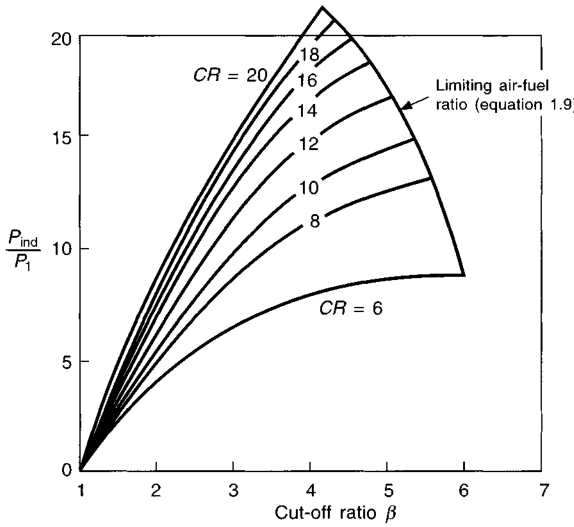

Figure 1.9 Constant pressure cycle. Indicated mean effective pressure *vs* cut-off ratio (eqn 1.10)

$$\text{or} \quad \left(\frac{p_{ind}}{p_1}\right)_{CP} = \frac{\dfrac{CV}{RT_1}\,\dfrac{1}{A/F}\,(\eta_i)_{CP}}{\left(1-\dfrac{1}{CR}\right)} \tag{1.10}$$

Equation (1.10) is represented by *Figure 1.9* and shows that, for any given compression ratio *CR*, efficiency decreases with increasing specific output, with a minimum value again on the limit line.

Figure 1.9 may be used both for naturally aspirated engines for which the trapped pressure p_1 is approximately equal to atmospheric pressure p_a as well as for supercharged engines with a supercharge (or boost) pressure ratio given approximately by $R_B = (p_1/p_a)$ (> 1).

1.3.1.2 The constant volume or Otto cycle (Figure 1.7b)

As already stated this cycle has only limited applicability to diesel engines, mainly under part load conditions. Heat transfer now occurs under constant volume conditions, both for the 'combustion' process 2–3 and the 'gas exchange' process 4–1. Nett cycle work

$$W_{nett} = W_{12} + W_{34}$$

$$= \frac{p_1V_1 - p_2V_2}{\gamma - 1} + \frac{p_3V_3 - p_4V_4}{\gamma - 1} \tag{i}$$

constant volume heat transfer

$$Q_{23} = mC_v(T_3 - T_2) = \frac{p_2V_2}{RT_2}\,\frac{R}{\gamma - 1}\left(\frac{p_3}{P_2} - 1\right)T_2$$

$$= \frac{p_2V_2}{\gamma - 1}(\alpha - 1) \tag{ii}$$

$$\text{where } \alpha = \frac{p_3}{p_2}$$

$$\text{But } p_2 = p_1(CR)^{\gamma},\ V_2 = \frac{V_1}{CR} = V_3$$

$$p_4 = p_3\left(\frac{1}{CR}\right)^{\gamma} = p_2\,\alpha\left(\frac{1}{CR}\right)^{\gamma} = p_1\,\alpha \tag{iii}$$

Substituting from (iii) in (i) and (ii) and writing for the ideal efficiency of the constant volume (CV) cycle

$$(\eta_i)_{CV} = \frac{W_{nett}}{Q_{23}} \tag{iv}$$

(iv) eventually reduces to

$$(\eta_i)_{CV} = 1 - \left(\frac{1}{CR}\right)^{\gamma - 1} \tag{1.11}$$

Equation (1.11) demonstrates that the efficiency of the constant volume cycle is a function of compression ratio *CR* only, and unlike the constant pressure cycle, independent of the level of heat addition, as expressed by the pressure ratio $p_3/p_2 = T_3/T_2 = \alpha$ (see *Figure 1.10*).

It is generally quoted in support of arguments to raise compression ratio in spark ignition (SI) engines.

1.3.1.3 The 'dual combustion' or composite cycle (Figure 1.7c)

As already stated, this cycle tends to approximate more closely to actual diesel cycles than either the pure constant pressure or constant volume cycles as described above. It lends itself particularly well to the representation of limited maximum cylinder pressure, as expressed by the pressure ratio $\alpha = p_3/p_2$ often specified in real diesel cycles, and to assessment of the effect of increased or retarded heat release, as expressed mainly by the volume ratio $\beta = V_4/V_3$.

The evaluation of cycle efficiency follows a similar pattern to that adopted above:

$$\text{nett cycle work} = W_{12} + W_{34} + W_{45}$$

$$= \frac{p_1V_1 - p_2V_2}{\gamma - 1} + p_3V_3(\beta - 1) + \frac{p_3V_4 - p_5V_5}{\gamma - 1} \tag{i}$$

Constant volume heat transfer

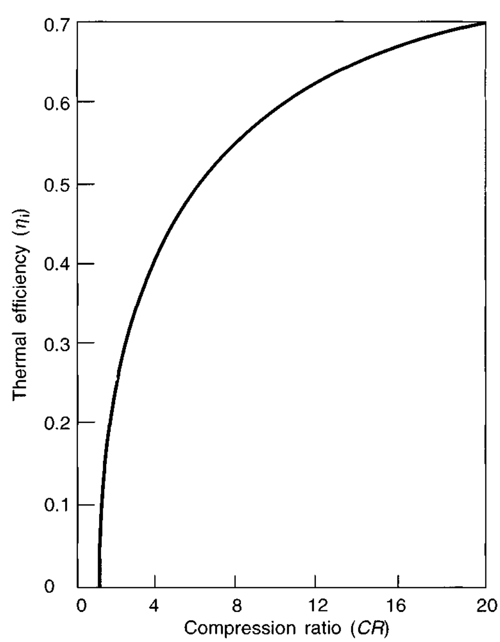

Figure 1.10 Constant volume cycle. Indicated efficiency *vs* compression ratio