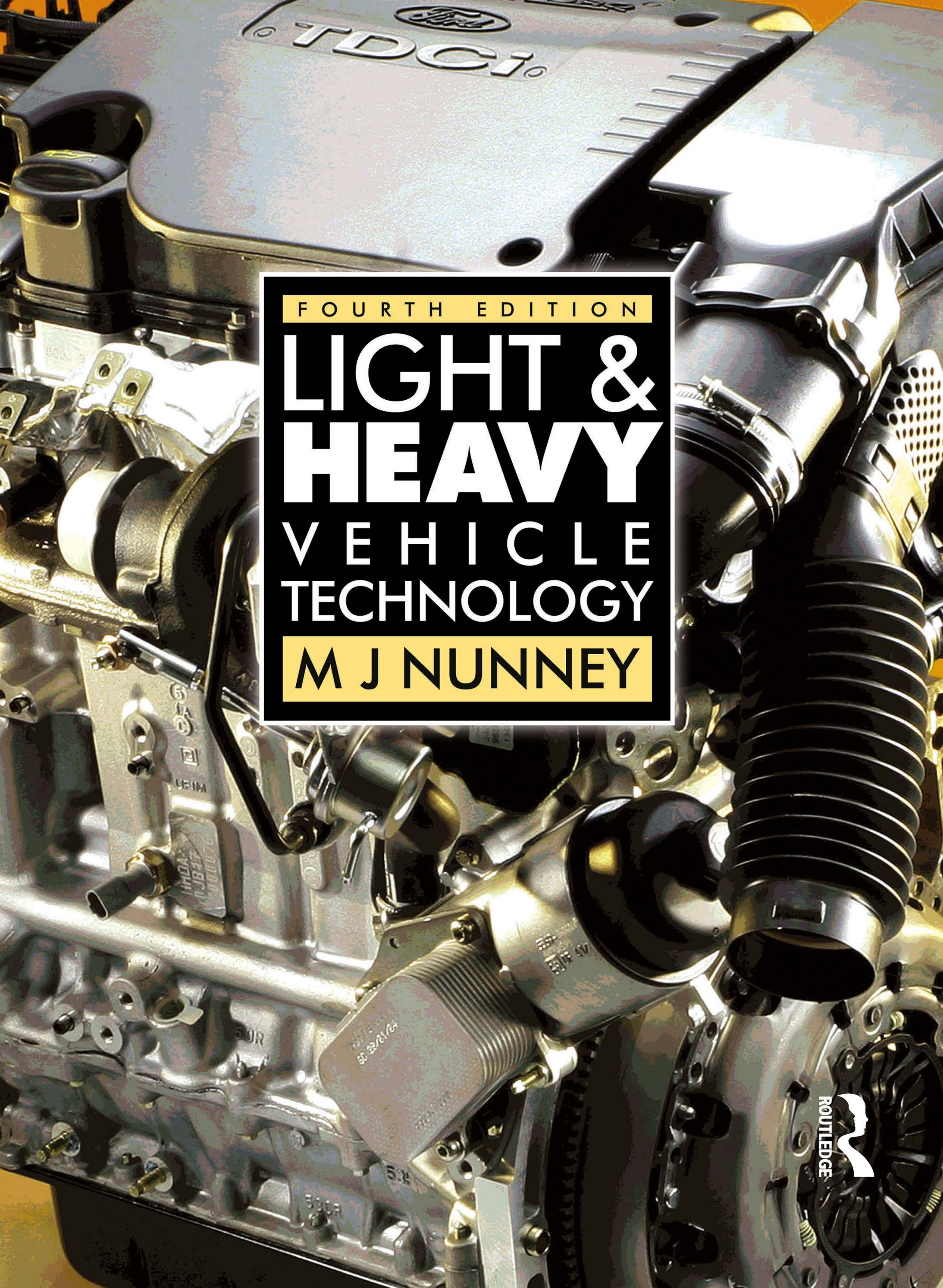

FOURTH EDITION

# LIGHT & HEAVY VEHICLE TECHNOLOGY

M J NUNNEY

ROUTLEDGE

# Light and Heavy Vehicle Technology

*When theory and practice do not agree, you should examine the facts to see what is wrong with the theory*

Charles F. Kettering, General Motors (1934)

# Light and Heavy Vehicle Technology

Fourth edition

M.J. Nunney

CGIA, MSAE, MIMI

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# Contents

# Preface

The purpose of this new fourth edition of *Light and Heavy Vehicle Technology* remains one of providing readily accessible information, which bridges the gap between the purely basic and the more advanced treatments of the subject. By understanding the reasons behind the design, construction and operation of the many and varied components of modern motor vehicles, the technician should be better equipped to deal with their servicing and overhaul. Some references to past automotive practice have been retained, not only because a technician may still be required to test and repair older vehicles, but also to provide a convenient transition to later practice.

Two entirely new sections of the book provide a topical introduction to alternative power sources and fuels, and battery-electric, hybrid and fuel-cell vehicles. Also, the number of entries in the list of automotive technical abbreviations has now increased to over 200. Finally, as in previous editions of the book, the tradition of including brief historical notes on the development of modern automotive concepts has been continued.

M.J. Nunney

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My thanks to the Rolls-Royce Heritage Trust for their permission to use the Dr Stanley Hooker quote on page 6 General note – the ever-increasing sophistication in the design and construction of modern passenger cars and commercial vehicles, makes it more than ever essential for service personnel to consult the vehicle manufacturer for up-to-date technical information and adjustments data in relation to a particular model, both in the interests of vehicle safety and customer satisfaction.

# Automotive technical abbreviations

| | |
|---|---|
| ABC | Active body control (Mercedes-Benz) |
| ABS | Anti-Blockier-System (German) – anti-lock braking system |
| ABDC | After bottom dead centre (engine timing) |
| AC | Alternating current |
| A/C | Air conditioning |
| ACL | Automatic chassis lubrication (commercial vehicles) |
| ACT | Air charge temperature |
| A/F | Air/fuel ratio |
| AIR | Air injection reactor (emission control) |
| ALB | Anti-lock brakes (Honda) |
| ARCS | Active roll control system (Citroen) |
| ASD | Automatic slip-control differential |
| ASF | Audi space frame (aluminium body construction) |
| ASR | Antriebs-Schlupf-Regelung (German) – anti-slip regulation or traction control |
| ATC | Automatic temperature control |
| ATDC | After top dead centre (engine timing) |
| ATF | Automatic transmission fluid |
| AWD | All-wheel drive (also 4WD) |
| AWS | All-wheel steering (also 4WS) |
| BAS | Brake assist system |
| BBDC | Before bottom dead centre (engine timing) |
| BDC | Bottom dead centre (engine timing) |
| BEV | Battery-electric vehicle |
| BHP | Brake horsepower |
| BMEP | Brake mean effective pressure |
| BOFT | Bearing oil film thickness |
| BSFC | Brake specific fuel consumption |
| BTDC | Before top dead centre (engine timing) |
| CAD | Computer aided design |
| CAFE | Corporate average fuel economy (American) |
| CAG | Computer aided gearshift (Scania) |
| CATS | Computer active technology suspension (Jaguar) |
| CB | Contact-breaker |
| CBE | Cab behind engine (commercial vehicles) |
| $C_d$ | Coefficient of drag (vehicle aerodynamics) |
| CD | Capacity discharge (ignition system) |
| CFC | Chlorofluorocarbon (refrigerant) |
| CGI | Compact graphite iron |
| CI | Compression ignition (diesel engines) |
| CN | Cetane number (diesel fuel ignition rating) |
| CNG | Compressed natural gas (fuels) |
| CO | Carbon monoxide (emission control) |
| $CO_2$ | Carbon dioxide (global warming) |
| COE | Cab over engine (commercial vehicles) |
| CP | Centre of pressure (vehicle aerodynamics) |
| CR | Compression ratio (engine) |
| CRS | Common rail system (diesel fuel injection) |
| CTX | Continuously variable transaxle (Ford) |
| CV | Constant velocity (universal joints) |
| CVT | Continuously variable transmission |
| $C_w$ | Coefficient of drag (German) – vehicle aerodynamics |
| $C_x$ | Coefficient of drag (French) – vehicle aerodynamics |
| DC | Direct current |
| DERV | Diesel engine road vehicles (fuel) |
| DI | Direct injection |
| DIS | Direct ignition system (no distributor) |
| DISI | Direct injection spark ignition |
| DOHC | Double overhead camshafts |
| DRP | Dynamic rear proportioning (brakes) |
| DSC | Dynamic stability control |
| DSG | Direct shift gearbox (Volkswagen group) |
| DWB | Double wishbone suspension |
| EBA | Emergency brake assist |
| EBFD | Electronic brake force distribution |
| EBS | Electronic braking system (air brakes) |
| ECI | Electronically controlled injection |
| ECM | Electronic control module |
| ECS | Evaporative control system (fuel system)<br>Electronically controlled suspension |
| ECT | Engine coolant temperature |
| ECU | Electronic control unit |
| EDC | Electronic diesel control |
| EFI | Electronic fuel injection |
| EGR | Exhaust gas recirculation (emission control) |
| ELV | End-of-life vehicle (materials recycling) |
| EMS | Engine management system |
| EP | Extreme pressure (lubricants) |
| EPAS | Electrical power-assisted steering (NSK-RHP) |
| EPHS | Electrically powered hydraulic steering (TRW) |
| EPS | Electric power steering |
| ESP | Electronic stability programme |
| ETC | Electronic traction control |
| ETS | Enhanced traction system (General Motors) |
| EUI | Electronic unit injector (Lucas Diesel) |
| EVC | Exhaust valve closed (engine timing) |
| EVO | Exhaust valve open (engine timing) |
| FCEV | Fuel-cell electric vehicle |
| FHP | Friction horsepower |
| FWD | Front-wheel drive |
| GCW | Gross combination weight (articulated vehicles) |
| GCWR | Gross combined weight rating (vehicle and trailer) |
| GDI | Gasoline direct injection (Mitsubishi) |
| GRP | Glass reinforced plastics |
| GTW | Gross train weight (drawbar vehicles) |
| GV | Governor valve (automatic transmissions) |
| GVW | Gross vehicle weight (rigid vehicles) |
| GVWR | Gross vehicle weight rating |
| GWP | Greenhouse warming potential (refrigerants) |
| HC | Hydrocarbons (emission control) |
| HDC | Hill descent control (ABS system) |
| HEV | Hybrid-electric vehicle |

| | |
|---|---|
| HFC | Hydrofluorocarbon (refrigerant) |
| HGV | Heavy goods vehicle |
| HT | High tension |
| HUCR | Highest useful compression ratio |
| HVAC | Heating, ventilation and air conditioning |
| IFS | Independent front suspension |
| IHP | Indicated horsepower |
| INJ | Injection (timing mark) |
| IOE | Inlet over exhaust (obsolete valve layout) |
| IPM | Integrated power module (hybrid electric vehicles) |
| IRS | Independent rear suspension |
| IVC | Inlet valve closed (valve timing) |
| IVO | Inlet valve open (valve timing) |
| KPI | King-pin inclination (steering) |
| LCV | Light commercial vehicle |
| LGV | Large goods vehicle |
| LI | Load index (tyres) |
| LNG | Liquefied natural gas (fuels) |
| LPG | Liquid petroleum gas (fuels) |
| LS | Leading shoe (drum brakes) |
| LSD | Limited slip differential |
| MAF | Mass air flow (engines) |
| MAP | Manifold absolute pressure |
| MOFT | Minimum oil film thickness |
| MON | Motor octane number (more demanding ON test) |
| MPI | Multi-point injection |
| MPV | Multi-purpose vehicle (people carrier) |
| NO | Nitrogen oxides (emission control) |
| NOAT | Nitrite organic acid technology (coolants) |
| NVH | Noise, vibration and harshness (vehicle refinement testing) |
| OAT | Organic acid technology (coolants) |
| OBD | On-board diagnosis |
| OD | Overdrive |
| ODP | Ozone depletion potential (refrigerants) |
| OHC | Overhead camshaft |
| OHV | Overhead valves |
| ON | Octane number (petrol anti-knock rating) |
| PAS | Power-assisted steering |
| PBD | Polybutadiene (tyres) |
| PCM | Power train control module (engine and transmission) |
| PCV | Positive crankcase ventilation (emission control) |
| PEM | Polymer electrolyte membrane (fuel cells) (or proton exchange membrane) |
| PFI | Port fuel injection (petrol engines) |
| PM | Particulate matter (diesel emission control) |
| PR | Ply-rating (tyres) |
| PSV | Public service vehicle |
| PTFE | Polytetrafluoroethylene |
| PTO | Power take-off (commercial vehicles) |
| PVC | Polyvinyl chloride |
| PZEV | Partial zero emission vehicle |
| RC | Roll-centre (suspension geometry) |
| RON | Research octane number (less demanding ON test) |
| RTV | Room temperature vulcanizing (sealant) |
| RWD | Rear-wheel drive |
| SAMT | Semi-automated mechanical transmission (Eaton) |
| SBC | Stand-by-control (electronic transmission control ZF) |
| SBR | Styrene-butadiene rubber (tyres) |
| SCA | Supplemental coolant additives |
| SCR | Selective catalytic reduction (emission control) |
| SCS | Stop control system (Girling) |
| SEFI | Sequential electronically controlled fuel injection (Ford) |
| SFC | Specific fuel consumption |
| SFI | Sequential fuel injection |
| SG | Spheroidal graphite (high-strength cast iron) |
| SI | Spark ignition (petrol engines) |
| SLA | Short and long arm (American) – suspension linkage |
| SOHC | Single overhead camshaft |
| SPI | Single point injection (petrol engines) |
| SRS | Supplemental restraint system (airbags) |
| SUV | Sports utility vehicle |
| SV | Side valves (obsolete valve layout) |
| TAC | Thermostatic air cleaner |
| TBI | Throttle body injection (SPI) |
| TC | Twin carburettors |
| TCI | Transistorized coil ignition |
| TCM | Transmission control module |
| TCS | Transmission controlled spark (engine intervention system) |
| TDC | Top dead centre (engine timing) |
| TDI | Turbocharged direct injection (diesel engines) |
| TEL | Tetra ethyl lead (petrol anti-knock additive) |
| TML | Tetra methyl lead (as above) |
| TPS | Throttle position sensor |
| TS | Trailing shoe (drum brakes) |
| TV | Throttle valve (engine and automatic transmissions) |
| TVS | Thermal vacuum switch (exhaust gas recirculation) |
| TWC | Three-way catalyst (emission control) |
| TXV | Thermostatic expansion valve (refrigeration) |
| UJ | Universal joint |
| ULEV | Ultra-low emission vehicle |
| VCP | Variable cam phasing (valve timing) |
| VCU | Viscous coupling unit (transmission) |
| VDC | Vehicle dynamics control (Bosch) |
| VGT | Variable-geometry turbocharger |
| VI | Viscosity index (lubricants) |
| VIP | Vehicle intrustion protection (Toyota) |
| VIVT | Variable inlet valve timing |
| VKPI | Virtual king-pin inclination (steering) |
| VSC | Vehicle skid control |
| VTG | Variable turbine geometry (turbocharging) |
| VTT | Variable twin turbo (turbocharging) |
| VVT | Variable valve timing |
| VVTL | Variable valve timing and lift |
| WOT | Wide-open throttle |
| ZEV | Zero emission vehicle |

# 1 The reciprocating piston petrol engine

## 1.1 MODERN REQUIREMENTS

### General background

The motor vehicle engine is basically a device for converting the internal energy stored in its fuel into mechanical energy. It is classified as an internal combustion engine by virtue of this energy conversion taking place within the engine cylinders.

Since the term 'energy' implies the capacity to perform work, the engine is thus able to propel the vehicle along the road and, within limits, overcome unwanted opposition to its motion arising from rolling friction, gradient resistance and air drag. To facilitate this process the engine is combined with a transmission system, the functioning of which is discussed later.

The vast majority of car engines are of the reciprocating piston type and utilize spark ignition to initiate the combustion process in the cylinders. However, the compression ignition or diesel principle to initiate combustion is increasingly challenging the petrol engine for car applications, especially in Europe. Both petrol and diesel engines operate on the four-stroke principle in which the piston travels one complete stroke for each of the successive events of induction, compression, combustion and exhaust.

The late Laurence Pomeroy, a distinguished motoring historian, once summarized the early history of the motor car as follows: From 1885 to 1895 men struggled to make the car go. From 1896 to 1905 they contrived to make it go properly. Between 1907 and 1915 they succeeded in making it go beautifully! What then are the requirements for the engine of the modern passenger car as reflected in many decades of further development and, not least, in the light of present-day energy conservation and environmental pollution considerations? These requirements can now add up to quite a formidable list. As we pursue our studies into the whys and wherefores of engine construction and operation, it will become evident that although some of the requirements are complementary, others are not, and therefore (as in most engineering) some compromise has generally to be accepted in the final product.

### Modern requirements

*Optimum performance*

With modern advances in engine design it is not particularly difficult to obtain sufficient power to give the car a high top speed, especially since the recent trend towards car bodies of lighter construction and more efficient aerodynamic shape. Today, however, a more important engine requirement than a further increase in top speed is an improved accelerating capability together with better flexibility in the low to middle speed range, or what is sometimes termed 'driveability'. A further performance requirement of a new engine design is that it must usually allow for possible future increases in cylinder size.

*Good fuel economy*

The overall aim of improving the fuel economy of cars is to minimize the amounts of crude oil used to provide petrol for their engines, because of constraints imposed by limited petroleum resources and rising costs. Fuel economy may also be made the subject of legislation, as it already is in America, where each manufacturer has to comply with corporate average fuel economy standards (or CAFE standards, as they are generally termed). For these reasons, further engine requirements are those of minimum weight so as to reduce total car weight; improved combustion efficiency, better to utilize the fuel; and reduced friction losses between the working parts.

*Low pollution*

Since the late 1960s increasingly stringent legislation has been applied to limit the levels of atmospheric pollutants emitted from car engines, especially the American FTP (Federal Test Procedure), the Japanese and later the European Community ECE/EEC test cycles, all of which differ in their requirements and are therefore not directly comparable. In Britain The Road Vehicles (Construction and Use) Regulations are also now such that there is a requirement for every motor vehicle to be so constructed that no avoidable smoke or visible vapour is emitted therefrom, and another that makes it an offence to use a vehicle which emits substances likely to cause damage to property or injury to persons. In general, legislation is concerned with carbon monoxide, which has toxic effects; unburned hydrocarbons, which contribute to atmospheric smog; and nitrogen oxides, which cause irritation to the eyes and lungs, and also combine with water to produce acid rain that destroys vegetation. To reduce these harmful emissions, not only is very careful control of the combustion process required in modern engine design, but also various sophisticated devices may have to be added for after-treatment of the exhaust gases. Of further concern to the environmentalist is the emission of carbon dioxide which, although non-toxic, is nevertheless an unwanted contributor to global warming. This has to the development of systems for deactivating half the number of cylinders on some large capacity V8 and V12 engines, to reduce fuel consumption and therefore the emission of carbon dioxide when full power is not required.

*Minimum noise level*

Noise is generally defined as unwanted sound. Reducing interior noise makes a car more attractive to the buyer. Reducing exterior noise to socially acceptable limits has been the subject of increasingly stringent legislation in the European Community and other countries since the early 1980s, and in

Britain is included in the Provision of the Motor Vehicles (Construction and Use) Regulations relating to noise. A similar function is performed in America by the EPA (Environmental Protection Agency) noise regulations. Since the engine is an obvious source of noise an important requirement is that its design and installation should minimize noise emission, not only that directly radiated from the engine itself to the exterior, but also that arising from vibrations transmitted through its mounting system to the car body interior.

*Easy cold starting*
An essential driver requirement of any engine, whether it be of past or present design, is that it should possess good cold starting behaviour and then continue to run without hesitation during the warming-up period. A present-day additional requirement is that the cold starting process should be accomplished with the least emission of polluting exhaust gases and detriment to fuel economy. To monitor the required enrichment of the air and fuel mixture for cold starting, increasingly sophisticated controls were applied first to carburettor automatic choke systems and then later to fuel injection cold start systems. These controls form part of what are now termed 'engine management systems'.

*Economic servicing*
An important owner requirement of a car is that its engine design should acknowledge the need to reduce servicing costs. This aim may be approached by minimizing the number of items that need periodic attention by a service engineer. For example, the use of hydraulic tappets eliminates altogether the need for adjustment of the valve clearances. It is also promoted by allowing ready access to those items of the engine involved in routine preventive maintenance, such as the drive-belt tensioner, spark plugs, and petrol and oil filters.

*Acceptable durability*
In order to reduce fuel consumption while still maintaining good car performance, it is now the trend to develop engines of smaller size with relatively higher power output. Furthermore, the installation of a turbo-charger permits an increase in power without imposing a corresponding increase in the size or weight of the engine itself. However, the greater heating effect on certain engine components may require changes to their material specifications and also the addition of an oil cooler. The components of modern engines have therefore tended to become more highly stressed, so that engine testing of ever-increasing severity by the manufacturers is now required to maintain durability in extremes of customer service.

*Least weight*
Another important design requirement of the modern petrol engine is that it should be made as light as possible. This is because a corresponding reduction in car weight can make significant improvements not only in fuel economy and acceleration capability, but also in general handling and ease of manoeuvring the car. Since reducing engine weight is not always consistent with maintaining durability, the need for adequate testing of the engine components is confirmed. Also special manufacturing techniques may have to be adopted to avoid damage to such items as castings with very thin walls.

*Compact size*
For the modern car, the manufacturer strives to provide the maximum interior space for the minimum possible exterior dimensions. Thus the trend is inevitably towards having the front wheels driven, with the power unit (engine and transmission) installed transversely between them; the conventional arrangement was to have a longitudinally mounted power unit from which the drive was taken to the rear wheels. It follows that the requirement now is for a more compact engine. This is because the engine length is controlled by the distance available between the steerable front wheels, less that required by the transaxle (combined gearbox and final drive); its width by the distance available between the radiator and the dash structure, less that required by the engine auxiliaries; and its height by the need for a low and sloping bonnet line, which contributes to an efficient aerodynamic body shape.

*Economic manufacture*
This is clearly a most important requirement for any new design of engine, since putting it into production demands a massive initial investment on the part of the car manufacturer. It is, of course, for this reason that the smaller specialist car manufacturer generally uses an existing engine from a volume producer. For economic manufacture a new design of engine should lend itself as far as possible to existing automatic production processes and require the minimum of special tooling. The cost of materials will be reduced in building a smaller engine, and the construction should be as simple as possible to minimize the number of parts to be assembled and thereby further reduce manufacturing costs. Similarly, to produce a range of large capacity V6 and V8 engines a modular design approach may be adopted, so that their major components can be produced on the same machinery.

*Aesthetic appearance*
In early years the under-bonnet appearance of high-grade cars of the 1920s and 1930s, such as Bugatti, Hispano-Suiza and Rolls-Royce, was much admired for the elegant proportions and beautiful finish of their engines. More recently manufacturers have recognized the customer appeal of a pleasing under-bonnet appearance. Not so much of the engine itself, which is usually buried deeper within the engine compartment, but of the neat arrangement and smooth contours of the modern comprehensive air intake system and its manifold runners that now lie above the engine.

## 1.2 ENGINE NOMENCLATURE

To understand the information given in an engine specification table, such as those included in a manufacturer's service manual or published in the motoring press, it is necessary to become familiar with some commonly used terms (Figure 1.1). The 'language' of the reciprocating piston engine is summarized in the following sections.

*Top dead centre*
The top dead centre (TDC) is of general application in engineering; it is any position of a hinged linkage in which three successive joints lie in a straight line. In the case of a motor

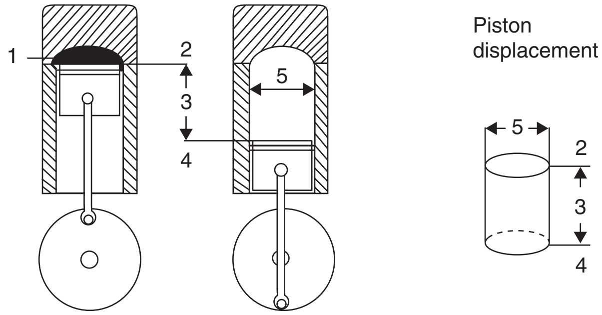


**Figure 1.1** Engine nomenclature (*Yamaha*)

1 volume of combustion chamber
2 top dead centre (TDC)
3 stroke
4 bottom dead centre (BDC)
5 bore

Piston displacement: volume of gases displaced by the piston as it moves from BDC to TDC.

vehicle engine, top dead centre refers to the position of the crankshaft when the piston has reached its closest point to the cylinder head. This results in the main, big-end and small-end bearings lying in a straight line. A motor vehicle service engineer often needs to establish top dead centre for checking the ignition and valve timing of an engine.

*Bottom dead centre*
The bottom dead centre (BDC) is, of course, the opposite extreme of crankshaft rotation when the piston has reached its furthest point from the cylinder head.

*Piston stroke*
In a general engineering sense, the stroke is the movement of a reciprocating component from one end of its travel to the other. In the motor vehicle engine the piston stroke, therefore, is the distance travelled by the piston in its movement from BDC to TDC or, of course, vice versa, and is expressed in millimetres (mm).

*Cylinder bore*
In engineering practice the term *bore* may refer to a hole through a bushing or pipe, or to the cutting of a large-diameter cylindrical hole, or to an actual measurement of the inside diameter of a hollow cylinder. It is the last named with which we are concerned here, where the bore refers to the inside diameter of the engine cylinder expressed in millimetres (mm).

*Piston displacement*
This term refers to the volume of cylinder displaced or swept by a single stroke of the piston, and is also referred to as *swept volume*. It is expressed in cubic centimetres ($cm^3$) and may be simply calculated as follows:

$$V_h = \frac{\pi d^2 s}{4000} \text{cm}^3$$

where $V_h$ is the piston displacement or swept volume ($cm^3$), $d$ is the cylinder bore (mm) and $s$ is the piston stroke (mm).

*Engine capacity*
Here we are referring to the total piston displacement or the swept volume of all cylinders. For example, if the swept volume of one cylinder of an engine is 375 $cm^3$ and the engine has four cylinders, then the engine capacity is 1500 $cm^3$ or 1.5 litres (1). This can be simply stated as:

$$V_H = V_h z$$

where $V_H$ is the engine capacity ($cm^3$), $V_h$ is the piston displacement ($cm^3$) and $z$ is the number of cylinders.

*Stroke/bore ratio*
Reference is sometimes made to the stroke/bore ratio of an engine and although the term itself is self-explanatory, its significance deserves further explanation because up to the 1950s the length of the piston stroke almost invariably exceeded the cylinder diameter, whereas until recently the converse situation has usually applied. The reason for the change from so-called *under-square* to *over-square* cylinder proportions from the 1950s onwards, was that for taxation and insurance purposes engines had previously been rated for horse-power by an RAC formula dating from the early years of motoring, which bore little relationship to the actual power they developed. Unfortunately it also meant that to keep the rated horse-power low for taxation purposes, but the actual power high, the engine designer was restricted to small bore and long stroke cylinder proportions. However, the introduction in 1947 of a flat tax on all passenger cars regardless of their engine size, relieved designers of this artificial restraint on cylinder bore size and led to improvements in engine performance.

The advantages claimed at the time for engines with larger bores and shorter strokes included increased size of valves and ports for better engine breathing; lower piston speeds that not only reduced mechanical loses due to friction and therefore improved fuel consumption, but also increased life expectancy of the cylinder bores; greater rigidity for the crankshaft by virtue of a smaller crankthrow (Section 1.8); and last but not least a reduction in engine height for a lower bonnet line. In current practice engines tend to be neither under-square nor over-square but to have a stroke/bore ratio closer to unity, so as better to achieve a compact combustion chamber and reduce harmful exhaust gas emission (Section 3.1).

*Mean effective pressure*
This term is used because the gas pressure in the cylinder varies from a maximum at the beginning of the power stroke to a minimum near its end. From this value must, of course, be subtracted the mean or average pressures that occur on the non-productive exhaust, induction and compression strokes. Engine mean effective pressure can be expressed in kilonewtons per square metre ($kN/m^2$).

*Indicated and brake power*
The most important factor about a motor vehicle engine is the rate at which it can do work or, in other words, the power it can develop. It is at this point that we must distinguish between the rate at which it might be expected to work (as calculated from the mean effective pressure in the cylinder, the piston displacement, the number of effective working

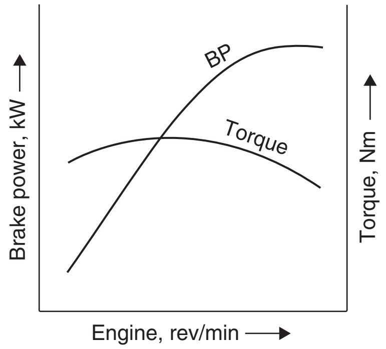


**Figure 1.2** Engine power and torque curves

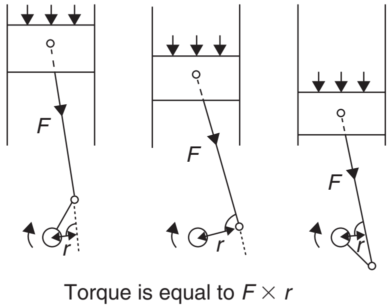


**Figure 1.3** Engine torque or turning effort

strokes in a given time and the number of cylinders) and the rate at which it actually does work (as measured in practice when the engine is running against a braking device known as a dynamometer).

The significance of this is that the *brake power* delivered at the crankshaft is always less than the *indicated power*, owing to internal friction losses in the engine. A simple expression for calculating in kilowatts (kW) the indicated power of an engine is as follows:

$$P = psAEz$$

where $P$ is the indicated power (kW), $p$ is the mean effective pressure (kN/m$^2$), $s$ is the piston stroke (m), $A$ is the piston area (m$^2$), $E$ is the number of effective working strokes per second and $z$ is the number of cylinders. (Since in the four-stroke cycle engine there is only one power stroke for every two complete revolutions of the crankshaft, the number of effective working strokes per second will correspond to one-half the number of engine revolutions per second.)

As mentioned earlier, a dynamometer is used in an engine testing laboratory to measure the brake power (or effective power) of an engine, because it acts as a brake to balance the torque or turning effort at the crankshaft through a range of speeds. A graph of the engine *power curve* can be drawn by plotting brake power values against engine speeds (Figure 1.2). Various standardized test procedures may be adopted in engine testing, such as those established by the American Society of Automotive Engineers (SAE), the German Deutsche Institut für Normung (DIN) and the Italian Commissione tecnica di Unificazione nell' Automobile (CUNA). In an engine specification table only the maximum brake power and corresponding crankshaft speed are usually quoted. For example, the 5.9 litre V twelve-cylinder 48-valve engine used in the high-performance Aston Martin DB9 Volante car is claimed to develop 330 kW at 6000 rev/min.

*Engine torque*

Also usually included on an engine performance graph is a *torque curve*, which is obtained by plotting crankshaft torque, or turning effort, against engine speed (Figure 1.2). The engine torque, of course, is derived from combustion pressure acting upon the cross-sectional area of the piston, the resulting force from which applies a turning effort to the crankshaft through the connecting rod and crankthrow arrangements (Figure 1.3). Engine torque, therefore, may be considered as the force of rotation acting about the crankshaft axis at any given instant.

Engine torque $T$ may be expressed in newton metres (Nm), and generally reaches a peak value at some speed below that at which maximum power is developed; the reason for this is explained at a later stage. An engine that provides good pulling power is typically one in which maximum torque is developed at moderate engine speeds. For example, the 1.4 litre four-cylinder 16-valve K Series engine of advanced construction (Figure 1.30), as originally designed by the Rover Group, developed its peak power of 70 kW at 6250 rev/min and a maximum torque of 124 Nm at 4000 rev/min. At the other extreme a heavy-vehicle diesel engine, such as the 11 litre six-cylinder turbocharged Scania with a peak power of 280 kW at a governed maximum speed of 1900 rev/min, develops a maximum torque of 1660 Nm at 1300 rev/min, which really is pulling power!

## 1.3 OPERATING PRINCIPLES

### The four-stroke petrol engine

As with the various repeating cycles of events in nature, so does the motor vehicle petrol engine need to operate on a constantly repeating cycle known as the four-stroke principle.

It would seem to be generally accepted that the first internal combustion engine to operate successfully on the four-stroke cycle was constructed in 1876 by Nicolaus August Otto (1829–91). This self-taught German engineer was to become one of the most brilliant researchers of his time and also a partner in the firm of Deutz near Cologne, which for many years was the largest manufacturer of internal combustion engines in the world.

Although the Otto engine ran on gas, which was then regarded as a convenient and reliable fuel to use, it nevertheless incorporated the essential ideas that led to the development in 1889 of the first successful liquid-fuelled motor vehicle engine. This was the twin-cylinder Daimler engine, patented and built by the German automotive pioneer Gottlieb Daimler (1834–1900) who, like Otto, had been connected with the Deutz firm. The Daimler engine was subsequently adopted by several other car manufacturers and, in most respects, it can be regarded as the true forerunner of the modern four-stroke petrol engine (Figure 1.4).

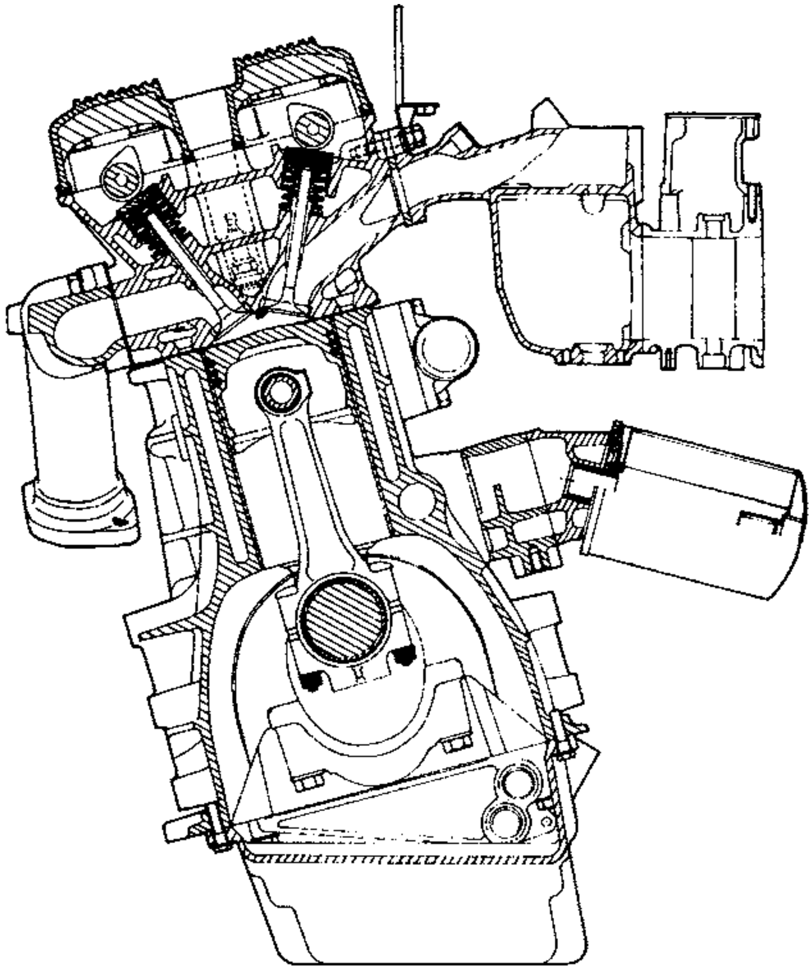

**Figure 1.4** Cross-section of a modern four-stroke petrol engine (*Jaguar*)

In this type of engine the following sequence of events is continuously repeated all the time it is running (Figure 1.5):

1 The *induction* stroke, during which the combustible charge of air and fuel is taken into the combustion chamber and cylinder, as a result of the partial vacuum or depression created by the retreating piston.
2 The *compression* stroke, which serves to raise both the pressure and temperature of the combustible charge as it is compressed into the lesser volume of the combustion chamber by the advancing piston.
3 The *power* stroke, immediately preceding which the combustible charge is ignited by the sparking plug and during which the gases expand and perform useful work on the retreating piston.
4 The *exhaust* stroke, during which the products of combustion are purged from the cylinder and combustion chamber by the advancing piston, and discharged into the exhaust system.

It thus follows that one complete cycle of operations occupies two complete revolutions of the engine crankshaft. Since energy is necessarily required to perform the initial induction and compression strokes of the engine piston before firing occurs, an electrical starter motor is used for preliminary

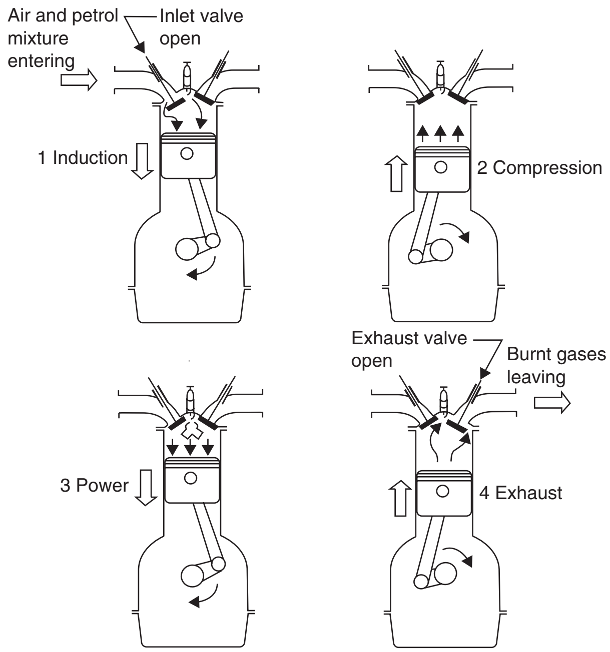


**Figure 1.5** The four-stroke petrol engine cycle

cranking of the engine. Once the engine is running the energy required for performing subsequent induction, compression and exhaust strokes is derived from the crankshaft and flywheel system, by virtue of its kinetic energy of rotation. *Kinetic energy* is a term used to express the energy possessed by a body due to its mass and motion. The principle of an engine flywheel is therefore to act as a storage reservoir for rotational kinetic energy, so that it absorbs energy upon being speeded up, and delivers it when slowed down.

In the four-stroke cycle, the functions of admitting the combustible charge before its compression, and releasing the burnt gases after their expansion, are performed by the engine inlet and exhaust valves. The opening and closing of the inlet and exhaust valves are not, in actual practice, timed to coincide exactly with the beginning and ending of the induction and exhaust strokes; nor is the spark timed to occur exactly at the beginning of the power stroke. At a later stage the reasons for these departures in *valve and ignition timing* from the basic four-stroke operating cycle will be made clear.

## The two-stroke petrol engine

As its designation implies, the two-stroke petrol engine (Figure 1.6) completes its working cycle in only two strokes of the piston, so that a combustible charge is ignited at each revolution of the crankshaft. Although in its simplest construction the two-stroke petrol engine needs no valves, the induction and exhaust process must be facilitated by a system of *scavenging* or forcible clearing of the cylinder gases. This may either take the form of a separate engine-driven pump, or utilize the motion of the engine piston itself in a sealed crankcase. The flow of gases entering and leaving the cylinder is controlled by the reciprocating movement of the engine piston, which thus acts as a slide valve in conjunction with ports cut in the cylinder wall. Although the two-stroke petrol engine was once favoured in Europe for some small inexpensive passenger cars, it generally became obsolescent because of the difficulty in reducing its harmful exhaust emissions.

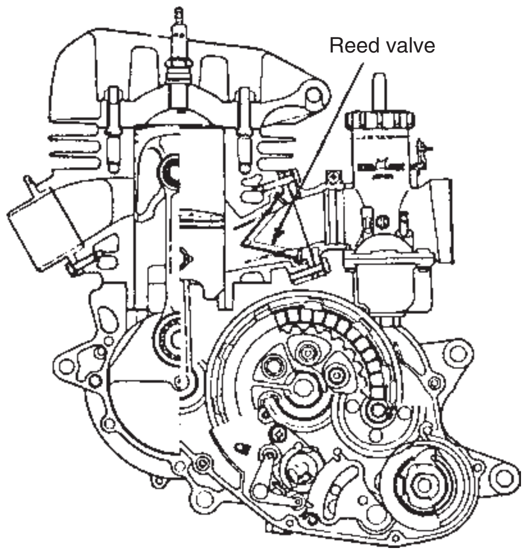


**Figure 1.6** Cross-section of a two-stroke petrol engine (*Honda*)

At the 11th Sir Henry Royce Memorial Lecture in 1966, Cyril Lovesey recalled a remark by another distinguished Rolls-Royce aero engine specialist, Dr Stanley Hooker, who amusingly described a four-stroke engine as one with ‘one stroke to produce power and three strokes to wear it out’. It is therefore perhaps not surprising that from time to time attempts are made to revive interest in the two-stroke petrol engine for automotive use, albeit in much more sophisticated forms of which an example will be later described.

It may be of interest to recall that the two-stroke and the four-stroke engine both originated in the late 1870s, so it might reasonably be assumed that both types of engine started out in life with an equal chance of success. The fact that the four-stroke engine became by far the more widely adopted type can probably be explained by its having a greater potential for further development. This is a criterion that can often be applied to rival ideas in all branches of engineering.

The first successful application of the two-stroke cycle of operation to an early gas engine is generally attributed to a Scottish mechanical engineer, Sir Dugald Clerk (1854–1932). It is for this reason, of course, that the two-stroke cycle is sometimes referred to as the *Clerk cycle*. Dugald Clerk, like several other pioneer researchers of the internal combustion engine, was later to achieve high academic distinction, culminating in his election as Fellow of the Royal Society in 1908.

The Clerk engine was scavenged by a separate pumping cylinder. A few early motor vehicle two-stroke petrol engines followed the same principle, but it later became established practice to utilize the underside of the piston in conjunction with a sealed crankchamber to form the scavenge pump. This idea was patented in 1889 by Joseph Day & Son of Bath, England and represented the simplest type of two-cycle engine.

In the two-stroke or Clerk cycle, as applied by Day, the following sequence of events is continuously repeated all the time the engine is running (Figure 1.7):

1 The *induction-compression* stroke. A fresh charge of air and fuel is taken into the crankchamber as a result of the depression created below the piston as it advances towards the cylinder head. At the same time, final compression of the charge transferred earlier in the stroke from the crankchamber to the cylinder takes place above the advancing piston.
2 The *power-exhaust* stroke. The combustible charge in the cylinder is ignited immediately preceding the power stroke,

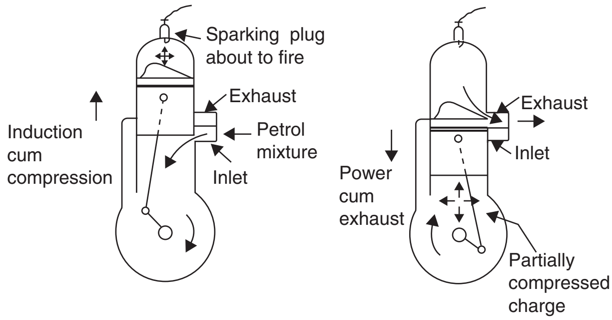


**Figure 1.7** The two-stroke petrol engine cycle

during which the gases expand and perform useful work on the retreating piston. At the same time, the previously induced charge trapped beneath the retreating piston is partially compressed. Towards the end of the stroke, the exhaust gases are evacuated from the cylinder, a process that is facilitated by the scavenging action of the new charge transferred from the crankcase.

The uncovering and covering of the cylinder ports of the piston, or *port timing*, is determined by considerations similar to those affecting the valve timing of the four-stroke engine and will explained at a later stage.

### Four-stroke versus two-stroke engines

The following generalizations may be made concerning the relative merits of two-stroke and four-stroke petrol engines in their basic form:

1. The two-stroke engine performs twice as many power strokes per cylinder per revolution. In theory at least, this might be expected to produce twice the performance of a four-stroke engine of equivalent size. Unfortunately, this is not realized in practice because of the difficulties encountered in effectively purging the exhaust gases from the cylinder and then filling it completely with a fresh combustible charge. The scavenging efficiency of the basic two-stroke petrol engine is therefore poor.
2. In performing twice as many power strokes per revolution, the two-stroke engine can deliver a smoother flow of power, but this may be less true at low engine speeds when irregular firing or 'four-stroking' can result from poor scavenging.
3. An obvious practical advantage of the basic two-stroke engine is the mechanical simplicity conferred by its valveless construction, which contributes to a more compact and lighter engine that should be less expensive to make.
4. Reduced maintenance requirements might reasonably be expected with the basic two-stroke engine by virtue of point 3. There is, however, the well-known tendency for carbon formation to have a blocking effect on the exhaust ports, which impairs engine performance by reducing scavenging efficiency.
5. The fuel consumption of the basic two-stroke engine is adversely affected by the poor cylinder scavenging, which allows part of the fresh charge of air and fuel to escape through the exhaust port before final compression of the charge takes place.
6. There is a greater danger of overheating and piston seizure with a two-stroke engine, which can set a limit on the maximum usable performance. It is more difficult to cool satisfactorily, because it does not have the benefit of the second revolution in the four-stroke cycle when no heat is being generated.
7. Lubrication of the two-stroke petrol engine is complicated by the need to introduce oil into the fuel supply to constitute what is generally termed a *petroil* mixture. The working parts of the engine are thus lubricated in aerosol fashion by oil in the air and fuel charge, and this tends to increase harmful exhaust emissions. It is for this reason that the basic two-stroke petrol engine is now obsolescent for cars.

*Scavenging: further details*

Frequent reference has been made to the inherently poor scavenging efficiency of the basic two-stroke petrol engine. The word 'basic' has been used deliberately and is intended to apply to the Day type of early two-stroke engine, which had a deflector-head piston to promote a cross-scavenging effect on the burnt charge leaving the cylinder (Figure 1.8a). This not entirely successful scheme persisted until the mid 1920s, when Dr E. Schnürle of Germany developed an alternative loop-scavenging system. In this the deflector on the piston head is omitted and two transfer ports with angled passages are disposed on either side of, instead of opposite, the exhaust port (Figure 1.8b). The loop-scavenge effect produced is such that before the two streams of fresh charge intermingle, they converge upon the cylinder wall at a point furthest away from the exhaust port, so there is less chance of escape.

The Day type of early two-stroke engine also used what would now be classified as a three-port system of scavenging. This system comprises inlet, transfer and exhaust ports, all in the cylinder wall, and necessarily imposes a restriction on the period during which a fresh charge of air and fuel may enter the crankcase.

To achieve more complete filling of the crankcase, the later two-port system of scavenging is now generally employed. In this system only the transfer and exhaust ports are in the cylinder wall, the inlet port being situated in the crankcase itself and controlled by either an automatic flexible reed valve (Figure 1.9a) or an engine-driven rotary disc valve (Figure 1.9b) which improve the torque and power characteristics respectively of an engine. The two-port system of scavenging thus allows the fresh charge to continue entering the crankcase

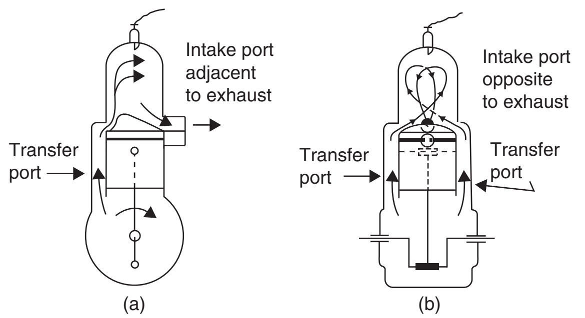


**Figure 1.8** Early three-port scavenging systems: (a) cross-scavenging (b) loop-scavenging

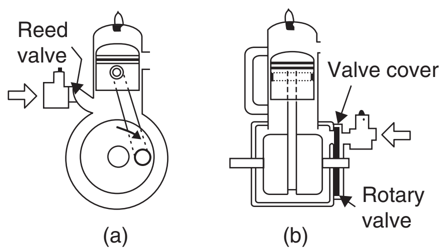


**Figure 1.9** Later two-port scavenging systems: (a) reed valve (b) rotary valve (*Yamaha*)

during the whole, instead of part, of the induction-compression stroke, albeit with a little extra mechanical complication.

### Further development of the two-stroke petrol engine

We take as an example the advanced concept of the Toyota S-2 (supercharged two-stroke) engine (Figure 1.10). The development objective of this engine is to deliver greater power more smoothly than a conventional one, and also to confer very high torque at low engine speeds. A Roots-type supercharger or blower (Section 9.2) is used to achieve positive scavenging of the exhaust gases. Unlike the two-stroke petrol engines so far described, the Toyota engine borrows many features from established four-stroke practice. It has a four-valve cylinder head with two valves for intake and two for exhaust, which do of course open and close twice as often in this two-stroke application. The engine is provided with electronically controlled ignition and fuel injection systems, with fuel being injected directly into the cylinders.

### The importance of engine compression

Early internal combustion engines were very inefficient because they were provided with a combustible charge that was ignited at atmospheric pressure. However, it was recognized as early as 1838 by William Barnett in the UK that compression of the charge before combustion was advantageous. Nearly 25 years later, a French railway engineer with the splendid name of Alphonse Beau de Rochas was granted a patent in respect of several ideas that related to the practical and economical operation of the internal combustion engine. Among these ideas he stated a requirement for the maximum possible expansion of the cylinder gases during the power stroke, since the cooler they become the more of their energy is transformed into useful work on the retreating piston. To assist in this aim, there was a further requirement for the maximum possible pressure at the beginning of the expansion process, which the motor vehicle service engineer recognizes as the *compression pressure* of an engine. The first successful engine to utilize this principle was, as mentioned earlier, the Otto engine.

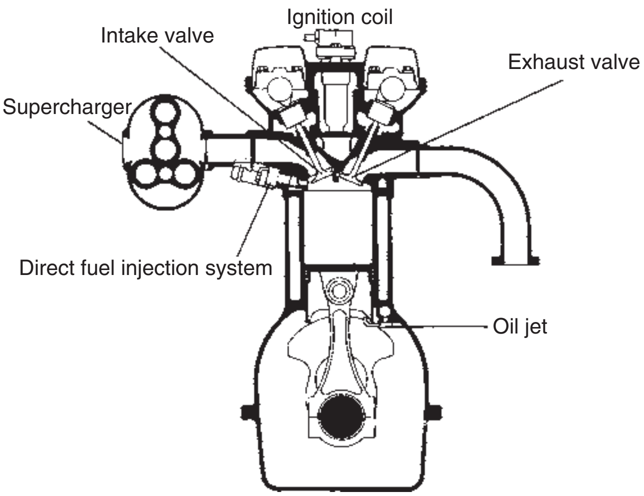


**Figure 1.10** Toyota S2 supercharged two-stroke engine

Within certain limitations, which will be better understood at a later stage, a high-compression engine is relatively more efficient than a low-compression one in terms of either improved fuel economy or greater power. This is simply because it better utilizes the internal energy received from its fuel or, in other words, it possesses a higher *thermal efficiency*. It is this particular feature of engine operation that explains the importance of checking compression pressures before attempting to tune an engine in service.

### Compression ratio

The extent to which the air and fuel charge in a petrol engine is compressed, prior to the power stroke, is known as the *compression ratio* of an engine. It is calculated as the ratio of the total volume enclosed above the piston at BDC to the volume remaining above the piston at TDC.

The swept volume has already been explained in Section 1.2. A complementary term is *clearance volume*, which is that volume remaining above the piston when it reaches TDC. In some engines the combustion chamber is formed mainly in the piston head (Section 3.3), so that the clearance volume is concentrated within, rather than above, the piston. Hence, the compression ratio (usually abbreviated to CR) may also be expressed as the swept volume plus the clearance volume divided by the clearance volume (Figure 1.11), as follows:

$$\varepsilon = \frac{V_h + V_c}{V_c}$$

where $\varepsilon$ is the compression ratio, $V_h$ is the cylinder swept volume (cm$^3$), and $V_c$ is the combustion space clearance volume (cm$^3$).

The calculated compression ratios for petrol engines are typically in the range 8:1 to 9.5:1. It should be appreciated, however, that in the petrol engine the calculated compression ratio is realized in practice only when the engine is running with a wide open throttle.

Furthermore, in the case of a two-stroke engine $V_h$ is generally considered as the cylinder volume from the point of exhaust port closing to TDC, which is therefore less than the

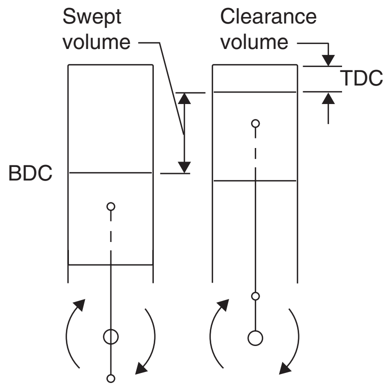


**Figure 1.11** Engine compression ratio

swept volume of an equivalent four-stroke engine and gives a lower compression ratio.

## 1.4 BASIC STRUCTURE AND MECHANISM

It is convenient to introduce the various components of the petrol engine in the following groups:

Cylinder block and crankcase
Piston and rings
Connecting rod and bearings
Crankshaft assembly and bearings
Cylinder head and gasket
Valve train and timing drive
Engine support mountings.

### Cylinder block and crankcase

In combination, the cylinder block and crankcase form the main structural component of the engine and perform several important functions, as follows:

1 Each cylinder must act not only as a pressure vessel in which the process of combustion can take place, but also as a guide and bearing surface for the piston sliding within it.
2 Since the engine cylinders have to be cooled effectively, the cylinder block must also form a jacket to contain the liquid coolant.
3 The crankcase provides an enclosure for the crankshaft and various other parts of the engine mechanism and must preserve accurate alignment of their supporting bearings under all operating conditions.
4 Pressure conduits in the form of either drilled or cast in ducts must also be incorporated in the crankcase to convey oil to the engine working parts.
5 The cylinder block is required to provide external mounting surfaces for various engine auxiliary units.

It has long since been established practice to combine the cylinder block and crankcase into a single unit, this generally being termed *monobloc* construction (Figure 1.12). The historical origins of this form of construction date back to the early 1920s, when there was a general trend towards simplification of the engine structure. Until that time the cylinder block and crankcase were produced as separate units, in cast iron and aluminium alloy respectively, and then bolted together.

In relation to modern engine design, the monobloc construction provides a very necessary rigid foundation for the engine and reduces manufacturing costs. It should be added, however, that this particular form of construction is not always the best compromise for heavy-duty diesel engines, which will be considered at a later stage.

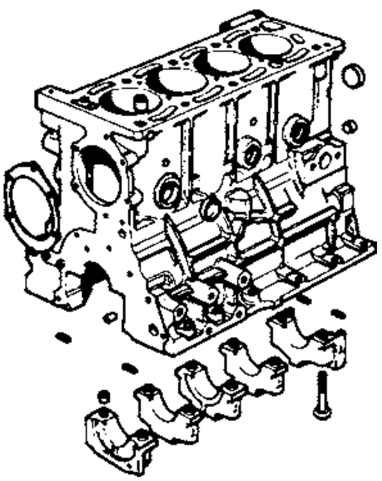

**Figure 1.12** Cylinder block and crankcase

### Piston and rings

The main function of the piston itself is twofold:

1 It acts as a moving pressure transmitter, by means of which the force of combustion is impressed upon the crankshaft through the medium of the connecting rod and its bearings.
2 By supporting a gudgeon pin the piston provides a guiding function for the small end of the connecting rod.

The piston assumes a *trunk* form to present a sliding bearing surface against the cylinder wall, which thus reacts against the side thrust arising from the angular motion of the connecting rod. Since the piston is a major reciprocating part, it must of necessity be light in weight to minimize the inertia forces created by its changing motion – bearing in mind that the piston momentarily stops at each end of its stroke! Another, perhaps obvious, requirement is that the piston must be able to withstand the heat of combustion and should operate quietly in its cylinder, both during warm-up and at the normal running temperature of the engine.

To perform its sealing function efficiently, the upper part of the piston is encircled by flexible metal sealing rings known as the *piston rings*, of which there are typically three in number for petrol engines (Figure 1.13). In combination the piston rings perform several important functions, as follows:

1 The upper *compression rings* must maintain an effective seal against combustion gases leaking past the pistons into the crankcase.
2 These rings also provide a means by which surplus heat is transmitted from the piston to the cylinder wall and thence to the cooling jacket.
3 The lower *oil control ring* serves to control and effectively distribute the lubricating oil thrown on to the cylinder walls, consistent with maintaining good lubrication and an acceptable oil consumption.

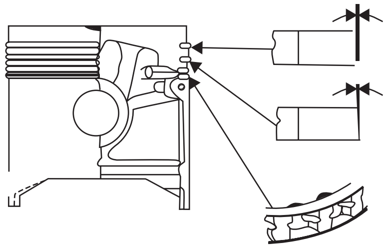

**Figure 1.13** Piston and rings (*Toyota*)

## Connecting rod and bearings

The function of the connecting rod and its bearings is to serve as a constraining link between the reciprocating piston and the rotating crankshaft. A fair analogy of this particular conversion of motion is to be found in the pedalling of a cycle, where the knee, calf and foot of the cyclist may be likened to the piston, connecting rod and crankpin of an engine.

The connecting rod is attached at what is termed its *big end* to the crankpin and at its *small end* to the gudgeon pin of the piston. Each connecting rod big-end bearing is divided into two half-liners, so as to make possible its assembly around the crankpin (Figure 1.14). The big-end bearing housing is therefore formed partly by the lower end of the connecting rod and partly by a detachable cap, the two halves being bolted together. No such complication arises in the case of the small-end bearing arrangements, this end of the connecting rod being formed as a continuous eye.

As a consequence of its reciprocating and partly rotating motion, the connecting rod is subjected to appreciable inertia forces. Its detail design must therefore be such as to ensure the maximum rigidity with the minimum weight.

## Crankshaft assembly and bearings

The crankshaft represents the final link in the conversion of reciprocating motion at the piston to one of rotation at the flywheel. In the case of the multicylinder engine, the crankshaft has to control the relative motions of the pistons, whilst simultaneously receiving their power impulses.

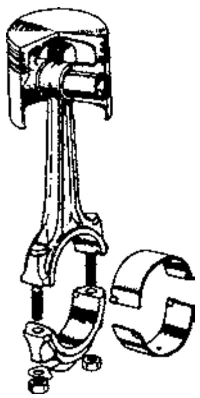

**Figure 1.14** Connecting rod and bearings

A one-piece construction is most commonly used for the motor vehicle crankshaft, which extends the whole length of the engine and must therefore possess considerable rigidity. The timing drive for the engine valve mechanism is taken from the front end of the crankshaft, as is the pulley and belt drive for the engine auxiliaries, such as the cooling fan and the alternator for the electrical system. Attached to the rear end of the crankshaft is the engine flywheel.

The crankshaft is supported radially in the crankcase by a series of bearings, known as the engine *main bearings*. Each main bearing is divided into two half-liners, similar to the big-end bearings, and again to allow assembly around the journals of the one-piece construction crankshaft (Figure 1.15).

## Cylinder head and gasket

The functions of the cylinder head may be listed as follows:

1 It must provide a closure or chamber for the upper part of each cylinder, so that the gas pressure created by the combustion process is constrained to act against the piston.
2 Associated with function 1 is the need to incorporate a gas porting system with inlet and exhaust valves, as well as a platform upon which to mount their operating mechanism. Provision must also be made for a screwed boss to retain the sparking plug.
3 Similar to the cylinder block, the head must form a jacket that allows liquid coolant to circulate over the high-temperature metal surfaces.
4 It is required to contribute to the overall rigidity of the engine structure and maintain a uniform clamping pressure on its sealing gasket with the cylinder block.

The sealing gasket is generally known as the *cylinder head gasket* or simply ‘head gasket’. In liquid-cooled engines the function of the cylinder head gasket is to seal the combustion chambers and coolant and oil passages at the joint faces of the cylinder block and head. The gasket is therefore specially shaped to conform to these openings, and is also provided with numerous holes through which pass either the studs or the set bolts for attaching the cylinder head to the block (Figure 1.16).

Historically, the cylinder head has not always been made detachable from the cylinder block. It was not until the early 1920s, as mentioned previously, that there was a general trend towards simplification of design which led to the abandonment of the cylinder block with integral head. This change

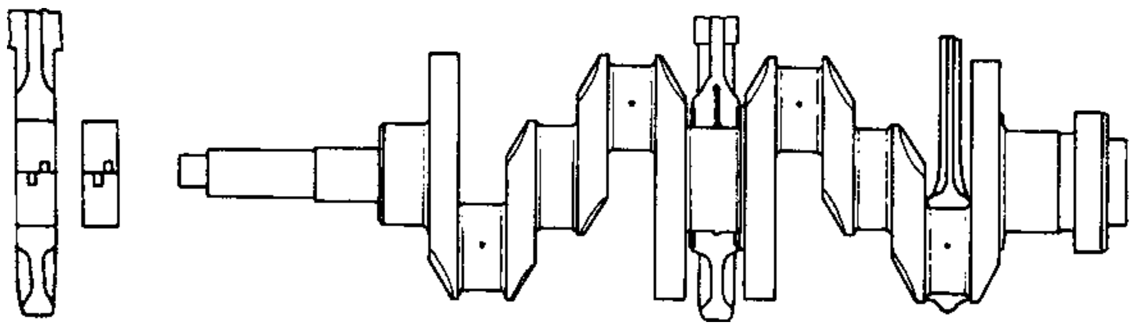

**Figure 1.15** Crankshaft and bearings (*Alfa-Romeo*)

facilitated both the production and the servicing of the motor vehicle engine, although it did introduce the risk of incorrect tightening down of the cylinder head, which can cause at least joint trouble and at worst distortion of the cylinder bores.

### Valve train and timing drive

The overall function of the valve train and timing drive is to provide first for the admission and then the retention of the combustible charge within the cylinder, and finally for the release of the burnt gases from the cylinder, all in synchronism with the motion of the pistons (Figure 1.17).

To perform this sequence of events in accordance with the requirements of the four-stroke cycle, a cam-and-follower mechanism driven at one-half crankshaft speed is used to operate the engine inlet and exhaust valves. There are several different methods of operating the valves from the cam-and-follower mechanism, but in all cases it is necessary for one or more camshafts to be driven from the front end of the crankshaft by what is termed the *timing drive*, and which will be examined in detail at a later stage.

It is, of course, the engine valves themselves that actually perform the functions of admitting the air and fuel charge before its compression, sealing it in the cylinder during compression and combustion, and then releasing the burnt gases after their expansion. Either single or paired inlet and exhaust valves serve each cylinder, and these are mounted in the cylinder head combustion chambers. Valve springs are fitted to ensure that the motion of the valves and their operating mechanism follows faithfully that intended by the cams, and also to maintain adequate sealing pressure when the valves are closed.

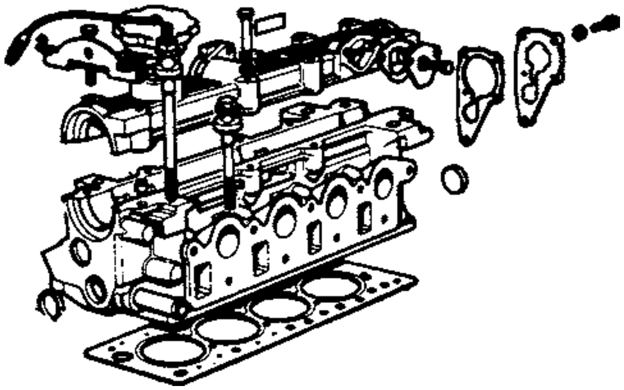

**Figure 1.16** Cylinder head and gasket

### Engine support mountings

The engine is subject to complex vibration effects, which may produce six different free motions or degrees of freedom and combinations of the same. These motions can generate what are termed 'bounce' and 'yaw' about a vertical axis, fore-and-aft movement and roll about a horizontal longitudinal axis, and sideways shake and pitch about a horizontal lateral axis. The three inertia axes intersect at the centre of gravity of the engine (Figure 1.18). For mounting the engine, a resilient anti-vibration system is therefore required, so that the vibratory forces are reduced to the relatively small spring forces transmitted by the support mountings themselves. Apart from supporting the static load of the engine unit and isolating the car structure from engine vibrations, the engine mounts also insulate the engine against vibrations of the car structure and mechanism. By thus minimizing unwanted movements of the engine under all running conditions, any contribution to car shake from this source is reduced.

The engine support mountings almost invariably feature rubber as their spring medium, since this material is highly resilient when loaded in shear (Figure 1.19a). If a rubber block is simply loaded in compression it inevitably bulges sideways (Figure 1.19b), because it is deformable as opposed to being compressible, so the effect is to increase the area of rubber under load and hence reduce its flexibility as a spring. In actual practice, a compromise is often sought by loading the rubber partly in compression and partly in shear by inclining the rubber mountings (Figure 1.19c). A widely used type of support mounting is the sandwich unit, which generally consists of either a rectangular or a circular block of rubber with metal attachment plates bonded to its upper and lower faces. For some applications, one or more metal interleaves may be incorporated in the mount (Figure 1.19d). These serve

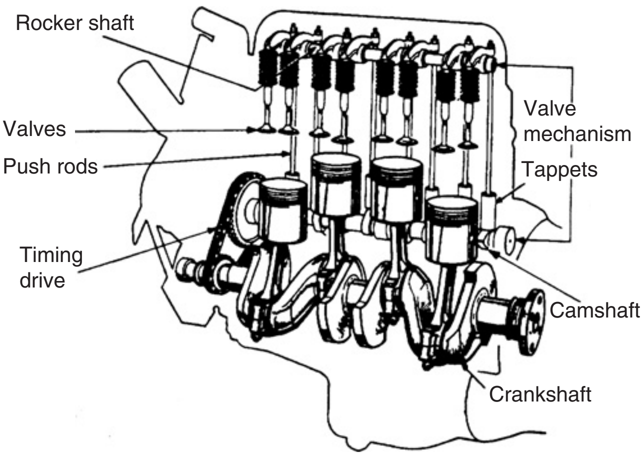


**Figure 1.17** Valve train for a push-rod overhead valve engine

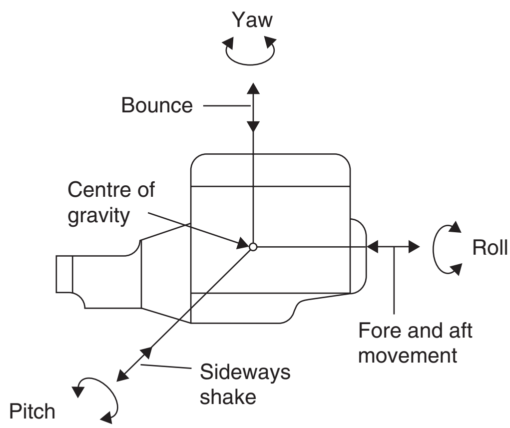


**Figure 1.18** Vibratory motions of an engine

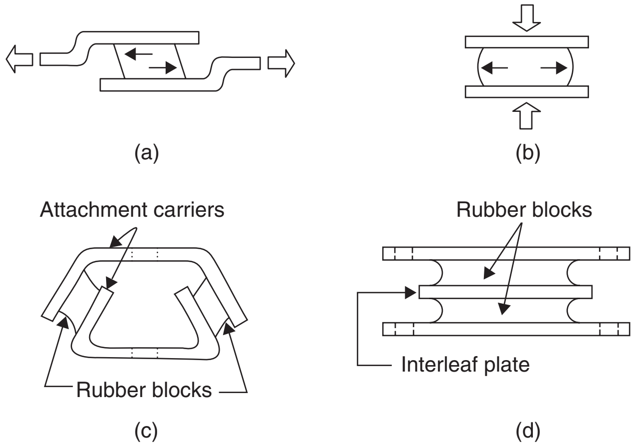


**Figure 1.19** Engine support mountings: (a) rubber loaded in shear (b) rubber loaded in compression (c) rubber loaded in combined compression and shear (d) rubber sandwich with interleaf

to restrict sideways bulging of the rubber, thus further reducing its flexibility under compression loading, but leaving its shear loading characteristics unaltered, and they also assist in dissipating the heat of internal friction created by flexing of the rubber. Another advantage of using rubber, rather than steel, for the spring medium is that transmission of sound through the mountings is less, since there is no metal-to-metal path for the sound to travel along. The various types of resilient mounting system employed for longitudinally and transversely disposed engine and transmission units are described in Section 30.1.

## 1.5 CYLINDER AND CRANKTHROW ARRANGEMENTS

### Cylinder number and displacement

Every new engine must be designed with a specific type of service in view, which then determines its general characteristics. Important among these for the car is smooth and efficient operation over a wide range of speeds and loads. Herein lies the explanation why no manufacturer lists a single-cylinder engine and few produce an engine with fewer than four cylinders. The following considerations are pertinent.

*Smoothness*
With a single-cylinder engine operating on the four-stroke cycle, it will be recalled that only one power impulse occurs for every two revolutions of the crankshaft. The fluctuations in crankshaft torque of a single-cylinder engine would, therefore, be quite unacceptable in motor vehicle operation. Hence the greater number of cylinders used, the shorter will be the interval between the power impulses and the smoother will be the flow of torque from the engine.

*Mechanics*
It will be evident from the explanation of the factors governing engine power given in Section 1.2 that the power output obtainable from a single-cylinder engine of realistic dimensions and running at a reasonable speed is unlikely to be sufficient for motor vehicle, as distinct from motorcycle, requirements. This is because a practical limit is set on individual cylinder size by dynamic factors, namely the inertia forces created by accelerating and decelerating the reciprocating masses comprising the piston assembly and the upper portion of the connecting rod.

If an unusually large, and consequently heavy, piston were adopted in a single-cylinder engine intended for high-speed operation, the dynamic effects could be such as to increase the magnitude of the inertia forces to a level that at least would make engine imbalance unacceptable and at worst would prove mechanically destructive. This is partly because the inertia forces are proportional to the cube of the piston mass (e.g. doubling the piston mass will cause the inertia forces to become eight times as great), and partly because they also vary as the square of the engine speed (e.g. doubling the engine speed will cause them to become four times as great).

*Temperature*
Another problem arising from the use of an unduly large cylinder bore is that cooling of the piston and valves can be seriously impaired, and this may lead to their failure from thermal overstressing. If this is to be avoided it would be necessary to incorporate special features of design, such as oil cooling of the piston, which is often practised in marine diesel engines with very large cylinder bores (although this may still be required for automotive engines with forced induction).

For these reasons it has therefore become established practice for the displacement of an engine to be shared among multiple small cylinders, rather than confined to a single large cylinder. To this must be added the proviso that an excessively large number of cylinders increases the friction losses in an engine, quite apart from the extra complication which makes it more costly to build and maintain. Some authorities now claim that 330 $cm^3$ represents an optimum size of cylinder.

### Arrangement of engine cylinders

Once the displacement and number of cylinders have been decided in relation to the required performance characteristics of a new engine, the next consideration is how the cylinders are to be arranged. In cars they may be arranged in three different ways, each with its own advantages and disadvantages.

*In-line cylinders*
As would be expected, in this arrangement all the cylinders are mounted in a straight line along the crankcase, which confers a degree of mechanical simplicity. Such engines are now produced with any number of cylinders from two to six (Figure 1.20), with four cylinders continuing to represent a very widely used cylinder arrangement (Figures 1.30 and 3.10). The single bank of cylinders may be contained in either a vertical or an inclined plane. The latter type is sometimes referred to as a sloper or slant engine. For this particular mounting of an in-line cylinder engine, the advantages usually claimed include a reduction in overall installation height and improved accessibility for routine servicing.

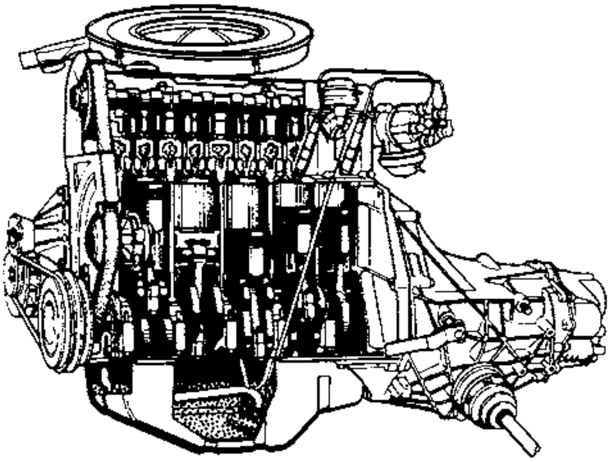

**Figure 1.20** An interesting in-line five-cylinder engine (*Audi*)

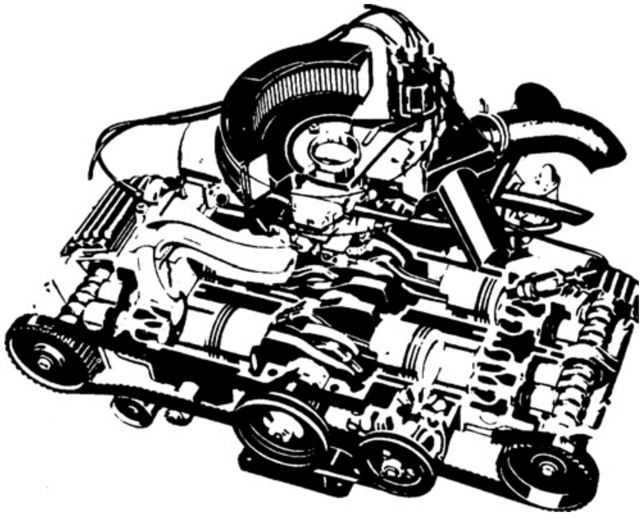

**Figure 1.21** A modern horizontally opposed four-cylinder engine (*Alfa-Romeo*)

With the exception of the in-line two-cylinder or parallel twin engine where both pistons move in unison to obtain even firing intervals, this arrangement of cylinders provides generally satisfactory balance in respect of the reciprocating parts, especially in six-cylinder versions. Apart from space requirements, a mechanical limitation is placed on the acceptable length of an in-line cylinder engine, because of the difficulty in controlling torsional vibrations of the crankshaft. This is a topic that will be discussed at a later stage.

*Horizontally opposed cylinders*

Horizontally opposed engines have their cylinders mounted on the crankcase in two opposite banks and are sometimes referred to as flat or boxer engines. They are typically produced in two-, four- and six-cylinder versions (Figure 1.21). The main advantages usually claimed for them include inherently good balance of the reciprocating parts, a low centre of gravity, which contributes to car stability, and a short engine structure. It is the latter feature that makes this arrangement of cylinders particularly suitable both for front-wheel-drive and rear-engined cars, since the engine can be mounted either ahead of or behind the driven wheels with the minimum of overhang. By virtue of its low overall height, the horizontally opposed engine can readily allow a sloping bonnet line in front-engined cars and also provide additional space for stowing luggage above it in rear-engined cars. Furthermore, it lends itself admirably to air cooling because with an in-line cylinder arrangement it is difficult to get the rear cylinders to run as cool as the front ones, unless the engine is installed transversely.

The disadvantages associated with horizontally opposed cylinders include the need for lengthy intake manifolds if a central carburettor is used, the duplication of coolant inlet and outlet connections in the case of liquid cooling, and much reduced accessibility for the cylinder heads and valve mechanism. Its greater width can also impose restrictions on the available steering movements of the wheels.

*V-formation cylinders*

With V engines the cylinders are mounted on the crankcase in two banks set at either a right angle or an acute angle to each other. They may be produced in four-, six-, eight- and occasionally twelve-cylinder versions (Figure 1.22). Where a V cylinder arrangement has been adopted in preference to mounting the cylinders in line, it has usually been in the interests of providing a more compact and less heavy engine. In particular, the overall length of the engine can be appreciably reduced, so that both the structure and the crankshaft can be made more rigid. The former is thus better able to accept greater combustion loads and the latter is less prone to torsional vibrations. For typically large-displacement V engines, the inherently wider cylinder spacings ensure adequate size of coolant passages, both around the cylinder walls and in the hot exhaust valve regions in the cylinder heads (Figure 1.36).

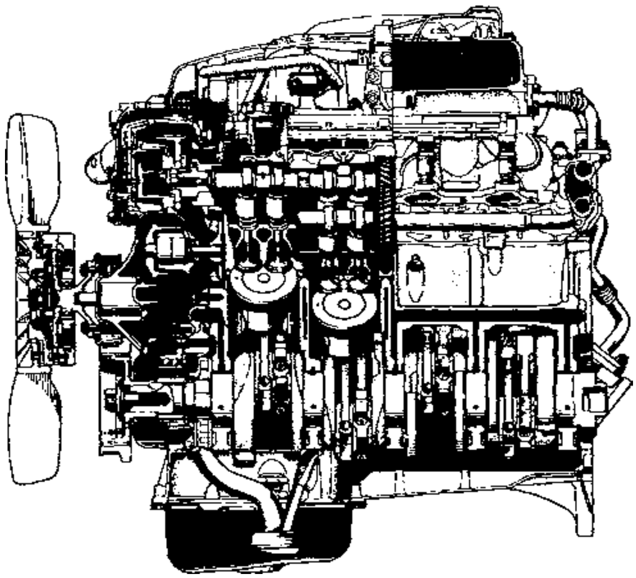

**Figure 1.22** A high-grade V eight-cylinder engine (*Lexus/Toyota*)

On the debit side, the V cylinder arrangement generally presents a more difficult balancing problem and also demands a more elaborate intake manifold from a central carburettor or a single-point fuel injection system. In common with horizontally opposed cylinder layouts, V engines tend to be more costly to produce, since there are more surfaces to machine and some duplication of structural features.

However, the V six-cylinder engine has become widely adopted in modern practice for medium class cars with engine capacities, ranging between 2.4 and 3.2 litres. Once described by D.A. Martens of Chevrolet as ‘a product of necessity’, the V6 engine can provide improved performance and enhanced smoothness when transversely mounted in front-wheel-drive cars, where an in-line four-cylinder engine may also be available as a lower cost option.

*W-formation cylinders*
A more recent development has been the ‘double-V’ or ‘W’ cylinder engine, introduced in eight-, twelve- and sixteen-cylinder versions by the Volkswagen group. Basically the W engine, as it is now generally known, comprises two wide-angle cylinder banks, each of which incorporates two offset rows of cylinders inclined at a much narrower angle. In the early 1990s Volkswagen had introduced a single bank narrow-V six-cylinder engine, which externally resembled an in-line unit, but internally had two offset rows of three cylinders inclined at an angle of only 15°. It therefore followed that if two narrow-V six-cylinder banks could themselves be mounted in a wide-V formation of 72° and share a common crankshaft, a W12 engine then becomes a practicality. Several advantages may be gained from this unorthodox, but albeit complex, layout when compared to a conventional V12 engine. A W12 engine can offer a more compact design and indeed may require no more installation space than a conventional V8 engine. It can also be made less heavy, whilst possessing greater structural rigidity with reduced vibration levels.

A distinguishing feature of the narrow-V type cylinder bank arrangement is that the axes of the cylinder pairs intersect below, instead of coinciding with, the axis of the crankshaft (Figure 1.23). This departure from the operating geometry of a conventional wide-angle V-cylinder layout becomes necessary to accommodate the motion of the pistons as they approach their bottom dead centre positions. Similar consideration applies to the need for either separate, or split-type, staggered crankpins for the cylinder pairs.

Historically, the narrow-V cylinder layout had its origins in piston aero engine practice. It was conceived by the Lancia company in Italy at the beginning of World War I, the aim being to reduce the width of a conventional V-cylinder engine for aircraft installation. Smaller narrow-V four-cylinder engines were subsequently used for many years in the passenger cars made by this company, the Lancia Lambda model of the 1920s being a well-known classic example.

## Crankthrow arrangements

The arrangement of the crankthrows in relation to the disposition of the engine cylinders and their number is determined by two sometimes conflicting considerations: acceptable engine balance, and equal firing intervals.

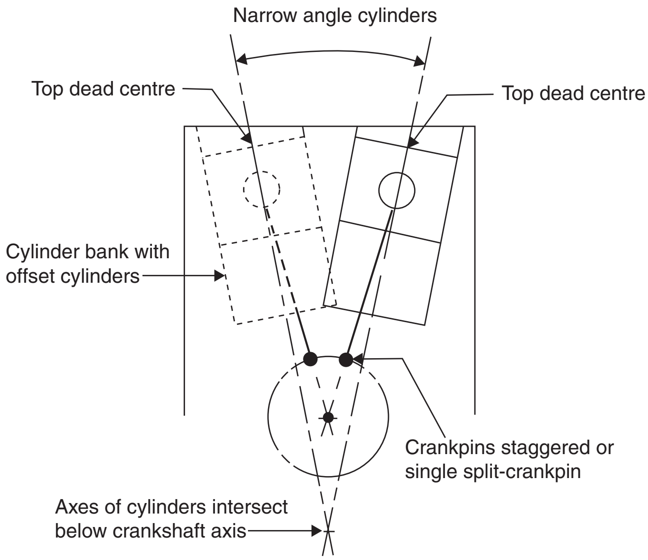


**Figure 1.23** Basic arrangement of cylinder pair in narrow-V engine

*Acceptable engine balance*
Theoretically, a perfectly balanced engine is one which, when running and ‘suspended in space’ from its centre of gravity, would exhibit no vibratory movements whatsoever. In reality, of course, the reciprocating engine can never be perfectly balanced. Apart from any rotation imbalance, there are also the inevitable torque irregularities, although as stated previously these can be minimized by the use of more than one cylinder.

Multiple cylinders further allow a much better standard of general engine balance, provided that the choice of a particular arrangement and number of cylinders takes into account the presence of what are termed primary and secondary inertia forces. The effect of these forces, which act in the reciprocating sense, will now be explained in simple qualitative terms.

*Primary inertia forces* These arise from the force that must be applied to accelerate the piston over the first half of its stroke, and similarly from the force developed by the piston as it decelerates over the second half of its stroke (Figure 1.24a). When the piston is around the mid-stroke position it is then moving at the same speed as the crankpin and no inertia force is being generated.

For an engine to be acceptable in practice, the arrangement and number of its cylinders must be so contrived that the primary inertia forces generated in any particular cylinder are directly opposed by those of another cylinder. Where the primary inertia forces cancel one another out in this manner, as for example in an in-line or a horizontally opposed four-cylinder engine with the outer and inner pair of pistons moving in opposite directions, the engine is said to be in *primary balance* (Figure 1.24b).
*Secondary inertia forces* These are due to the angular variations that occur between the connecting rod and the cylinder axis as the piston performs each stroke. As a consequence of this departure from straight-line motion of the connecting rod, the piston is caused to move more rapidly over the outer

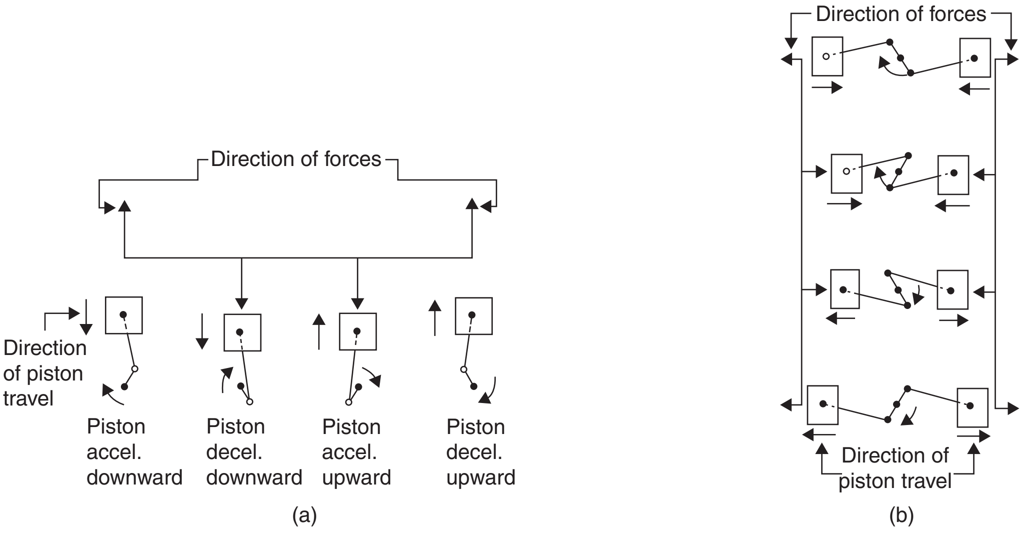


**Figure 1.24** Primary inertia forces: (a) unbalanced (single-cylinder) (b) balanced (horizontally opposed cylinders)

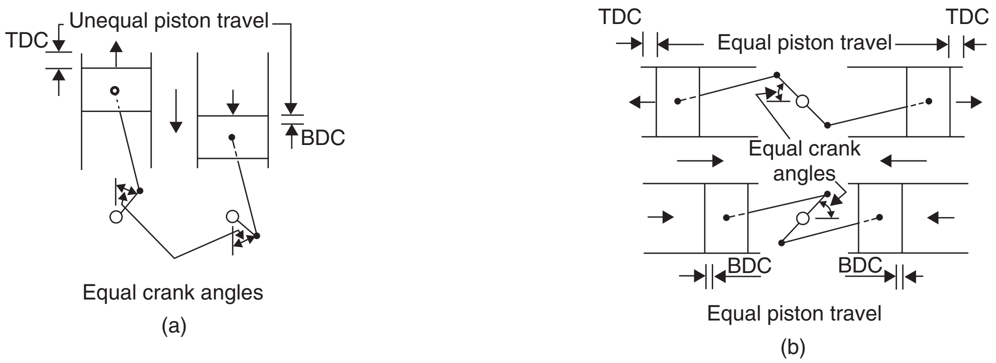


**Figure 1.25** Secondary inertia forces: (a) unbalanced (single-cylinder) (b) balanced (horizontally opposed cylinders)

half of its stroke than it does over the inner half. That is, the piston travel at the two ends of the stroke differs for the same angular movements of the crankshaft (Figure 1.25a). The resulting inequality of piston accelerations and decelerations produces corresponding differences in the inertia forces generated. Where these differing inertia forces can be both matched and opposed in direction between one cylinder and another, as for example in a horizontally opposed four-cylinder engine with corresponding pistons in each bank moving over identical parts of their stroke (Figure 1.25b) the engine is said to be in *secondary balance*.

It is not always practicable for the cylinders to be arranged so that secondary balance can be obtained, but fortunately the vibration effects resulting from this type of imbalance are much less severe than those associated with primary imbalance and can usually be minimized by the flexible mounting system of the engine. This is confirmed by the long-established and popular in-line four-cylinder engine, which possesses primary balance but lacks secondary balance. However, the continuing search for greater refinement of running with this type of engine led, in the mid 1970s, to a revival of interest in the use of twin counterbalancing shafts for cancelling out these secondary inertia forces. The modern application of this system of harmonic balancing will be described in Section 1.8.

*Equal firing intervals*

The arrangement of the crankthrows is also determined by the requirements for even firing intervals of the cylinders and for spacing the successive power impulses as far apart as possible along the crankshaft, so as to reduce torsional deflections or twisting effects. For any four-stroke engine the firing intervals must, if they are to be even, be equal to 720° divided by the number of cylinders.

For in-line four-cylinder engines the first and fourth crankthrows are therefore indexed on one side of the crankshaft and the second and third throws on the other side (Figure 1.26a). The *firing order* of these engines, numbering from the front, may then be either 1-3-4-2 or 1-2-4-3 at 180° intervals. Similarly, in the case of in-line six-cylinder engines, the crankthrows are spaced in pairs with an angle of 120° between them. Hence, the first and sixth crankthrows are paired, as are the second and fifth, and likewise the third and fourth (Figure 1.26b). The firing order may then be such that no two adjacent cylinders fire in succession; that is, either 1-5-3-6-2-4 or 1-4-2-6-3-5 at, of course, 120° intervals.

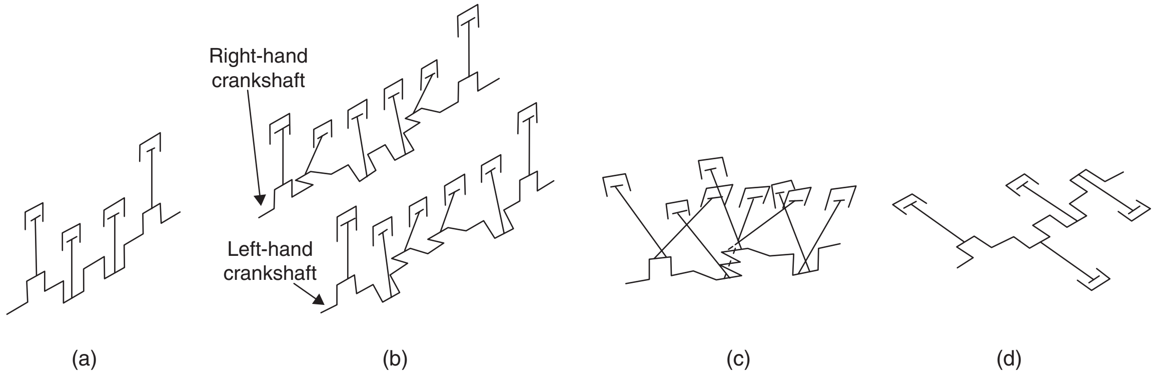


**Figure 1.26** Cylinder and crankthrow arrangements: (a) in-line four-cylinder (b) in-line six-cylinder (c) eight-cylinder (d) horizontally opposed four-cylinder

For V and horizontally opposed engines (Figures 1.26c and d), the firing orders follow a sequence similar to those of in-line engines, but the crankthrows are so disposed that firing alternates between the cylinder banks as it proceeds to and fro along the crankshaft.

The V6 engine is nevertheless deserving of special comment. When the Italian Lancia company pioneered the V6 engine layout in 1950, they inclined the two banks of cylinders at 60° to each other, which together with a six-throw crankshaft conferred equally spaced firing intervals of 120°. However, this was not generally true of other early V6 engines, where their designers had adopted an included angle of 90° between the cylinder banks and used a shorter three-throw crankshaft with the three pairs of connecting rods each sharing common, double-length, crankpins. This arrangement resulted in firing intervals of 90° and 150° alternatively instead of even 120° intervals, so although these engines were more economical on fuel than a V8 engine they were not particularly smooth running. To overcome the problem of uneven firing intervals with a 90° V6 engine, Buick in America retained a three-throw crankshaft but ingeniously replaced the common, double-length, crankpins by adjacent single crankpins that were staggered by 15° in opposite directions to produce a so-called 'split-pin' crankshaft. It may be of interest to recall that the reason for adopting a 90° included angle between the cylinder banks of early V6 engines was basically one of production economy. In the case of American V6's it was to utilize tooling that already existed for their (90°) V8 engines, while conversely for European V6's it was to create new tooling that could also be used to meet any future demand for V8 engines.

Some designers of modern even firing V6 engines (even firing especially being desirable where turbocharging is featured) have in fact reverted to a 60° included angle between the cylinder banks, but have retained a three-throw crankshaft supported in four main bearings. The required angular spacing for the crankpins of each throw is then obtained by using what is termed a '*flying arm*' (Figure 1.77). This can be likened to a crankweb that does not connect with a main journal and which may also be utilized for balancing purposes.

*Cylinder numbering sequence*

The numbering sequence for the cylinders of motor vehicle engines has long been defined by the various standards organizations. In British and European practice, as defined by British Standards BS 5672: 1991 and International Organization for Standardization ISO 1204:1990(E) respectively, the cylinders of an in-line or single bank engine are numbered consecutively as viewed from the flywheel end of the engine (which is in contrast to earlier established practice where the cylinders were numbered consecutively as viewed from the nose of the crankshaft). Both a capital letter and an identical system of numbering is used to designate the cylinders of V and horizontally opposed or double bank engines, the prefix letters A and B denoting the left- and right-hand banks of cylinders respectively as viewed from the flywheel end of the engine. In American practice, the suffix letters R and L are preferred to denote the right- and left-hand banks of cylinders respectively. Owing to the differences that can arise, past and present, the service engineer should always consult the manufacturer's engine specification for confirmation of the cylinder numbering system that applies.

## 1.6 CYLINDER BLOCK, CRANKCASE AND HEAD

### Cylinder block construction

Cylinder block construction takes two forms. *Closed-deck* construction (Figure 1.27a) represents long-established practice and resembles a deep box-like enclosure for the cylinder barrels that also serves as a coolant jacket (Figure 1.28). Transfer ducts are provided in the top face or closed deck of the cylinder block, so as to permit the circulation of coolant to the cylinder head. With the *open-deck* construction (Figure 1.27b) the cylinder barrels are free-standing in that they are attached only to the lower deck of the cylinder block, which in past applications utilized detachable cylinder liners that tended to result in a less rigid construction. By dispensing with a continuous top face, the open-deck construction nevertheless reduces the complexity of the cylinder block casting. Where gravity sand casting is used it facilitates the coring for the mould into which the metal is poured. However, the increasing preference for using aluminium alloy, rather than grey cast iron, for the cylinder block and crankcase of modern lighter weight engines, has led to their manufacture by high-pressure die casting in the interests of economical mass production. Since this method of casting necessarily

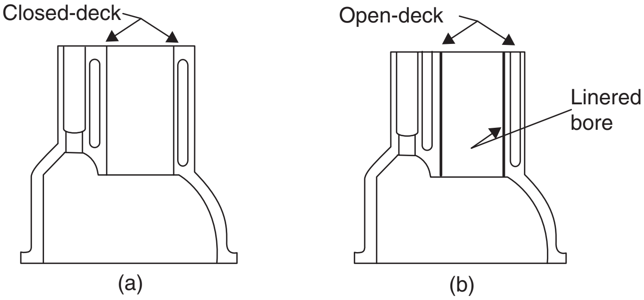


**Figure 1.27** Cylinder block construction: (a) closed deck (b) open-deck

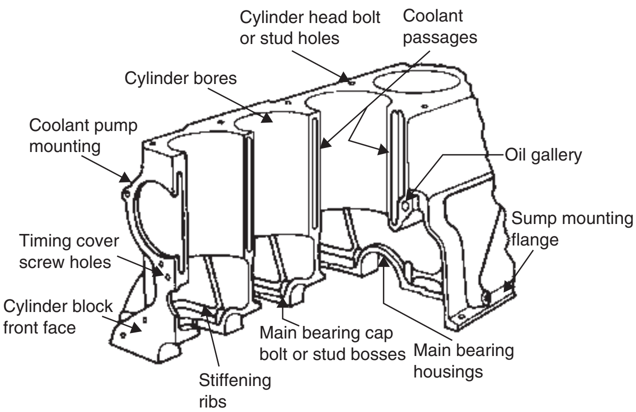


**Figure 1.28** Cut-away view of cylinder block

involves the use of steel instead of sand moulds, the need for an open-deck construction to allow withdrawal of the steel cores becomes mandatory. Also, the liners may be cast directly into the cylinder block to restore structural rigidity. An open-deck construction further allows inspection of the coolant jacket for accumulated deposits. To perform this operation in a closed-deck cylinder block requires the addition of detachable cover plates.

In both closed-deck and open-deck cylinder blocks numerous internal supporting bosses must be provided for either the studs, or the setbolts, which clamp the cylinder head to the block. Wherever practicable, these bosses are aligned with the crankcase bulkheads that support the main bearings. This is to secure a direct path of load transference between the cylinder head and main bearing caps and thus minimize bending stresses within the cylinder block and crankcase structure (Figure 1.29). A noteworthy modern example of engine construction where this principle is carried to its logical conclusion can be found in the K Series engine originally developed by the Rover Group (Figure 1.30). In this design ten long bolts pass down through the cylinder block so that they clamp together in sandwich fashion the cylinder head, cylinder block, main bearing ladder and main bearing oil feed rail; all of these layers are located relative to one another by tubular dowels. This type of through-bolt construction therefore makes for a very even distribution of clamping load virtually from top to

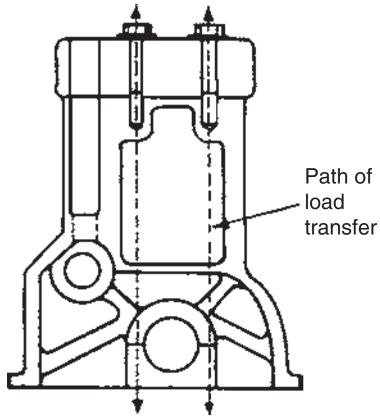


**Figure 1.29** Direct load transference between the studs of the cylinder head and main bearing caps

bottom of the engine, which is particularly beneficial in view of its structural material being aluminium alloy.

For closed-deck constructions, the internal supporting bosses for the head studs are arranged symmetrically about the cylinder walls, so that the cylinder head clamping load is as evenly distributed as possible around them. The object is, of course, to avoid any tendency towards cylinder bore distortion, which could lead to blow-by of the combustion gases and increased oil consumption. In the case of open-deck constructions, the supporting bosses for the head studs are usually disposed along the walls of the coolant jacket.

## Cylinder bores

Clearly, the cylinder bores constitute the most important feature of the cylinder block. Since they act as a guide and a sealing surface for the sliding piston and rings, their accuracy of machining must be such as to minimize any out-of-roundness and taper effects, and to ensure that they are truly at right angles to both the crankshaft and the top deck of the block.

The cylinder bores must also be given a carefully controlled surface finish, because too rough a surface would cause wear, and too smooth a surface would hinder the running-in process. A suitable surface finish is usually obtained by final honing to give a cross-hatched finish (Figure 1.31), which retains the oil in the bores to lubricate the pistons and so reduces friction losses. The question of the most suitable surface finish for new cylinder bores is one of long standing, and it is perhaps significant that an American engineer once observed that somehow the engine knows how to finish the bore better than we do!

Cylinder bore wear in service may be attributed to various factors, which will be reviewed later in connection with the associated wearing of piston rings.

As the degree of wear varies in different parts of the cylinder, an overall assessment of bore wear is most conveniently made by using a dial gauge equipped with an extension spindle, so that wear readings can be observed outside the cylinder. The gauge is self-aligning in the cylinder bore and in use is rocked over the cylinder axis from side to side, the maximum dial reading corresponding to the diameter being measured.

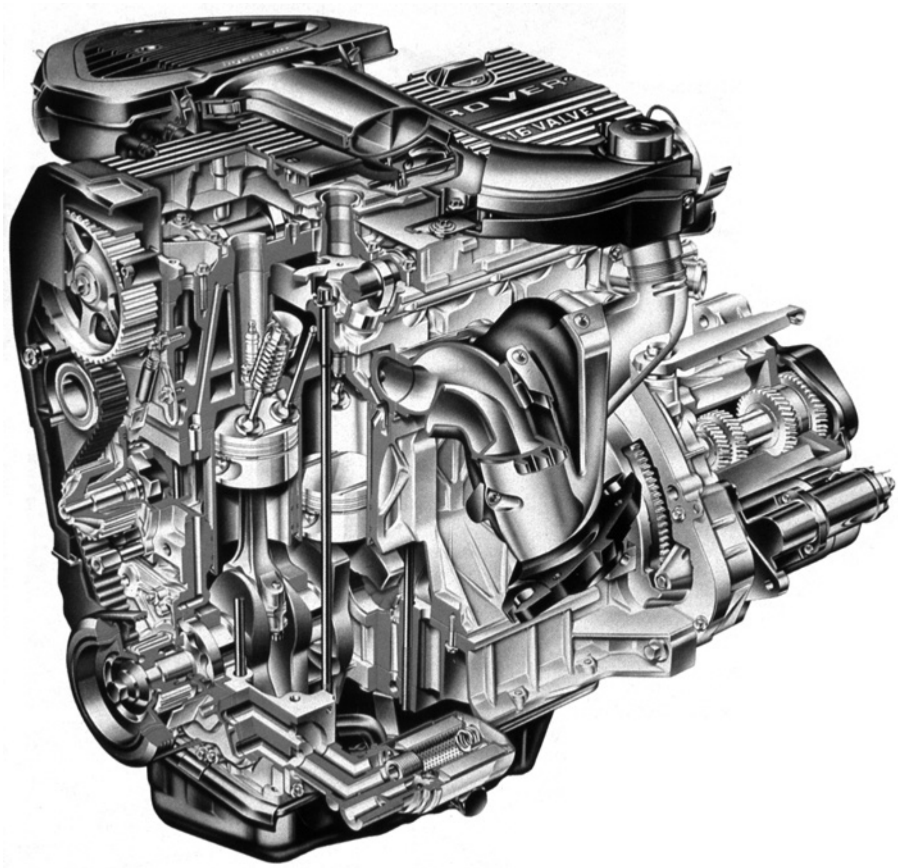


**Figure 1.30** A modern example of a through-bolted engine construction

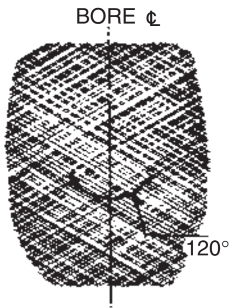


**Figure 1.31** Honing pattern for cylinder bore

## Crankcase construction

For in-line and V-formation engines, the most commonly used form of crankcase resembles a tunnel structure which extends downwards from the cylinder block (Figure 1.32). The roof is formed by the lower deck of the cylinder block and it is closed off at the base by either a detachable sump or a transmission housing. The crankshaft is underslung in the crankcase and supported by front, intermediate and rear main bearing bulkheads that form a series of crank chambers.

Until recently, it was customary for the side walls or *skirt* of the crankcase to be extended below the axis of the crankshaft, so as to increase the resistance to bending of the structure in the interests of engine smoothness, and also to simplify

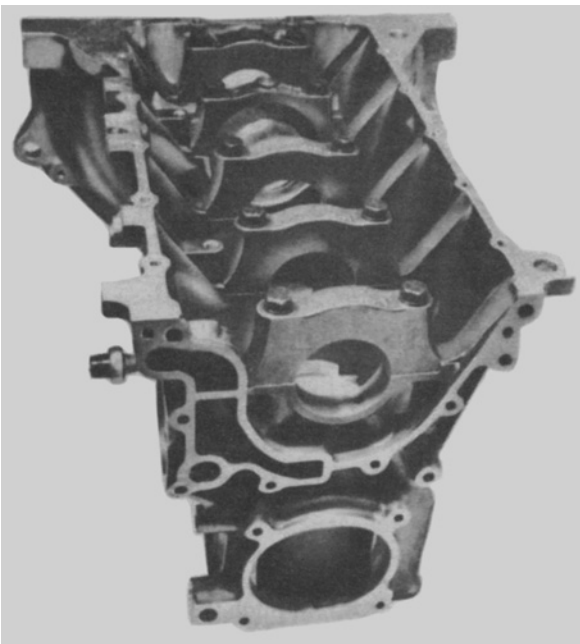

**Figure 1.32** Underside view of an aluminium alloy crankcase for a four-cylinder engine with five main bearings (*Chevrolet*)

the attachment of the sump. With the aid of modern computer stress analysis techniques, however, the present trend is to return to the much earlier practice of ending the crankcase side walls at the same level as the crankshaft axis, without sacrificing rigidity and with a consequent saving in weight.

For horizontally opposed engines, the cylinder blocks are again cast integral with the crankcase, which is usually divided on its vertical centre line (Figure 1.33). The crankcase halves are then clamped together by through-bolts on either side of the crankshaft main bearings. A series of bolts is also fitted around the peripheral joint faces of the crankcase, it being usual for the sump to be made integral with each half. Less commonly, a one-piece construction is used with an open underside, detachable main bearing caps and oil sump (Figure 1.33).

### Main bearing locations

For in-line and V engines, the upper main bearing halves are carried direct in saddles formed in the crankcase bulkheads, whilst detachable inverted caps of great rigidity accommodate the lower main bearing halves. These bearing caps are usually recessed into the underside of their respective bulkheads and secured to them by either studs or setbolts, which thus support the maximum combustion loads imposed upon the crankshaft (Figure 1.34a). In some designs where the crankcase-to-sump joint face is at the same level as that of the crankshaft axis, the bearing caps may be located laterally by a pair of dowels, since it is no longer expedient to recess them into the lower face of the crankcase (Figure 1.34b).

Of more recent application is a form of crankcase construction that embodies a one-piece main bearing ladder or deck (Figures 1.30 and 1.35). This in effect integrates the main bearing caps into a single rigid structural element, rather similar to the bed plate construction found in very large diesel engines.

In some later designs of V eight-cylinder engine with aluminium alloy crankcases, the fastening arrangements for the main bearing caps may be duplicated, so that each cap is retained by four nuts and studs threaded vertically into the cylinder block above (Figure 1.36). As a further contribution to crankcase rigidity and bearing support, transverse bolting for the main bearing caps may also be employed (Figure 1.36), a feature that has previously been found in diesel engine practice (Figure 2.6). To complement these more comprehensive main bearing cap fastening arrangements, aircraft style 'bihexagon' or 'twelve-point' headed bolt and nut forms may

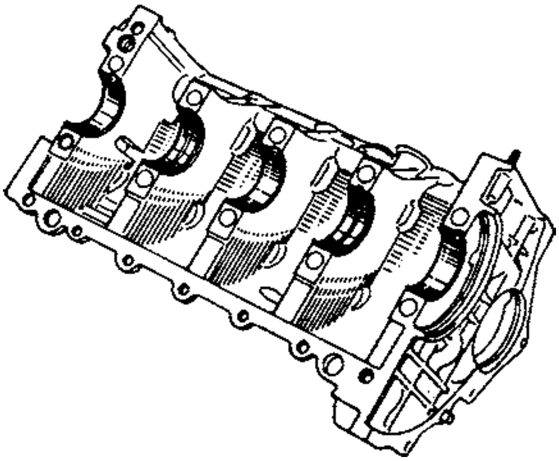

**Figure 1.35** A one-piece main bearing deck (*Renault*)

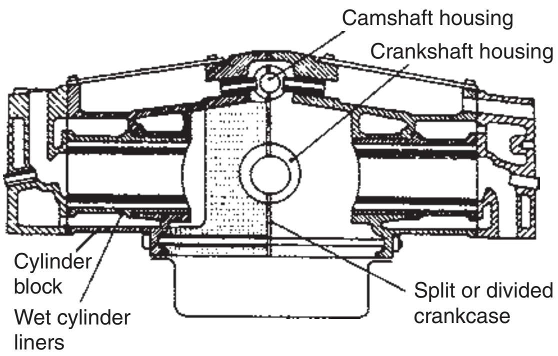


**Figure 1.33** Horizontally opposed cylinders with divided crankcase

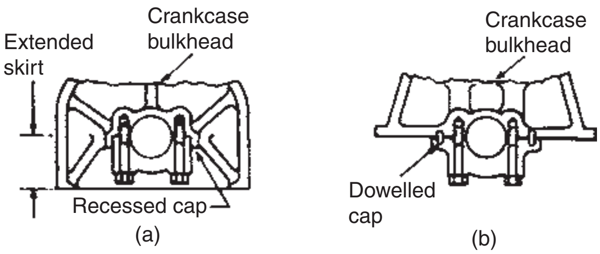


**Figure 1.34** Crankcase bulkhead and main bearing cap locations

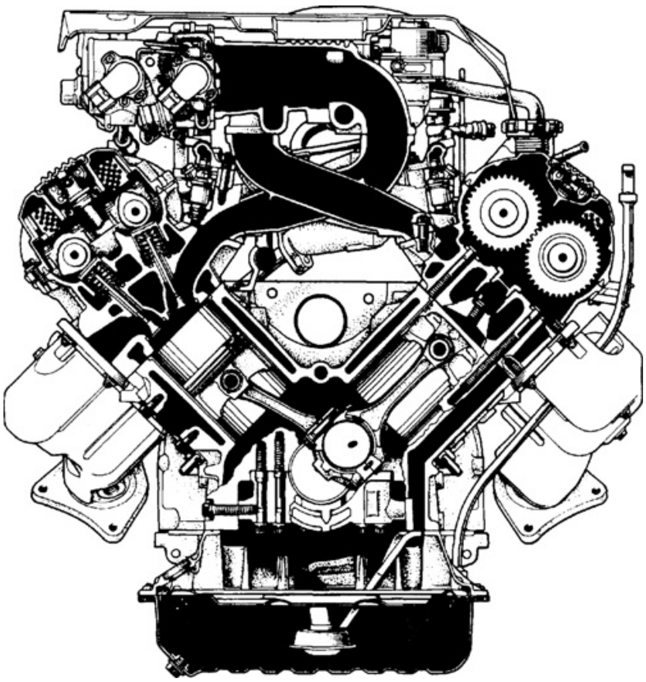

**Figure 1.36** Vertical and transverse fastening for the main bearing caps in a high-grade V eight-cylinder engine (*Lexus/Toyota*)

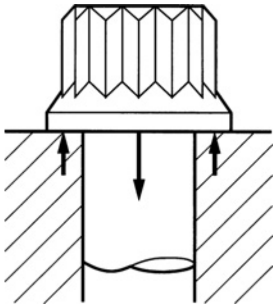

**Figure 1.37** Bihexagon or twelve-point headed bolt

be specified (Figures 1.36 and 1.37). The main advantage of the bihexagon pattern is that for a given thread size it saves weight, because its head diameter is much smaller than a corresponding hexagon head, whilst its integral washer face restores an equivalent clamping area. Comparable torque tightening values can still be applied, since the increased number of socket-engaging flats compensate for the reduced turning radius of the bolt head or nut. This form of screw fastener is finding increasing favour among engine designers (Figure 1.30).

With horizontally opposed engines, complementary saddes for the main bearing half-liners are machined in each half of the crankcase. The main bearings cannot, therefore, be changed unless the crankcase halves are taken apart. This drawback of internal inaccessibility has been overcome in a more recent version of the horizontally opposed engine which features a one-piece construction for the cylinders and crankcase, as described in the previous section.

### Camshaft bearing locations

A crankcase-mounted camshaft is supported either to one side of, or directly above, the crankshaft, depending on whether an in-line or a V cylinder arrangement is used. With both locations the camshaft bearings are usually carried in webbed extensions of the corresponding main bearing bulkheads. Spanning the underside of the camshaft may be a lubrication trough, which is formed as an integral part of the crankcase. The camshaft followers or tappet barrels slide in bores machined either directly in the material of the crankcase (Figure 1.17), or in bolted-on tappet blocks. In the case of horizontally opposed engines, the bearing bores for a central camshaft are formed similar to those for the main bearings beneath them (Figure 1.33).

### Cylinder block and crankcase materials

The combination of cylinder block and crankcase is the single largest and most expensive component of an engine and may be produced from either cast iron or aluminium alloy.

Cast iron is an alloy of iron and carbon; there are two general classifications, known as white cast iron and grey cast iron. Although both varieties have a carbon content between 2.5 and 4.0 per cent, the difference between them is concerned with the condition in which the major portion of the carbon exists in the metal structure. Grey cast iron (so called owing to its grey rather than white appearance when fractured) is used for cylinder block and crankcase manufacture, because most of the carbon is present as flakes of graphite. This feature not only makes the material more readily machinable, but also provides a satisfactory wear- and corrosion-resistant bearing surface for the cylinder bores. The rigidity of cast iron is such that it exhibits very little tendency towards distortion under the loads and temperatures encountered in the highly stressed engine structure. In addition, it possesses useful sound-damping properties.

Apart from its low cost, an outstanding characteristic of grey cast iron is the ease with which it can be cast into intricate shapes of thin section. Using modern casting techniques, the thickness of the cylinder block walls can therefore be minimized to save weight, which otherwise is the only real disadvantage with the cast iron cylinder block and crankcase.

The term ‘aluminium’ is generally used to describe not only the very soft and ductile commercially pure variety, but also the numerous aluminium alloys that comprise aluminium with usually more than one element added to it. Only the latter are of interest for cylinder block and crankcase construction, since they can be made harder and more readily machinable than commercial aluminium, which tears badly and poses screw-threading problems. The main attraction of using an aluminium alloy for casting the engine structure is the saving in weight that it affords; its density is about one-third that of cast iron. On the debit side, its strength is about two-thirds that of cast iron. This means that metal sections have to be thickened to compensate for the lower strength, so that in reality the saving in weight is generally nearer to one-half that of the cast iron version.

There are two main classifications for aluminium alloys: those which can be hardened by cold-working processes, and others that can be heat treated to obtain the desired mechanical properties. It is the latter alloys of the aluminium-silicon type that find the widest application for cylinder block and crankcase manufacture, because they retain their strength at moderately high temperatures, possess good casting fluidity and are the most resistant to corrosion.

Although an appreciably lighter construction can be obtained by using aluminium alloy, its wear-resistant properties are less acceptable for the cylinder bores, so that pistons of similar material cannot run directly in them. To overcome this disadvantage, either cast iron cylinder liners, iron-coated pistons or, more recently, silicon-carbide particles dispersed in nickel-plated cylinder bores may be employed. An aluminium alloy engine structure is also less tolerant both of careless handling, especially in respect of screw-thread connections, and of accidental over-heating through loss of coolant.

Historically, aluminium alloys were introduced extensively during the beginning of the aircraft industry. It is perhaps of interest to recall that one of the earliest examples of a cast aluminium crankcase was that used in the engine of the Wright brothers’ first aircraft.

### Cylinder head construction

In petrol engines the lower deck of the cylinder head contains the cylinder combustion chambers (Figure 1.38). Less commonly, these may be formed either within the piston heads, or by the combination of an inclined top deck for the

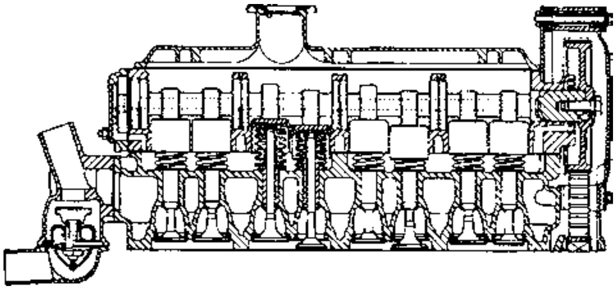

**Figure 1.38** Sectional view of a cylinder head (*Fiat*)

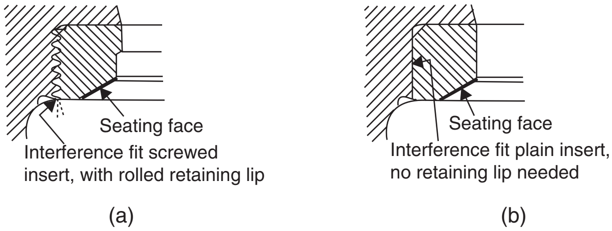


**Figure 1.39** Valve seat inserts: (a) early practice (b) later practice

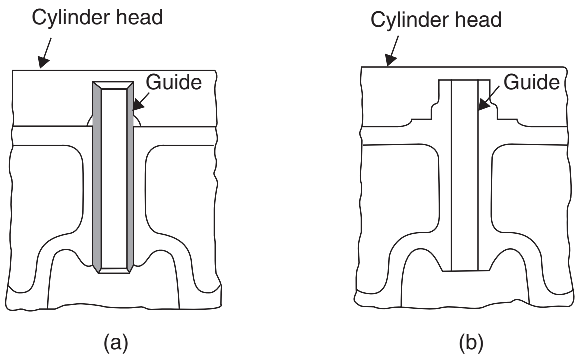


**Figure 1.40** Valve guides: (a) removable (b) integral

cylinder block and specially shaped piston crowns. With the latter arrangements, the valves seat directly in the flat underside of the cylinder head.

The top deck of the cylinder head provides a platform for mounting the overhead valve mechanism and, in the case of some overhead camshaft installations, it may be extended forwards to enclose the upper timing drive system. The walls of the cylinder head form the coolant jacket and provide attachment faces for the intake and exhaust manifolds. An upper continuous flange is formed by the walls for mounting the valve cover and serves to raise the sealing joint face above the level of oil draining from the valve mechanism. The coolant outlet connection at the front part of the cylinder head may also serve as a housing for the thermostat, this being described later. Sandwiched between the upper and lower decks of the cylinder head and surrounded by coolant are the inlet and exhaust valve ports. These are necessarily curved and kept as short as possible, the latter to avoid excessive heat transfer both from the coolant to the induction system and from the exhaust system to the coolant. If one port serves two adjacent cylinders it is said to be *siamesed*.

### Valve seats

The valves may seat either directly in the material of the cylinder head or, in harder-wearing rings, inserted therein. Valve seat inserts are usually confined to engines where the cylinder head material is aluminium alloy. Modern valve seat inserts usually take the form of plain rings of greater depth than width, their proportions being such as to confer adequate resistance against distortion (Figure 1.39b). In earlier applications, the more expensive screwed-in type of insert was sometimes used, but it could suffer from inferior heat transfer (Figure 1.39a). For both types it is necessary that they be made an interference fit in the cylinder head. An infrared oven is a feature of the production line for the cylinder head of the Jaguar AJ6 engine, this being used to heat the aluminium alloy cylinder head before the valve seat inserts and valve guides are pressed into place.

Valve seat inserts for aluminium alloy cylinder heads are produced from either a nickel alloy iron casting, or a sintered powder metal the composition of which can be tailor-made to impart the desired hot strength, hardness and corrosion resistance, together with a matching coefficient of expansion. Metal sintering is later explained in Section 1.7 in relation to engine connecting rods.

The use of hard alloy inserts or of local heat treatment to harden the exhaust valve seat areas is also required where cast iron cylinder heads are retained for modern engines designed to run on unleaded petrol. This is because the lubricating effect of lead compounds, formed by the combustion of lead anti-knock additives, previously tended to prevent valve seat recession caused by high temperatures and severe mechanical stress. Although the progressive loss of performance associated with the early stages of exhaust valve recession is hardly likely to be noticed, when the point is reached that valve clearance is lost and one or more valves are prevented from closing properly (Section 1.10), the engine will begin to misfire and potentially serious valve and engine damage can occur.

### Valve guides

Coaxial with the valve seatings are the valve guides, which are carried in bosses extending from inside the valve ports to the top deck of the cylinder head. Their length must be such as to present an adequate bearing surface to resist any side loading on the valve stems, and also to provide a ready path of heat transfer from the exhaust valve head. The working surface of the valve guide must possess low friction to minimize resistance to sliding of the valve stem. They are further arranged to project above the level of oil draining from the valve mechanism on to the top deck of the cylinder head.

The valve guides may be either removable from the cylinder head (Figure 1.40a) or, as is sometimes the case in American practice, cast integrally with it (Figure 1.40b). Separate guides are, of course, required with aluminium alloy cylinder heads. Removable guides are always made an interference fit in the cylinder head material, thereby assisting heat transfer from the valve to the cooling medium, which surrounds their supporting bosses. Grey cast iron is generally the preferred material for the valve guides, although sintered metal may also be used, which is hardened and tempered to match the hardness of the valve stems (as previously mentioned metal sintering is later explained in Section 1.7). It may be of interest to add that cast iron inlet and phosphor-bronze exhaust valve guides were specified at the design stage of the Rolls-Royce V8 engine, the latter material being chosen to avoid valve stem scuffing.

### Cylinder head materials

The material used for cylinder head construction is either cast iron or aluminium alloy, as in the case of the cylinder block and crankcase. In favour of the much widely used cast iron cylinder head is its greater rigidity and better noise damping properties. However, apart from effecting a saving in weight, the greater heat conductivity of aluminium alloy is beneficial in maintaining a more uniform temperature throughout the cylinder head. This may sometimes permit the compression ratio of an engine to be raised slightly without incurring detonation or knocking. An aluminium alloy head has often been used in combination with a cast iron block and crankcase, and it is usually considered essential for air-cooled engines to ensure efficient heat transfer.

A service difficulty sometimes encountered with aluminium alloy cylinder heads is that they can prove obstinate to remove if corrosion deposits build up around their fixing studs, and it is for this reason that some designers specify setbolts. Owing to the greater thermal expansion and contraction of an aluminium alloy head, a smoother surface finish is generally specified for its mating face to prevent a ratcheting action against the cylinder head gasket.

### Tightening down the cylinder head

Before carrying out this operation it is always advisable to consult the particular manufacturer's service instructions (Figure 1.41), especially in respect of the following:

1 Check whether the screw threads and washer faces require lubrication, and if so the type of lubricant to be used. The point here is that the presence or otherwise of a lubricant on screw-thread assemblies will affect the clamping load they exert for any given value of torque tightness.
2 Establish the correct sequence of nut or setbolt tightening, the number of stages in which it is to be achieved and the final torque value to be attained. For improved accuracy in tensioning the cylinder head fastenings a turn-of-nut method (as it is known to structural engineers) may be used in conjunction with initial torque tightening. Following the latter, the setbolts are marked on their heads with paint spots that all face in the same direction; then each nut is given a further specific amount of turn, perhaps in two stages of a quarter-turn each. With the setbolts equally tightened the paint spots will all be facing in the same but of course new direction. Whichever tightening procedure is used, the numerical sequence of tightening specified usually involves starting from the centre and working alternately towards each end.

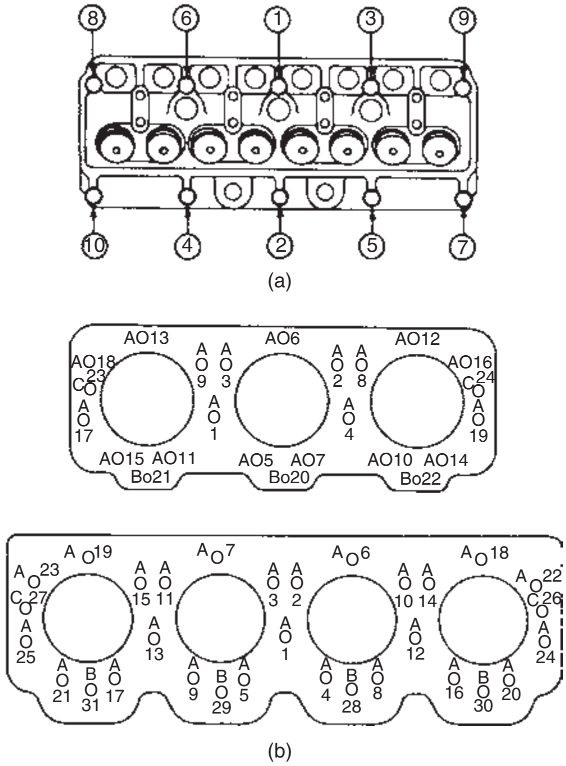


**Figure 1.41** Examples of cylinder head nut or bolt tightening sequence: (a) car petrol engine (*Toyota*) (b) commercial vehicle diesel engines (*Gardner*)

Any retightening of the cylinder head after the engine has been run should be done strictly in accordance with the manufacturer's recommendations. Its purpose is to compensate for slight settling of the gasket material after initial exposure to heat and vibration and therefore restore the required sealing pressure. The consequences of incorrect tightening of the cylinder head can prove very expensive to rectify. They may include at least coolant and compression leaks via the head gasket, and at worst the failure of screw-thread fastenings and distortion of the cylinder bores.

### Cylinder head gasket

Although the mating faces of the cylinder head and block are machined smooth, flat and parallel, in reality there are always minute surface irregularities and structural deflections to be accommodated. A static seal or head gasket is therefore required, which possesses the necessary degree of both plasticity (pliability) and elasticity (springiness). The effect of tightening down the cylinder head on to the block places the gasket under compression, so that sufficient friction is created between the sealing surfaces to resist extrusion of the gasket by cylinder gas pressure.

However, there is a great deal more to gasket application than this may suggest, because varying amounts of sealing pressure between the cylinder head and block are required at different locations. Sealing against the combustion gases clearly demands far higher unit pressures than are required to seal against the passage of coolant and lubricant through the gasket, while on the other hand very high unit loads are

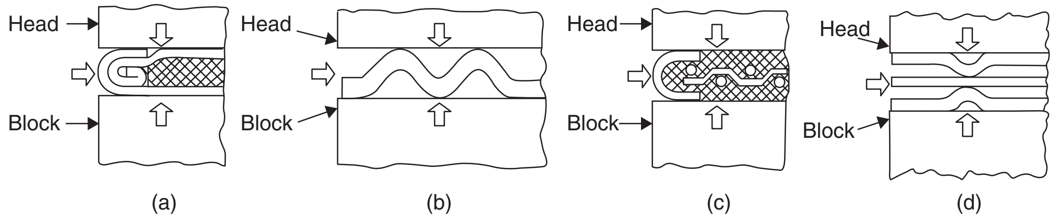


**Figure 1.42** Types of cylinder head gasket: (a) metal and asbestos layers (b) embossed steel sheet (c) metal-cored composition materials (d) multi-layer steel

concentrated on those areas immediately surrounding the cylinder head studs or setbolts. The gasket must therefore be designed so that its distribution of sealing pressure conforms to an optimum pattern for any given engine. This requirement has tended to become even more demanding with modern engine design, because not only has there been a trend towards increased peak combustion pressures, but also in the interests of saving weight the structural rigidity of the cylinder block may be compromised, which makes clamping loads more difficult to control.

Apart from maintaining a seal by compensating for the non-uniform loading between the cylinder head and block, the gasket material must also be able to withstand the intense heat of combustion, the penetrating effect of the coolant and the chemical effect of the lubricant. It is for these reasons that the cylinder head gasket has always been regarded as the most critical sealing application on any engine.

*Types of cylinder head gasket*

In earlier practice, cylinder head gaskets for engines of moderate size and power were traditionally of sandwich construction. They compromised a centre layer of asbestos millboard between two sheets of either copper or tin-plate steel, together with eyelet reinforcements around the combustion chamber openings, which protected the edge of the gasket and increased the sealing pressure in these regions (Figure 1.42a). The embossed steel sheet type of gasket was later introduced for heavier-duty applications, because it better retained cylinder head torque tightening owing to its all-metal construction (Figure 1.42b). A plastics resin coating enabled it to accommodate the minute irregularities of the mating surfaces. By the early 1970s a different type of gasket construction had gained acceptance, which comprised a metal core of tanged or perforated steel sheet faced with fibre composition materials such as asbestos and plastics (Figure 1.42c). In a later version the fibre material facings are bonded to a plain steel sheet to provide a laminated construction, instead of being clinched to the metal core. Eyelet reinforcements around the cylinder bore areas are again used in both forms of composite gasket.

Since exposure to asbestos is now recognized as being a health hazard (and is a topic that will receive further mention in connection with friction materials), the manufacture of non-asbestos cylinder head gaskets has now become established. Gaskets of this type are typically based on a steel core with either aramid fibre, glass fibre or graphite facings with zinc-coated or tin-plated steel beading for protecting their sealing edges.

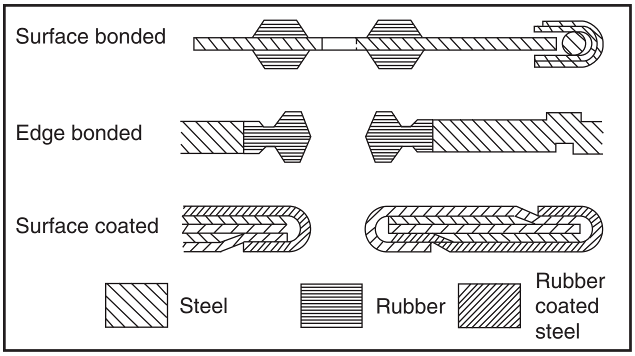


**Figure 1.43** Steel and rubber cylinder head gasket constructions (*Cooper Payen T&N*)

A further development has been the introduction of steel and rubber gasket constructions. Three types of steel and rubber gasket that have been developed by Cooper Payen Ltd are shown in Figure 1.43. In the surface-bonded type a rubber bead is bonded on to a stainless steel core around the oil and water ways. A wire and ring eyelet assembly provides the cylinder bore sealing system. Since the rubber requires relatively little load to seal effectively, the surface-bonded gasket confers effective sealing with moderate overall clamping load. It is specified primarily for petrol engine applications. The edge-bonded type comprises a relatively thick steel plate, its edge being made sufficiently wide to allow the direct bonding of a rubber grommet, which provides sealing against oil and coolant. As this type of gasket is associated with medium to heavy diesel engine applications, the cylinder bore sealing system is generally obtained by deformation of the steel plate between the cylinder head and liner protrusion. The surface-coated type is of laminated construction, which incorporates several laminations of rubber-coated and plain steel, the rubber coating providing interlaminar and surface sealing against oil and coolant. Another feature of this particular construction is that it allows a preferential thickness of the gasket in the cylinder bore area. The various types of elastomer used with these steel and rubber gaskets, in ascending order of temperature requirements, are nitrile butadiene, silicone and fluorocarbon.

Since the late 1990s the multi-layer steel or MLS type of cylinder head gasket has gained widespread application. It is constructed from two or more layers of steel laminate, one or more of which will include embossed sealing beads (Figure 1.42d). The gasket may also receive a special surface treatment. In a typical application for an engine with an aluminium cylinder block and head, a 0.5 mm (0.02 in) ultra-thin two layer gasket may be specified. The advantages claimed for using this type of cylinder head gasket are those of improved sealing, enhanced reliability and reduced crevice or 'dead' space. This becomes possible because the thinness of the gasket allows it to be made closer fitting around the tops of the cylinder bores. By reducing crevice space here the accumulation of unburned gases and hence their contribution to hydrocarbon emissions is discouraged.

*Cylinder head gasket misbehaviour*
A cylinder head gasket that is leaking or 'blowing' may do so in various ways, as follows:

*Internal leaks* Between two adjacent cylinders, as evidenced by weak compression pressure in both; through to the coolant jacket, causing bubbling in the radiator and loss of coolant; and through to the coolant jacket, allowing coolant into the cylinder to cause misfiring and steam issuing from exhaust.
*External leaks* Outwards from the joint to produce a spitting noise when opening the throttle.

## Cylinder liners

During an earlier era of rapid bore wear, cylinder liners were quite widely used either as original equipment or as an overhaul feature. This was because the particular grade of iron from which they were cast centrifugally could be selected for its wear-resistant properties, rather than for the free-flowing characteristics required of an iron for casting the cylinder block and crankcase. However, later developments in the fields of piston ring coatings, lubricating oil formation, oil and air filtration equipment and cooling system control have all combined to minimize cylinder bore wear, so that the need for detachable liners on this score seldom arises. Cylinder liners are now generally specified either to provide a suitable wear-resistant surface for the cylinders of aluminium alloy engines, or to simplify the production of cast iron engines by permitting an open-deck form of cylinder block.

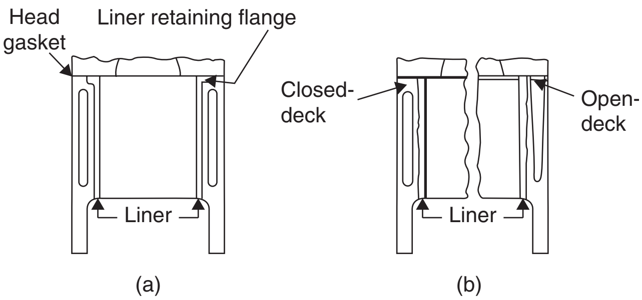


**Figure 1.44** Dry cylinder liners

*Dry cylinder liners*
A detachable dry liner takes the form of a plain or a flanged sleeve (Figure 1.44a), the entire outer wall of which is maintained in intimate metal-to-metal contact with the cylinder block. This is of closed-deck construction and may be of either cast iron or, less commonly, aluminium alloy.

Non-detachable dry liners have been cast integrally with aluminium alloy cylinder blocks of both closed- and open-deck constructions (Figure 1.44b). In a recent V eight-cylinder engine of aluminium alloy construction produced by General Motors, the cast iron liners are retained in their respective positions while the molten alloy is injected into the die cavity surrounding them. Modern cast-in iron liners have a wall thickness that is typically in the region of 3 mm (0.12 in).

Dry liners generally contribute to the rigidity of the cylinder block, but tend to introduce a barrier to heat flow at the adjoining surfaces. This effect is minimized where the cylinder block is made from aluminium alloy, as a consequence of its good heat conductivity.

*Wet cylinder liners*
The wet type of liner always takes the form of a flanged sleeve, the outer wall of which is largely exposed to the coolant in the cylinder jacket. It may be incorporated in both closed- and open-deck cylinder block constructions. Clearly, the wet cylinder liner is better cooled than the dry type and can more easily be renewed when worn, as will be explained later. It contributes little to the rigidity of the cylinder block, however, and there is always the possibility that coolant leaks into the crankcase may occur.

Two distinct methods of locating wet liners may be used, according to whether they are being installed in closed- or open-deck cylinder blocks, as follows:

*Closed-deck* The cylinder liner is provided with a top flange only and is suspended through the coolant jacket from where it is clamped between the cylinder head and the upper deck of the cylinder block (Figure 1.45a).

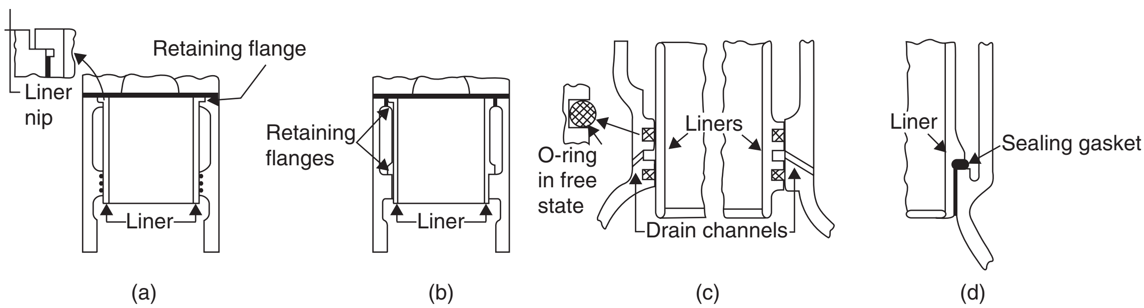


**Figure 1.45** Wet cylinder liners in (a) closed-deck and (b) open-deck cylinder blocks. Sealing arrangements for (c) suspended wet liners and (d) wet liner held in compression

*Open-deck* Here the cylinder liner must be provided with a top and a lower flange and it is held in compression within the coolant jacket between the cylinder head and the lower deck of the cylinder block (Figure 1.45b).

An advantage of the first arrangement is that the cylinder block is relieved of stresses that would otherwise be imposed by the axial expansion of the liner upon heating. With the second arrangement, the less intrusive top flange generally permits better cooling around the upper part of the liner.

For closed-deck cylinder blocks, two oil-resistant (synthetic) rubber O-ring seals encircle the lower part of the liner and are deformed into grooves where it passes freely through the lower deck of the cylinder block. The sealing rings may be grooved into either the cylinder block or the liner itself (Figure 1.45c shows both forms), and a third unfilled groove between them communicates with a drilling in the block that leads to atmosphere. This drilling serves as a drain channel for any coolant and, similarly, oil that may have seeped past the top and bottom sealing rings, respectively.

With open-deck cylinder blocks, a compression sealing gasket is generally used between the flange towards the bottom of the liner and its seating in the lower deck of the block (Figure 1.45d).

In both types of liner installation the cylinder head gasket completes the sealing arrangements for the top end of the liner.

A compromise arrangement of cylinder liner is the so-called 'damp liner', which is part wet and part dry. The upper part of the liner is made thicker in section than the lower part, so that it acts as a wet liner, while the thinner lower part is made a slip fit in the cylinder block and is therefore dry. Since a step is formed between the upper wet and lower dry sections of the liner, it is retained by being nipped between the cylinder head and the abutment in the cylinder block on which the step rests. Although this construction results in a shorter coolant jacket, the direct cooling is nevertheless concentrated where it is most needed in the higher temperature region of the cylinder. There is, in fact, a tendency to shorten the coolant jacket in modern design, because it not only contributes to the rigidity or stiffness of an aluminium alloy cylinder block, but also accelerates the warm-up process to assist emission control.

*Cylinder liner installation*

It will be evident from the previous descriptions that the dry liner is usually (although not invariably) made an *interference* fit in the cylinder block. Typically, the block is bored out to provide an interference fit of 0.06–0.09 mm (0.0025–0.0035 in) between the cylinder and the liner, which will then need to be lubricated and pressed in under a load of about 2000–3000 kg (2–3 tons) (Figure 1.46). To avoid any possibility of liner bore distortion, the cylinder block studs are usually refitted before the liner bore is honed to final size.

The production method of installing dry liners into the aluminium alloy closed-deck cylinder block of the Jaguar AJ6 engine is of interest, since it does not involve the use of a press. A two-stage infrared oven is used to heat the entire cylinder block for three minutes at a time in each stage, so that when it emerges from the oven the bores have expanded sufficiently for the liners to be slid into position by hand. Immediately all six liners are in place, the cylinder block enters a special cooling tower where its cooling is rigidly controlled.

In contrast, wet liners are generally made a *slip fit* in the cylinder block (Figure 1.47). A typical cylinder liner-to-block clearance would be 0.05–0.15 mm (0.002–0.006 in). Even so, a manufacturer may recommend that the cylinder block be preheated, so that there is no hindrance to correct insertion of the liners and seals. There is always the danger that if an engine with wet liners is cranked over with the cylinder head removed, the liners could be dragged clear of their locations by the rising pistons. The temporary fitting of retaining clamps on the liners is therefore the safest practice in these circumstances.

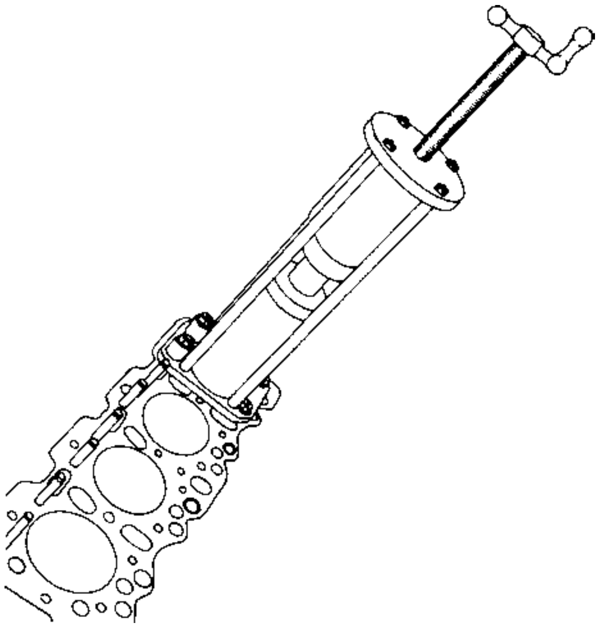

**Figure 1.46** Installing a dry cylinder liner with a Flexi-Force hydraulic press (*Brown Brothers*)

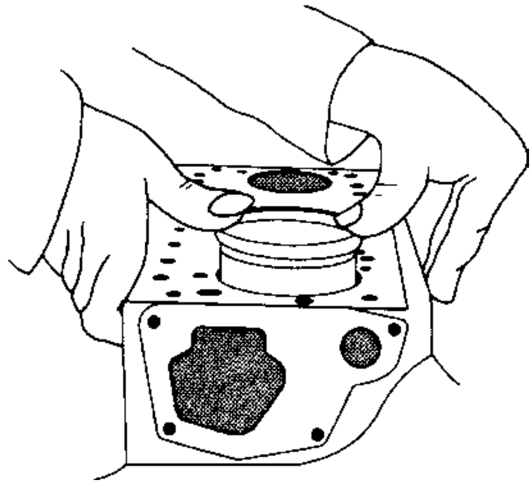

**Figure 1.47** Inserting a wet cylinder liner by hand (*Perkins*)

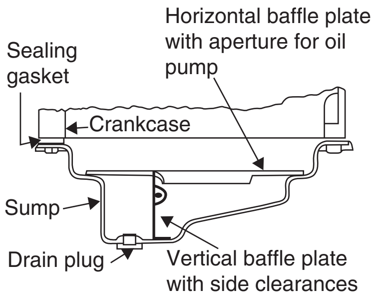


**Figure 1.48** Crankcase sump

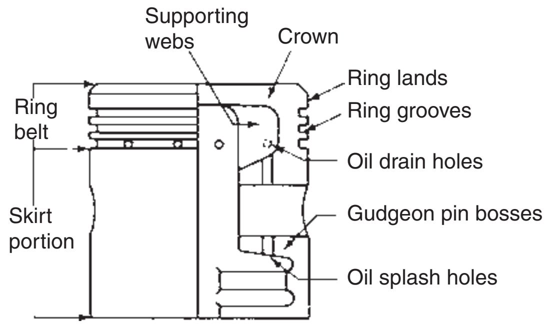


**Figure 1.49** Piston construction and nomenclature

A final consideration in fitting flanged liners, either wet or dry, is the provision of a small amount of *nip*. This refers to the amount the liner top flange protrudes above the top deck of the cylinder block, so as to promote an efficient gasket seal when the cylinder head is clamped on to it (Figure 1.45a). In practice, a liner nip of 0.05–0.12 mm (0.002–0.005 in) is fairly typical.

### Crankcase sump or oil pan

This unit acts as a reservoir to store the oil that is required by the engine lubrication system. It further serves as a vessel in which any sludge, water and metal particles in the oil can settle out, and also provides an opportunity for any entrained air to escape from the oil.

The sump is either of pressed steel construction, or produced from an aluminium alloy. In its latter form it better contributes to the rigidity of the crankcase and also assists with heat dissipation. Attachment of the sump to the crankcase is usually by means of setscrews through mating flanges, between which is sandwiched a flexible packing or gasket.

Baffle plates are normally fitted in the sump to minimize both oil surging and agitation, the former arising from the changing motion of the car and the latter from the oil flung from the crankshaft bearings (Figure 1.48). A screwed plug is incorporated at the lowest point in the sump for draining the oil in service.

In a recent Japanese application the oil sump is of two-piece construction, comprising an upper cast aluminium section that also houses a harmonic balancer system chain driven from the crankshaft, and a lower stamped steel section.

## 1.7 PISTONS AND CONNECTING RODS

### Piston construction and nomenclature

The piston crown may be either flat topped or specially shaped in order to conform to the particular design of combustion chamber of which it forms one wall. Combustion loads are transmitted directly from the crown to the gudgeon pin bosses through intermediate supporting webs, which also facilitate the flow of heat to the encircling piston rings and thence to the cylinder walls. The ring belt immediately below the crown is thus largely relieved of loads that would otherwise tend to deform its grooves. Any closing in of the grooves would, of course, prevent the free radial movement of the rings and thus impair their sealing ability.

The main part of the piston below the ring belt is termed the skirt, and this is made as close fitting as practicable in the cylinder, thereby ensuring quiet operation and the maintenance of the rings at their most favourable attitude to the cylinder wall. Furthermore, the skirt must present an adequate bedding area to the cylinder wall, not only to minimize contact pressure, but also to assist with heat dissipation. It should be appreciated though that the piston skirt is not normally in direct contact with the cylinder wall, but is separated from it by a film of oil. Recessed flats may be provided at the termination of the gudgeon pin holes.

Modern practice favours the use of a *solid skirt* piston of rigid construction, because of the high combustion loads now encountered (Figure 1.49). Basically, its advantages are that it can be made thinner in section to withstand a given loading, so that it affords a saving in weight. It does, however, need a good deal of modification to provide acceptable expansion control of the skirt, as will be explained later. Another type of skirt which was once widely used is that known as the *split skirt* (Figure 1.50a). This incorporated a near-vertical slot extending from the centre of an upper horizontal slot down to the base of the skirt on the non-thrust side of the piston. Here it should be explained that the thrust side of the piston reacts against the side force arising from the angular motion of the connecting rod on the power stroke, while its non-thrust side reacts against the lesser side forces on the compression and exhaust strokes. (The two sides are also known as the major and minor thrust faces.) The split skirt piston was originally introduced to provide quiet running and, by virtue of its skirt flexibility, to accommodate a certain degree of cylinder bore distortion where this was prone to occur.

### Piston materials and expansion control

In low-speed engines of early design, the material from which the pistons were made was cast iron to match that of the cylinders. With increasing engine speeds and output, however, it has long since become established practice for the pistons to be cast from aluminium alloy, materials of this type combining lightness in weight with high thermal conductivity. They have a moderate silicon content so that their mechanical strength is better maintained at high operating temperatures, which may now exceed 300°C, whilst their coefficient of thermal expansion is lower than that usually associated with aluminium alloys (Section 2.4). The addition of silicon also increases their resistance to corrosion and wear.

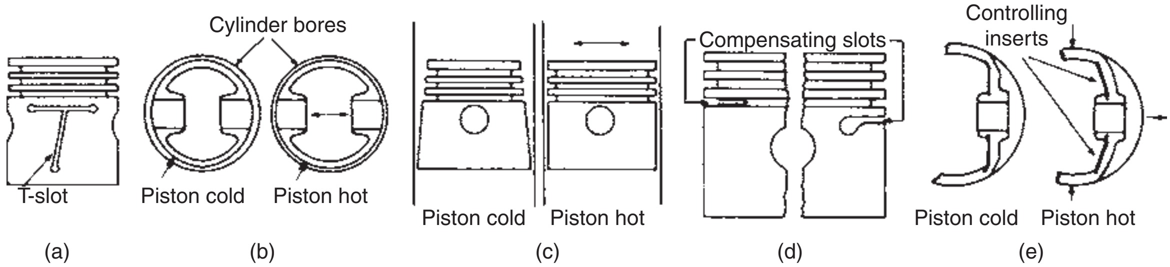


**Figure 1.50** Methods of expansion control for pistons. (a) split skirt piston with compensating T-slot. Solid skirt pistons with (b) ovality (c) taper (d) compensating slots (e) controlling inserts

A more recent development has been the *squeeze cast* piston, which was introduced by AE Piston Products in the late 1980s. The squeeze casting technique retains the forming capabilities inherent to the casting process, while demonstrating the strength and integrity of forging. In practice a predetermined amount of molten aluminium is poured into the die, which is then closed and applies pressure to the casting as it solidifies, hence the term squeeze casting.

The associated advantages that accrue from this process include the following:

1 More economic use of aluminium, because the risers and runners needed for gravity diecasting are no longer required.
2 Porosity is minimized by the application of pressure, which compensates for the normal shrinkage that accompanies cooling and solidification of the molten aluminium.
3 Greater refinement of material structure, which arises from the increased heat flow at the interface where the aluminium is squeezed against the die.
4 Improved accuracy and surface finish, which again derives from the more intimate contact between the aluminium squeezed against the die.
5 Alternative materials can be readily introduced into the casting, such as the expansion control inserts or struts mentioned below and piston ring groove carriers (Section 2.4), with freedom from microporosity and hence improvement in the integrity of the bond.
6 A high top ring for better emission control (as explained later) can also be accommodated, by virtue of the enhanced material properties and strength of the squeeze cast piston.

Although the aluminium alloys chosen for the piston material expand less when heated than most other light alloys, they nevertheless expand at nearly twice the rate of the cast iron used for most engine cylinders. In the absence of special expansion control features, this relative difference in expansion rates would result in seizure of the piston when hot, and could be avoided only by tolerating a fitting in the cylinder when cold that would be too slack and would result in piston slap.

A basic method of controlling thermal expansion is to machine the piston to a special form, which is both oval in contour and tapered in profile (Figure 1.50b and 1.50c). The ovality is such that when the piston is cold, the minor axis of the skirt lies in the direction of the gudgeon pin. When the piston is hot the skirt then assumes a circular shape, because of the greater expansion occurring in the mass of metal comprising the gudgeon pin bosses.

The direction of taper allows for additional clearance when cold near the top of the skirt, since this part ultimately attains a greater running temperature and thus expands more than the cooler running lower portion. Immediately below the ring belt, where the skirt temperature is greatest, the degree of taper may be intensified to produce an overall barrelled shape. Similarly, the clearances around the ring lands are progressively increased towards the piston crown, thereby avoiding contact when hot with the cylinder wall.

In order that the piston can be made as close fitting as possible in the cylinder, either compensating slots or controlling inserts or both may be incorporated in its construction. With split skirt pistons, circumferential expansion of the skirt is simply absorbed by temporary closing in of the near-vertical compensating slot. For solid skirt pistons it is usual for part-circumferential slots to be located within or, in some designs beneath the oil control ring groove (Figure 1.50d). The angular disposition of these horizontal slots is such that the flow of heat from the piston head is diverted from the thrust faces of the skirt, thereby reducing expansion across them.

In modern practice, widespread use continues to be made of controlling inserts or struts, which are stamped from alloy steel and cast parallel with the piston axis and integrally around each gudgeon pin hole. Each insert is pre-heated and placed in the die before pouring; no bond is attempted as they are not intended to add strength to the piston. The inserts are covered with aluminium only on the side away from the piston axis, so that they act in the manner of bimetallic strips when the piston is heated. Since the steel inserts have a lower coefficient of expansion than the aluminium piston, they are compelled to bow outwards and thus restrain expansion of the skirt across the thrust axis (Figure 1.50e). It is perhaps of interest to recall that this principle of controlling thermal expansion of the piston skirt was originally incorporated in the American Nelson-Bohnalite piston of the early 1920s, which was also widely used elsewhere. It was regarded by one prominent automobile engineer of the time as being 'the first piston to have been designed with an intelligent appreciation of the difficulties to be overcome'.

To ensure freedom from piston scuffing on cold starting or hot running conditions, it is important that the surface finish of the piston skirt is oil retentive. Various surface treatments may be applied to aluminium alloy pistons, including tin plating to assist the running-in process, graphite coating to avoid scuffing under borderline conditions of lubrication, and PTFE coating (Section 13.2) to reduce power losses through friction. Further developments in the surface treatment of pistons for modern high-performance engines include molybdenum spraying of the skirt, and nickel coating of the piston to combat erosion damage where it occurs in engines that are continuously operated close to their knock limit (Section 3.2).

The fitting clearance for a piston must always be sufficient to maintain an oil film during engine operation and thereby prevent scuffing or seizure of the piston in the cylinder bore. It rarely exceeds 0.05 mm (0.002 in) measured at the top of the skirt across the thrust axis, although it is unlikely to be less than this in the case of turbo-charged engines. The fitting clearance should always be in accordance with the manufacturer's specification. If an engine is fitted with wet cylinder liners, it is usual for matched liners and pistons to be made available in service.

## Gudgeon pins and their location

The gudgeon pin is the vital mechanical link that hinges the piston to the connecting rod and, although it is of deceptively simple appearance, it must be recognized as being a precision engineered component. This is because it has to satisfy several conflicting requirements; namely, it must combine strength with lightness, be close fitting but with freedom to move, and resist wear without promoting scuffing. The gudgeon pin is of hollow construction and typically produced from a fine-grained plain carbon steel with controlled hardenability. It is lapped to a mirror finish of 0.05–0.10 (2–4 $\mu$in). The diameter of the gudgeon pin may be up to 40 per cent of the piston diameter, so that maximum bearing pressure in the piston bosses does not exceed 55 MN/m$^2$ (8250 lbf/in$^2$). Under load its ovality and longitudinal bending are not expected to exceed 0.025 mm (0.001 in) and 0.075 mm (0.003 in) respectively. The methods used for gudgeon pin location depend on whether the arrangement is a semi-floating or fully floating one.

*Semi-floating gudgeon pin*

This is held rigidly in the connecting rod eye and oscillates only in the piston bosses (Figure 1.51a). A semi-floating gudgeon pin is matched by grading to give a clearance of 0.0075–0.0125 mm (0.0003–0.0005 in) in the piston pin holes. Current practice is for the gudgeon pin to be retained by an interference fit, rather than by clamping it in a split eye with a pinch-bolt as in earlier designs. The latter method introduced a discontinuity into the connecting rod eye that could be a source of weakness. Two advantages of the semi-floating arrangement are that there is less length of gudgeon pin subject to bending loads, and it eliminates one potential source of noise that can otherwise develop in the small-end bearing of a fully-floating arrangement.

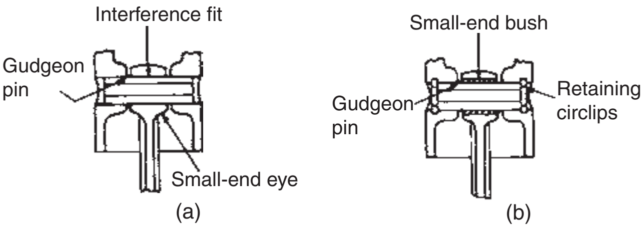


**Figure 1.51** Methods of gudgeon pin location: (a) semi-floating (b) fully-floating

*Fully-floating gudgeon pin*

This is able to oscillate not only in the piston bosses, but also in the connecting rod eye that is bushed for the purpose, as described later. In order to prevent it from escaping sideways and contacting the cylinder wall, the fully-floating gudgeon pin must be located axially in the piston bosses. This is because it is free to oscillate both in the connecting rod eye and the piston bosses at normal operating temperature. Location is usually provided by spring retaining rings or circlips, which are expanded into grooves near the outer end of each boss and thus act as removable shoulders (Figure 1.51b). A fully floating gudgeon pin is matched by grading to give a clearance of 0.0025–0.0075 mm (0.0001–0.0003 in) in the piston pin holes.

*Offset gudgeon pin*

When the power stroke begins, the piston is forced by a combination of gas pressure and connecting rod angularity to move laterally across the cylinder; this effect is more pronounced with shorter centre distances for the connecting rod, as found in modern engines. The manner in which this movement occurs from the minor to the major thrust side can be such as to cause piston knocking. To minimize noise from this source the axis of the gudgeon pin can be slightly offset from that of the piston and in the direction of the major thrust side. The effect is to tilt the piston during the compression stroke, so that contact is first established between the lower part of the skirt and the major thrust side of the cylinder. At the beginning of the power stroke it then only remains for the upper part of the piston to move across the cylinder to establish full skirt contact with the major thrust side, this movement being beneficially damped by the friction of the rings in their grooves. The amount of gudgeon pin offset used is generally in the region of 1.5 mm (0.06 in) but it can also be influenced by piston clearance and a 'listening session' on the engine development test bed. A stiffer design of piston skirt has sometimes been necessary where an offset gudgeon pin is used.

## Types of piston ring and nomenclature

There are two basic types of piston ring used in petrol and diesel engines. These are designated compression and oil control rings, and in each case the nomenclature is most conveniently presented by illustration (Figures 1.52 and 1.56).

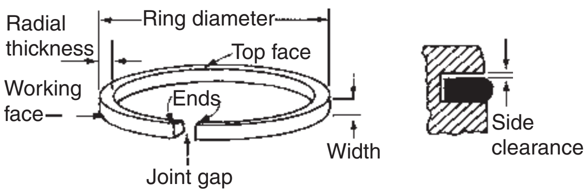


**Figure 1.52** Piston ring nomenclature

For modern petrol engines two compression rings and one oil control ring are used on each piston. In earlier practice it was customary for more than three rings to be used. For example, the engines with a fairly low compression ratio used in Rolls-Royce cars of the late 1920s originally had five-ring pistons. The basic reason why fewer rings can now be used is that the higher mean cylinder pressures better augment the sealing action of the rings. Various forms of compression ring may be specified, the differences between them lying in their cross-sectional shapes and the application of wear-resistant surface treatment. Top compression rings are usually of plain rectangular section, their inner and outer edges being slightly chamfered to prevent sticking in the groove (Figure 1.53a). The working surface of the ring may also assume a barrel form, instead of being flat and parallel to the cylinder wall, so that it is better able to accommodate any slight piston rock where the skirt length may be limited (Figure 1.53b).

The second compression ring principally serves to reduce the pressure drop across the top ring, and it can with advantage be made relatively more flexible so as also to assist oil control. Various departures are therefore made from the basic rectangular cross-section, the main object being to compensate for torsional deflection of the ring under combustion pressure, so that top edge contact with the cylinder wall is avoided. Otherwise, the ring tends to pump oil towards the combustion chamber and therefore opposes the action of the oil control ring. To reverse this effect taper-faced and stepped torsional rings are widely used, the two features being combined in some designs (Figures 1.54 and 1.55). Another form of second compression ring with efficient oil control properties is the Napier type, which has a reduced width of taper face with a hook-like undercut below, this being where the bulk of the oil has to be controlled.

The basic slotted type of oil control ring is simply a modification of the plain rectangular-section compression ring. It takes the form of an outward-facing channel section from which collected oil escapes through slots machined radially to the back of the ring (Figure 1.56a). The oil then returns to the sump via communicating holes drilled through the piston wall at the back of the groove. A fairly high radial pressure is exerted against the cylinder wall by virtue of the narrow working surfaces of the ring, so that any tendency for it to ride over the oil film is counteracted.

Oil control requirements have become more exacting for modern engines because the use of high compression ratios increases the depression acting on the rings during the induction stroke, especially on the overrun with a closed throttle. To improve oil control under these conditions various forms of composite rings are commonly used (Figures 1.56b and c). They generally feature two or more independent working faces, comprising flexible rails with rounded wiping edges, which act in conjunction with a spring expander. This presses the rails axially against the sides of the ring groove and radially against the cylinder wall. Since the flexible rails can act independently of one another, they do not lose their effectiveness in the presence of any piston rock.

### Piston ring materials and expansion control

Compression rings and slotted oil control rings are produced from descending grades of cast iron, because this material provides a satisfactory wear-resistant surface and retains its elasticity at high temperatures. Composite oil control rings for petrol engines utilize thin steel rails with chromium-plated

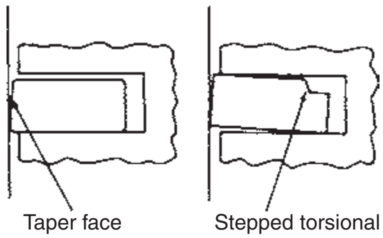


**Figure 1.54** Second compression rings

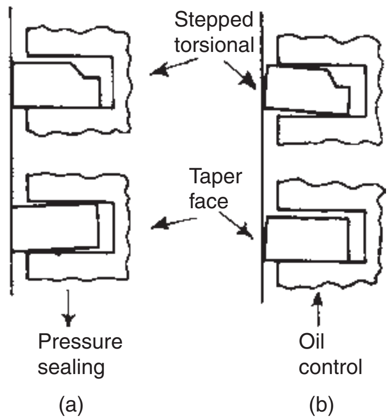


**Figure 1.55** Action of second compression rings

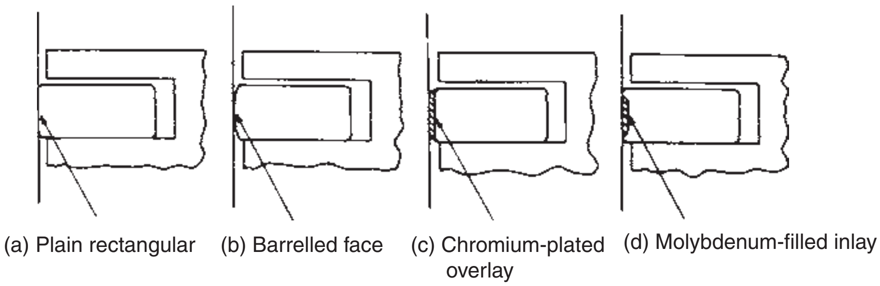


**Figure 1.53** Top compression rings

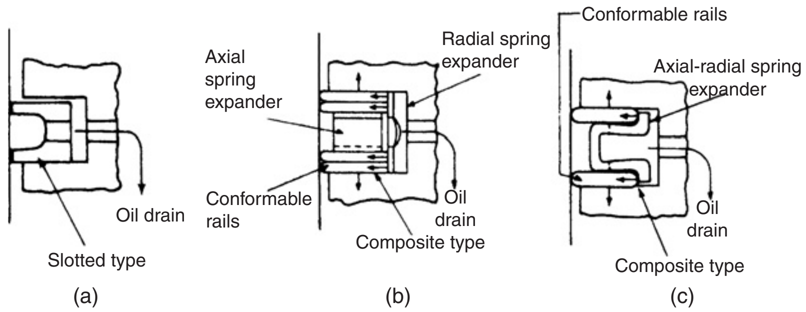


**Figure 1.56** Oil control rings: (a) slotted type (b) and (c) composite types

working edges, whilst in the spring-loaded conformable ring used in diesel engines a cast iron slotted ring is expanded by a stainless steel backing spring.

Since the top ring not only is subject to the immediate effects of combustion pressure and temperature, but also must withstand abrasive and corrosive conditions, a wear-resistant coating is now generally applied to its face. This may comprise either a hard chromium-plated overlay or a molybdenum-filled inlay (Figures 1.53c and d). The former treatment was actually first applied to piston rings of military vehicle engines during the African campaign in World War II, and is especially beneficial in resisting abrasive and corrosive wear. Molybdenum inlay was a later development for piston rings, following its introduction on Chevrolet and Cadillac engines in 1963, and is particularly resistant to scuffing wear. Similar benefits are, of course, conferred on the cylinder wall itself.

A piston ring must be provided with a gap at one point on its circumference for three reasons:

1 So that it can be expanded over the piston head and then released into its groove.
2 To allow it to be compressed into the cylinder and thus exert an initial sealing pressure, which is then greatly augmented by gas pressure when the engine is running.
3 To accommodate circumferential expansion of the ring when it is hot.

With regard to point 3, although the coefficients of expansion of the cast iron and alloy steels from which the piston rings are made relate to that of the cylinder material, the circumferential expansion of the rings is greater because they attain a higher mean temperature, otherwise there would be no heat flow from one to the other. It is also for this reason that a piston ring, which maintains a uniformly distributed pressure against the cylinder when cold, is not likely to do so when hot, and this has therefore to be taken into account during manufacture.

The fitting clearances for piston rings are generally as follows:

*Piston ring closed gap* To compensate for circumferential expansion this should usually be not less than 3/1000 of the cylinder bore diameter, and is measured with the ring installed in its cylinder. In practice, a gap of between 0.30 and 0.35 mm (0.012 and 0.014 in) is likely to be specified.

*Piston ring side clearance* To allow for free radial movement of the ring in its groove this should usually be not less than 0.035 mm (0.0015 in) for cylinder diameters of between 50 and 100 mm (2 and 4 in) (Figure 1.52).

## Piston ring misbehaviour

This is indicated by loss of cylinder compression, poor oil control and noisy operation of an engine. These complaints can result from wear, sticking or breakage of a ring, although the fault may not necessarily lie with the ring itself. The faults can be summarized as follows:

*Abrasive wear* This has been much reduced by modern air filtration equipment and closed-circuit crankcase ventilation, which prevent airborne dust particles from entering the cylinders and the crankcase, thereby protecting the compression and oil control rings, respectively. Without some abrasive wear the piston rings would never, of course, bed in at all!

*Scuffing wear* A definite explanation for this particularly severe type of wear has long been the subject of engineering research (since it can also occur, for example, between gear teeth), but it is generally thought to result from the formation and tearing of tiny welds on the sliding surface of the ring, under the effects of pressure and temperature. If it occurs it does so rather suddenly, typically during the running-in of an engine.

*Corrosive wear* This is attributed to chemical attack by the products of combustion and occurs when these are able to condense on the ring surfaces at low temperatures. It is now minimized by better regulation of the cooling system, so that the engine warms up more rapidly and takes longer to cool down, and also by applying various surface treatments to the ring face.

*Sticking* This occurs when the ring groove temperature becomes excessive and causes breakdown of the lubricating oil reaching the ring, so that solid products build up in the groove and prevent free radial movement of the ring. The role played by modern lubricating oils in protecting against this condition will be referred to again later.

*Breakage* Apart from the breakage that can result from the sticking of a ring, another condition known as ring flutter may cause the breakage of rings in all cylinders if an engine is persistently over-revved. Following experiments made in the late 1940s by Dr P. de K. Dykes at Cambridge University, England, it is now recognized that if the reciprocating inertia

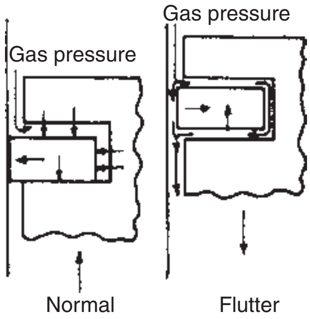


**Figure 1.57** Piston ring sealing action

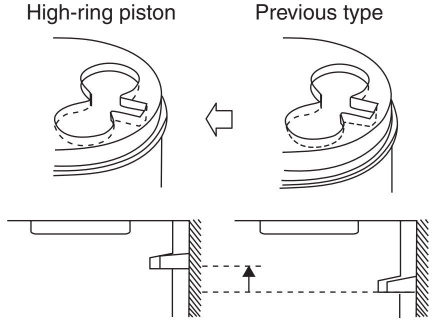


**Figure 1.58** Comparison between previous and high-ring pistons (*Toyota*)

force acting upwards on the ring exceeds the gas pressure that is forcing it downwards and outwards, then during the end of the compression stroke and the beginning of the power stroke the ring will lose contact with the lower side of its groove (Figure 1.57). The consequent release of gas pressure from behind the unseated ring results in it collapsing radially inwards, and this can lead to failure.

## Further developments in piston and rings

Since the normal operation of the pistons and their rings accounts for about 75 per cent of the total friction losses in an engine, it is perhaps to be expected that more recent developments have sought to reduce these losses in the interests of improving fuel economy.

To reduce the viscous friction that is caused by shearing of the oil film between the piston and its cylinder during operation, there is a trend towards lessening the contact area of the piston skirt, preferably without shortening the skirt as this could result in undesirable tilting of the piston and unsatisfactory ring behaviour. This conflict has been resolved in the case of the AEconoguide type of piston by specially contouring the skirt so as to form three low-area contact pads on each side.

The advent of direct fuel injection for passenger car petrol engines (Section 6.9) can lead to a requirement for relatively complex shaped piston crowns. Since these tend to raise the centre of gravity of the piston, this could be detrimental to its quiet operation, unless countered by an adequate length of skirt to restore accurate guidance. Hence, the piston can become heavier for this type of application. Also, the presence of a bowl-like depression in the piston crown can lead to a concentration of thermal stresses around the rim of the bowl and any projections therefrom (similar to that encountered in the piston crowns on direct-injection diesel engines). This therefore has to be taken into account during the design and manufacturing processes to avoid crack formation.

A reduction in friction of the piston rings, which receive uncertain lubrication especially when sliding against the upper cylinder wall, can be attributed to the recent Japanese development of the two-ring piston, which carries one plain compression ring and one three-piece oil control ring. Associated advantages include a decrease in piston *compression height*, that is the distance from the gudgeon pin axis to the piston crown, so that not only can the piston be made lighter but also the height of the cylinder block can be reduced. An improvement in both fuel economy and engine manufacturing costs is therefore made possible.

Other developments in connection with piston rings mainly concern their angular and axial dispositions on the piston and their surface treatment. In the early 1980s a trend began in the use of piston ring grooves that were machined with a very slight upward tilt from the piston axis, thereby creating a similar effect to an internally stepped (or bevelled) piston ring. The purpose of this feature is therefore to ensure that when expansion and bending are taken into account, the ring groove and ring are presented squarely to the cylinder wall and not deflected downwards.

Beginning in the late 1980s it was recognized that by moving the piston rings nearer to the piston crown in both petrol and diesel engines, consistent always with maintaining durability, because it increases mechanical and thermal loading of the top ring groove and ring lands, the efficiency of combustion could be improved for increased power and reduced emissions of hydrocarbons. The reason for this is that the ‘dead’ space above the top ring and between the piston land and cylinder wall is useless for combustion of the gases compressed therein. Thus, if this space is reduced with what is sometimes called a ‘high-ring’ piston (Figure 1.58), smaller amounts of unburnt gases are expelled from the exhaust. In a recent example the top ring land height has been optimized at 3.0 mm (0.12 in).

Carbonitriding may be used as a less expensive alternative to chromium plating for cast iron top compression rings. This is a surface hardening treatment that involves the simultaneous diffusion of carbon and nitrogen into the ring material. Top compression rings produced from high alloy steel have also been recently introduced, which can allow ring widths as low as 1 mm (0.04 in) to be achieved, as compared with the more usual 1.5–1.75 mm (0.06–0.07 in) found in modern practice. The advantages associated with this type of compression ring include the following:

1 Improved fatigue resistance by virtue of their higher tensile strength.
2 Less tendency towards flutter conferred by their reduced inertia.
3 Increased conformability to improve control of blow-by and oil consumption.
4 Readily incorporated in ‘high-ring’ pistons for better emission control.

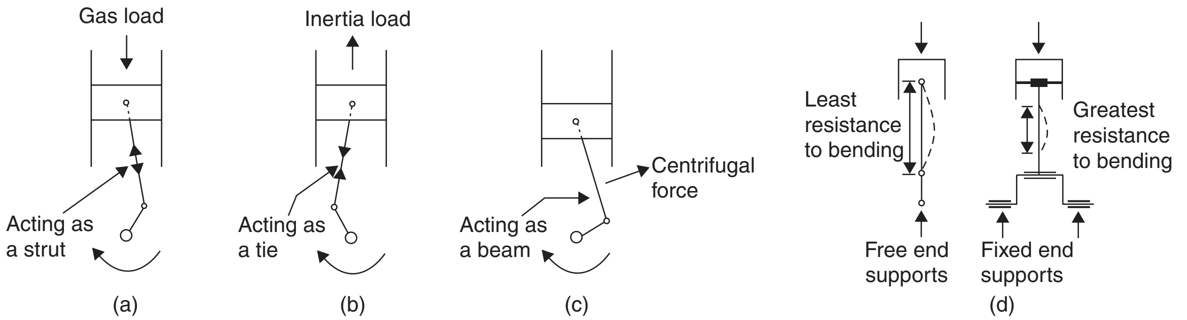


**Figure 1.59** Forces acting on the connecting rod: (a) compression (b) tension (c) bending (d) concentration of bending load

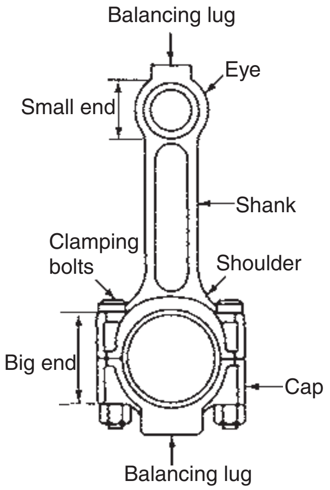


**Figure 1.60** Connecting rod assembly

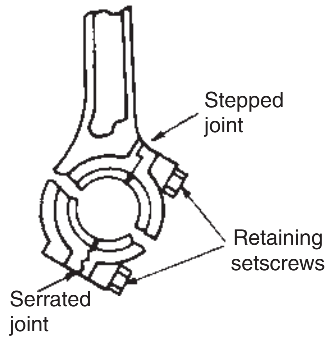


**Figure 1.61** Types of angled big end

The Nitrolite steel top compression rings produced by Hepworth & Grandage are hardened on all surfaces for reduced side face wear and have smooth blended edges, instead of the chamfers required on chromium-plated rings, to provide improved control of blow-by and oil consumption.

## Connecting rods

The nature of the loading on the connecting rod is such that it is subjected to a combination of axial and bending stresses (Figure 1.59), the former arising from reciprocating inertia forces and cylinder gas pressure, and the latter from centrifugal effects. The shank of the connecting rod is provided with an I cross-section for maximum rigidity with minimum weight (Figure 1.60). Since the end supports for the rod are free in the plane of rotation and fixed in the plane containing the crankpin and gudgeon pin axes, the largest dimension of the I-section is disposed in the plane of rotation to resist the greater bending effect therein. The depth of web in the I-section varies in accordance with any taper of the shank, but the flange width and web thickness remain constant over this length. To reduce stress concentrations the big-end arch and the small-end eye are merged very gradually into the shank portion of the connecting rod.

In some designs the big-end bearing parting line is arranged diagonally (Figure 1.61), because otherwise the width of the housing would be such that the connecting rod could not be passed through the cylinder for assembly purposes. To resist the greater tendency for the cap to be displaced sideways relative to the rod, either a serrated or a stepped joint is generally preferred for their mating faces. Hence, the securing setscrews in their clearance holes are relieved of all shear loads. Where the parting line between the rod and cap is arranged at right angles to the axis of the shank, the cap may be secured by either bolts and nuts, studs and nuts, or setscrews. They are produced from high-tensile alloy steel with special care being taken in their detail design to avoid stress-raising corners, which would lower their fatigue resistance. Their clamping load must always be such as to exceed the inertia forces acting on the rod. Reference has already been made to bihexagon or twelve-point headed screw fasteners in Section 1.6 and for the same reason they may be advantageously used to retain the big-end bearing cap (Figures 1.30 and 1.36).

A more recent development in connecting rod fabrication is for a forged sintered metal rod and cap to be produced in one piece and then forcibly separated. Basically, the sintering process involves highly compressing a metal powder, the composition of which can be tailor-made for the application, in a slim I-section mould. Next, the already formed connecting rod is heated to an elevated temperature in a sintering oven, which fuses together the metal particles to complete the process. The manufacturing advantages of what have become known as 'sinter-forged, fracture-split' connecting rods, are generally those of consistency in fabrication, minimal machining requirements and optimum material properties. In production a fully automatic process allows the cap portion, which is initially defined by a fine groove that corresponds to the parting line, to be split apart from the arch of the rod. This process results in the fractured halves of the rod and cap presenting an ideal mating surface for a perfect fit and improved accuracy of location, as compared to that possible with individually machined parts and, of course, it saves machining time. Although novel to the motor industry in the early 1990s, this type of 'fracture-split' connecting rod construction was originally developed in the late 1960s by the McCulloch Corporation in America, who had investigated various means of improving the structural integrity and fatigue resistance of connecting rods used in their very high speed chain-saw engines.

When a new engine is being developed it is now usual for a component supplier to design the piston and connecting rod as one interacting system, so that the lowest possible reciprocating weight can be attained.

### Connecting rod materials

A prime requirement for a connecting rod material is that it should possess a high strength-to-weight ratio. Connecting rods produced from aluminium alloy were used to a limited extent in earlier practice, the primary object being to reduce the loadings on the big-end and main bearings. Also they offered the possibility of being run directly on the crankpins without an intervening separate bearing material. However, their use declined mainly because of an unsatisfactory and unpredictable fatigue life, which made them liable to failure in service. Connecting rods are therefore either forged from a high-strength alloy steel or, as in later American practice, cast from a high-duty iron. With the latter method of manufacture a closer weight tolerance can be maintained than is possible with forgings. However, forged sintered metal connecting rods have more recently found favour for the reason mentioned earlier.

### Small-end bearing

Although the gudgeon pin and small-end bearing directly react against the combustion load, they are nevertheless made appreciably smaller in diameter than the crankpin and big-end bearing. This difference in size is not simply to accommodate the small end of the connecting rod within the piston, but can be justified by the small-end bearing benefiting from a much reduced share of the total reciprocating and rotating inertia forces created by the connecting rod and piston.

Where a fully floating type of gudgeon pin is used, a separate bearing bush is pressed into the eye of the connecting rod, its length being such as to limit the maximum bearing pressure to 62 MN/m$^2$ (9300 lbf/in$^2$). Since this bearing is difficult to lubricate effectively, because of its oscillating rather than rotating motion, it must possess a high degree of durability. It is now generally of composite construction and comprises a steel backing lined with a hard lead-bronze alloy. The assembly fit of the gudgeon pin in the small-end bush tends to be critical, since too much clearance can produce a small-end tapping noise. Too little clearance when cold can result in the piston being rocked by the angular motion of the connecting rod, thereby causing a temporary piston knocking noise.

Typical pin-to-bush clearances would be 0.005–0.010 mm (0.0002–0.0004 in) for a high-grade petrol engine and 0.012–0.030 mm (0.0005–0.0012 in) in the case of a heavy-duty diesel engine.

### Big-end bearing and nomenclature

Since the early 1930s it has become established practice for the big-end (and main) bearings to take the form of thin, flexible half-liners, their nomenclature being most conveniently presented by illustration (Figure 1.62). They are of composite construction and consist of a preformed thin steel backing or shell to which is bonded one or more very thin layers of relatively soft bearing material. Multi-layer bearings of this type therefore became known as thin-wall bearings, and were originally introduced by the Cleveland Graphite Bronze Company in America. Before the advent of thin-wall bearings, the housing was often lined direct with the bearing material, which then required boring and hand scraping to achieve a satisfactory fitting. Thin-wall bearings possess several important technical advantages, including a much improved fatigue resistance, a more compact installation and better suitability to mass production requirements. *Fatigue resistance* concerns the ability of the bearing to withstand fluctuating loads at fairly high temperatures, and is the single most important property required of a bearing. It is achieved in the thin-wall bearing because there is less deformation taking place in the much thinner layer of bearing material.

### Big-end bearing materials

A heavier-duty bearing material may be selected for the connecting rod big-end bearing than is used for the crankshaft main bearing, because it is subjected to centrifugal loading by the partly rotating motion of the lower part of the connecting rod. This effect is absent in the case of the main bearings supporting a counterbalanced crankshaft. The big-end (and main) bearings must not only possess adequate fatigue resistance, but also provide satisfactory wear qualities. These requirements tend to conflict in practice, since the relatively soft materials that have the best anti-wear properties are

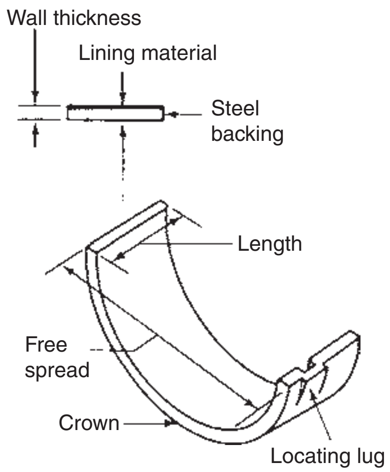


**Figure 1.62** Big-end half-liner

usually the least resistant to fatigue. The various bearing lining materials used can be listed as follows:

*White metal alloys* These are either tin based or lead based and are sometimes referred to as Babbitt metals, although strictly speaking this description should be confined to the tin-based alloy first developed in 1839 by Isaac Babbitt of Massachusetts. They contribute to a low rate of wear on the crankpins because of their good *embeddability*, which means they can readily bury any unfiltered wear particles entering the bearing oil film clearance and thus prevent scoring. The good *conformability* of these alloys also means they can tolerate any slight misalignment and deflections that may affect a bearing. Their load-carrying capacity is, however, limited by a modest fatigue resistance that decreases rapidly with increasing temperatures.

*Copper-lead mixtures* Harder materials of this type were originally developed to meet the combination of increased loads and higher operating temperatures. They possess a higher fatigue resistance than white metal alloys, but show inferior embeddability and conformability. These disadvantages were largely overcome by plating the lining material with a very thin overlay of a softer lead-based alloy, which also improves its corrosion resistance. It is usual for copper-lead bearings to require surface-hardened crankshaft journals.

*Aluminium-tin alloys* Bearing alloys containing 20 per cent tin and the balance aluminium, which compared favourably with copper-lead bearings in respect of fatigue resistance, were pioneered by the Glacier Metal Company in the 1950s. Their wear performance is usually such that a plated overlay is seldom required, since they are entirely compatible with both forged steel and the later nodular cast iron crankshafts. Furthermore, their corrosion resistance and thermal conductivity are both high for this type of bearing.

However, in more recent years not only has there been a trend towards reducing the overall length of engines, especially in relation to those transversely mounted, but also power outputs have increased as a result of pressure-charging, multiple valve cylinders and direct fuel injection, all of which have raised combustion pressures. The net effect of these changes has been a trend to shorter and more heavily loaded bearings. To meet this trend aluminium-tin-silicon bearing alloys have been developed to provide improved fatigue strength and antiseizure characteristics, whilst remaining highly resistant to corrosive attack by the acidic products present in used engine oil. In the case of the aluminium-tin-silicon bearings produced by the Glacier Metal Company, their patented alloy designated Glacier AS124 has a nominal composition of 12 per cent tin and 4 per cent silicon in a matrix of aluminium containing 2 per cent copper.

## Location of big-end bearings

Since these bearings are made detachable from the big end of the connecting rod, they must be located in both the rotational and axial senses, as follows:

1 To prevent rotational movement, the two half-liners are retained in their housing by an interference fit which, by virtue of maintaining an intimate metal-to-metal contact between liners and housing, also facilitates heat flow from the bearing. The interference fit is obtained by extending the half-liners a few hundredths of a millimetre beyond their true parting line, so that they are compressed into their housing when the bearing cap is tightened down. This difference in circumferential length between the pair of abutting liners and the closed bore of the big-end housing is known as the bearing *nip* or *crush* (Figure 1.63).
2 To prevent axial movement the two half-liners incorporate lugs that register with offset notches in both the connecting rod and its cap (Figure 1.62). The latter is fitted so that its notch faces the same side as the notch in the rod.

*Checking bearing liner nip*

The following precautions should generally be observed when carrying out this operation, although it is always advisable to consult the particular manufacturer's service instructions:

1 Ensure that the bearing seatings are absolutely clean.
2 Check that the correct replacement bearing half-liners are being fitted.
3 Note that the half-liners have a certain amount of free spread, so that they can be sprung and retained in position during assembly.
4 Position the half-liners so that their lugs register correctly with the locating notches in the rod and cap seatings.
5 Oil the bearing working surfaces and fit the cap the correct way round.
6 Check that the specified bearing nip or crush is present by first tightening the cap nuts or setscrews to the torque value specified by the manufacturer, then slackening one side to finger tightness and inserting a feeler gauge of appropriate thickness between the joint faces of the rod and cap. This procedure may be repeated on the other side. A bearing liner nip in the region of 0.08–0.10 mm (0.003–0.004 in) is fairly typical. Finally, of course, the cap is retightened.

Where a check on the actual bearing clearance is called for, this should usually be about 1/1000 of the crankpin diameter. An insufficient clearance space for the oil film could lead to an excessive rise in bearing operating temperature, which lowers the fatigue resistance of its material. The bearing clearance may conveniently be established by using the proprietary *Plastigage* method. This involves placing cross-wise in the bearing a suitable length of the thread-like plastics material, so that when the bearing cap is replaced and fully tightened down the material flattens out, because it is initially

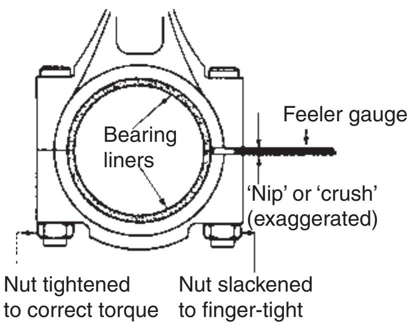


**Figure 1.63** Checking bearing liner nip or crush

larger in diameter than the clearance in the bearing. The bearing cap is then removed and the width of the flattened strip of Plastigage is matched against a graduated gauge supplied with the material, so that the corresponding clearance value for the bearing can be checked against that specified by the engine manufacturer. Since the Plastigage material is soluble in oil, care must be taken to wipe clean the bearing surfaces before checking their clearance and, of course, the crankshaft must not be turned during the checking procedure.

### Controlling axial movement of the connecting rod

It was once conventional practice to control the axial movement of the connecting rod from the crankpin, so that when the big end came into contact with either shoulder of the crankthrow there still remained a clearance between the small end and the adjacent gudgeon pin boss in the piston. In 1984 the Swedish Volvo company introduced a new method of controlling the axial movement of the connecting rod from the piston instead of the crankshaft, which has now been adopted by other manufacturers in the interests of reducing engine friction losses. Friction is reduced by virtue of the larger-diameter rotating location area being replaced by the smaller-diameter oscillating location area.

## 1.8 CRANKSHAFT ASSEMBLY AND MAIN BEARINGS

### Crankshaft construction and nomenclature

A one-piece, as opposed to a built-up, construction is most commonly used for motor vehicle crankshafts. It consists of a series of crankthrows connected together by the main bearing journals. Each crankthrow is formed by a pair of webs, these being united by the crankpins to which the big ends of the connecting rods are coupled (Figure 1.64). As mentioned in Section 1.5, the angular spacing of the crankthrows is related to engine balance and firing intervals.

The proportions of petrol engine crankshafts are usually such that the crankpin has a diameter of at least 0.60 of the cylinder bore dimension and a length of not less than 0.30 of the pin diameter. Web thickness of the crankthrow is generally in the region of 0.20 of the cylinder bore dimension. The main bearing journal is made larger than that of the crankpin, with a diameter of up to 0.75 of the cylinder bore dimension and a length of about 0.50 of the journal diameter.

Adequate crankshaft rigidity to resist both bending and twisting is a major requirement for smooth operation. With current short-stroke engines, the proportions of the crankshaft are generally such that in themselves they contribute to greater rigidity. This results from the combination of a smaller crankthrow radius and larger bearing diameters, which permit a beneficial overlap between the main journals and the crankpins (Figure 1.65).

Since the crankshaft is subjected both to bending and to torsional load reversals, it must also be designed to resist failure by fatigue. This condition may be initiated at any point where there is a concentration of stress or, in other words, a heavy loading confined to a very small area. In practice, it may occur at any abrupt change of cross-section, or from the sharp edge of an oil hole or a corner of a keyway. To avoid such stress raisers and therefore extend the fatigue life of the crankshaft, the areas in question are provided with carefully controlled small radii. For example, the corners of each main bearing journal and crankpin may be subject to what is termed ‘cold rolling’ to a specified fillet radius. This confers a beneficial compressive stress on the crankshaft material. The process of cold rolling basically involves rotating the crankshaft against small hardened steel rollers, which are forced against the corners of the crankshaft journals with a pressure sufficient to cause local plastic deformation and therefore compression of their surface layers. This widely used process actually dates back to 1938, when J.O. Almen of General Motors in America suggested its use to restore the durability of a Chevrolet truck crankshaft following an increase in engine piston stroke.

When the crankshaft is rotating, centrifugal force acting upon each crankthrow and the lower part of its associated connecting rod tends to deflect the crankshaft. Since this deflection is resisted by the main bearings, their loading is correspondingly increased. To reduce these loads, counterbalance weights are either formed integrally with, or separately attached to, the crankthrow webs (Figures 1.66a and b). The former arrangement is now most commonly used, the crankwebs being extended opposite to the crankpin and spread circumferentially.

### Crankshaft flywheel

A one-piece construction in cast iron is generally employed for flywheels used in conjunction with friction clutches. For

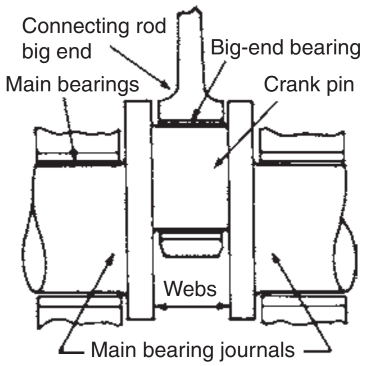


**Figure 1.64** Basic crankthrow arrangement

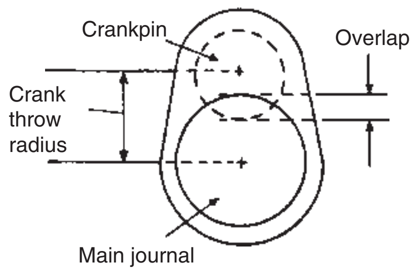


**Figure 1.65** Overlap between main journal and crankpin

motor car applications the flywheel is typically of the plain type (Figure 1.67a), although in a few designs a relatively thin disc or *flex-plate* is used to mount a separate cast iron rim (Figure 1.67b). The object of the latter construction is to minimize any disturbance of the flywheel that could result from bending vibration of the crankshaft (Section 1.9) and thus promote smoother running. A similar construction may also be adopted for use with a torque converter automatic transmission, where the flywheel effect derives from the impeller casing, which is mounted from a flex-plate bolted to the end of the crankshaft (Figures 1.67c and 17.6). In the case of some commercial vehicles a pot, instead of a plain, type of flywheel is required to accommodate a heavy-duty, twin-plate, friction clutch (Figures 1.67d and 13.30).

Radial location of the flywheel hub is afforded by a spigot on the rear end of the crankshaft. Owing to its appreciable inertia, the flywheel is located in the rotational sense by dowel pins and clamped firmly to the rear face of the crankshaft by a ring of bolts. The rim of the flywheel provides a mounting for the starter ring gear and may also bear timing marks for checking the valve and ignition settings, relative to prescribed positions of the crankshaft.

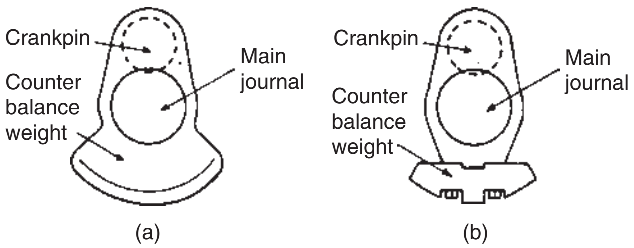


**Figure 1.66** Counterbalance weights: (a) integral (b) separately attached

### Flywheel with torsional vibration damper

Originally developed in the mid-1980s by Toyota for application to a motor car turbocharged diesel engine, the flywheel with torsional vibration damper or *dual-mass* flywheel as it is now often termed, has in more recent years become increasingly adopted for petrol engines where manufacturers seek additional refinement for the transmission system. The purpose of the dual-mass flywheel is to reduce the extent to which periodic fluctuations in engine torque are passed on to the transmission system, which otherwise create vibration, noise and can lead to wear of components. Typically noticeable with a dual-mass flywheel installation is therefore a reduction in transmission gear noise at low engine speeds. In this context there is a greater opportunity with a modern five-speed and reverse, all-synchromesh, gearbox for light load rattles to occur between the teeth of the more comprehensive train of constant-mesh gears (Section 18.3).

As its name suggests, a dual-mass flywheel basically comprises a two-piece flywheel with an *engine-side mass* and a *transmission-side mass* (Figure 1.68). The latter is supported from the former by an interposed ball-bearing race and its relative oscillatory movements are cushioned by a series of circumferentially spaced compression springs, which are retained in windows shared by the two masses. Frictional resistance to dampen the oscillatory movements between the two masses is supplied in a similar manner to that later

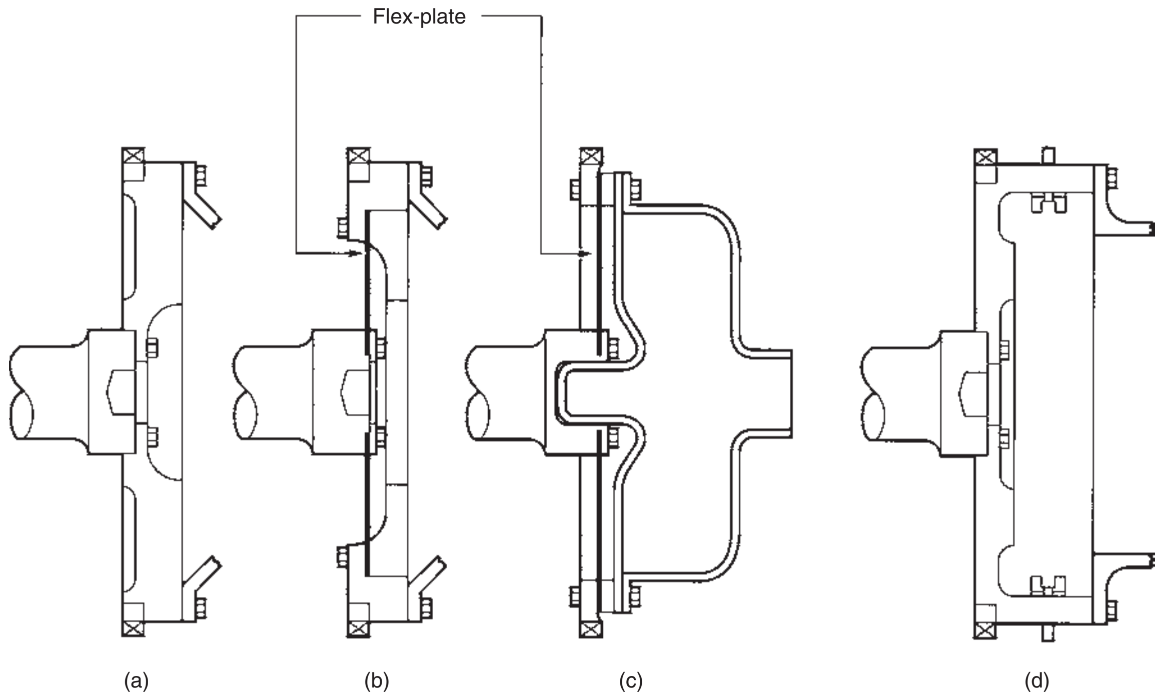


**Figure 1.67** Types of engine flywheel

(a) Plain type for mounting friction clutch
(b) Flex-plate type for mounting friction clutch
(c) Flex-plate type for mounting torque converter
(d) Pot type for mounting heavy-duty friction clutch

described for the centre-plate of a friction clutch (Section 13.3). In an alternative construction hydraulic damping is used between the two masses, fluid-filled chambers with restricted egress and ingress being arranged circumferentially. The changes in volume and restricted fluid movements within the chambers that accompany the relative oscillations between the two masses, therefore, confer the necessary damping properties.

## Crankshaft timing wheel and pulley

Ahead of the front main journal of the crankshaft, a cylindrical extension is machined to accept the driving wheel of the timing drive (the various arrangements of which are described at a later stage), an oil flinger (where fitted) and the driving pulley for the belt-driven engine auxiliaries.

These components are made close fitting on the shaft extension or nose and are prevented from turning relative thereto by either a single parallel-faced key, or a series of Woodruff keys, the latter being self-aligning by virtue of their semi-circular form (Figure 1.69).

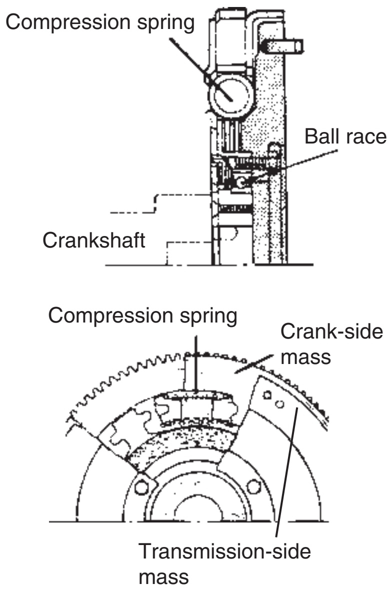


**Figure 1.68** Flywheel with torsional damper (*Toyota*)

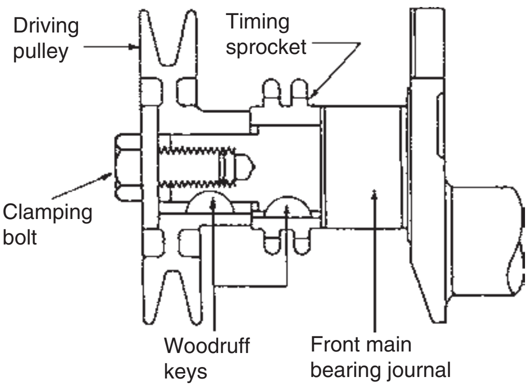


**Figure 1.69** Attachment of timing wheel and pulley to nose of crankshaft

The complete assembly of timing wheel, oil flinger and driving pulley are retained endwise on the shaft extension by means of a large setscrew which enters a hole tapped axially in the nose of the shaft and exerts its clamping load via a thick plain washer. In some engines, a driving gear may additionally be sandwiched either between an inner timing wheel and the pulley for a low-mounted oil pump, or between the front main bearing journal and the outer timing wheel for a concentric oil pump (Figure 1.30). In modern engine design a further requirement can be for a crankshaft mounted sprocket wheel, which provides a chain drive to harmonic balancer shafts housed below the crankshaft.

## Crankshaft materials

Until the early 1960s petrol engine crankshafts were traditionally forged from high-strength, low-alloy steels, and indeed these are still used for heavy-duty applications. Since then, however, the majority of car manufacturers have favoured the use of crankshafts produced from iron castings of the spheroidal graphite type, or SG iron as it is commonly termed (Figure 1.70).

High-strength cast irons of this type were first developed in the late 1940s, both in Britain and in America, and their distinguishing feature is that the graphite structure takes the form of spheroidal nodules. This feature confers higher strength, better ductility and greater toughness than the flake graphite structure of normal grey cast iron. It is obtained by injecting a trace of magnesium into the iron melt, which causes the graphite flake to gather into little balls or, more technically, spheroidal nodules, that greatly strengthen the grain structure of the material.

The material composition for a high-strength cast iron crankshaft would generally be as follows:

| | | | |
|---|---|---|---|
| Carbon | 3.50–4.20% | Sulphur | 0.03% max. |
| Manganese | 0.30–1.00% | Chromium | 0.20% max. |
| Silicon | 1.80–2.75% | Nickel | 1.00% max. |
| Phosphorus | 0.08% max. | Brinell hardness no. | 217–286 |

The Brinell hardness test, so named after J. A. Brinell who introduced it in 1900, measures the resistance to penetration of a material by a harder one in the form of a steel ball, and gives a useful indication of the tensile strength of the material being tested.

## Crankshaft manufacture

A forged crankshaft, like any other forged metal component, is manufactured by a process in which the metal in a more or less plastic, rather than molten, state is forced to flow into the

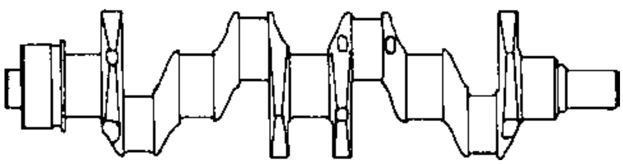

**Figure 1.70** General form of a high-strength cast iron crankshaft for a four-cylinder engine

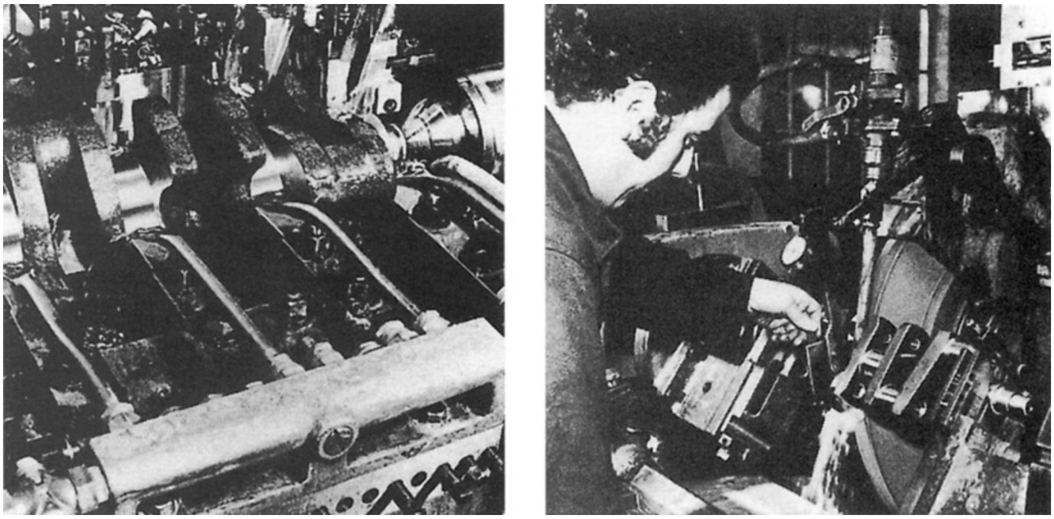

**Figure 1.71** Crankshaft turning and grinding (*Laystall*)

desired shape by means of hammering, squeezing and bending. In the case of motor vehicle crankshafts produced in large quantities, the actual shaping is performed by drop-hammer forging in closed dies. The latter are upper and lower blocks of metal in each of which an impression has been formed of the crankshaft.

A cast crankshaft is, in contrast, one that is manufactured by a process in which the metal in a molten state is poured into a mould and allowed to solidify. Motor vehicle crankshafts are not, however, produced in conventional sand moulds, but are cast vertically by the shell moulding process. In this technique, a thin shell-like mould of sand and synthetic resin is made by bringing these materials into contact with a heated metal pattern, the contours of which are exactly reproduced in the shell mould. Two such shells clamped together then form a complete mould.

Among the important advantages offered by the modern shell moulding process is that castings can be produced to much closer tolerances. This in turn reduces the amount of machining required afterwards, and in the case of the crankweb faces can eliminate it altogether.

After heat treatment to remove residual stresses and to give the specified tensile strength of material, usually about 63 kg/mm$^2$ (40 tons/in$^2$), the crankshaft must be machined to its final dimensions (Figure 1.71). This involves principally the rough turning, finish grinding and final lapping of the main journals and crankpins. With SG iron crankshafts it is good practice that the bearing journals and pins be final lapped with the crankshaft rotating in the same direction as it does in the engine.

### Crankshaft main bearings and nomenclature

In modern high-compression-ratio engines, the number of main bearings employed to support the crankshaft has tended to increase. This is because the crankshaft is subjected to greater bending loads, resulting from the higher peak gas pressures acting upon the pistons. Hence, the crankthrows must receive adequate support from adjacent bearings to

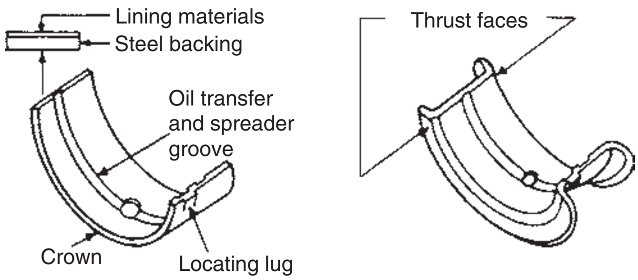


**Figure 1.72** Main bearing half-liners

minimize shaft deflections. It is therefore now customary for a main bearing journal to be used at each end of the crankshaft and between each cylinder of in-line engines, and each pair of cylinders in the more compact horizontally opposed and V engines.

The thin-wall main bearings are of similar form to the big-end bearings and must be rigidly supported in the crankcase so as to preserve not only the geometrical truth of their working surfaces but also their correct relative alignment with one another. These requirements must be met to avoid high localized pressures, which may result in destruction of a bearing through breakdown of its oil film with consequent overheating. Thicker than normal main bearing half-liners may be specified for V cylinder engines with aluminium alloy crankcases, so that optimum running clearances are better maintained even though the crankcase and crankshaft differ in their thermal expansion characteristics.

Similarly, the nomenclature of thin-wall main bearings follows that of the big-end bearings with the addition, of course, of an oil groove (Figure 1.72), the purpose of which is explained in Section 4.4. As an alternative to separate crankshaft thrust washers, one pair of the set of main bearing half-liners may be provided with integral flanged ends to serve as thrust faces. Such an arrangement simplifies the production build of an engine but does, however, lack the

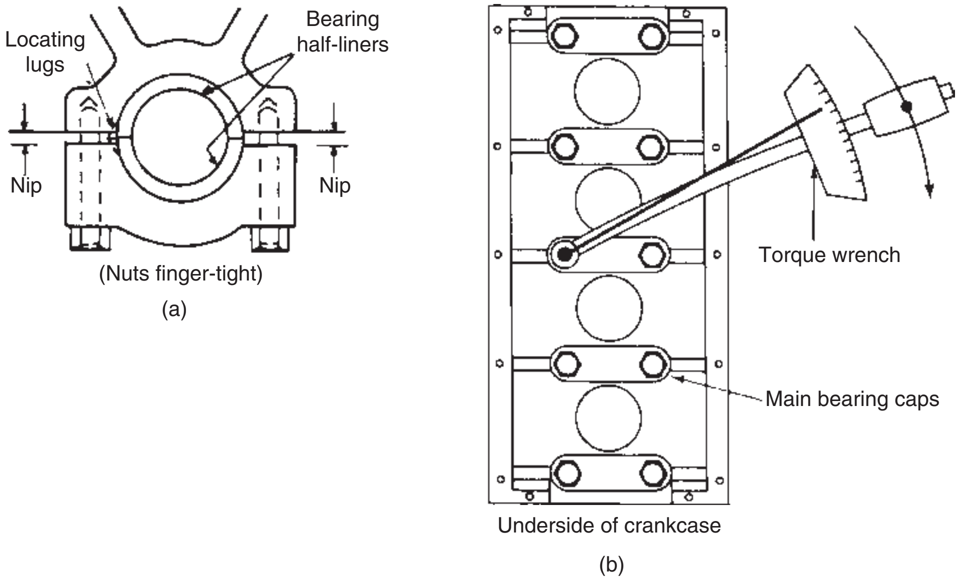


**Figure 1.73** Installation of crankshaft: (a) main bearing nip (b) torque tightening the main bearing caps

inherent self-aligning and good heat-conducting properties of separately mounted thrust washers.

### Main bearing materials and location

Again, the choice of materials and the methods of location for the main bearing half-liners correspond to those of big-end practice. However, another method of providing lateral location that is sometimes used for heavy-duty main bearings is the dowel-and-hole arrangement. In this method the dowels are fixed securely in the crankcase bearing saddle and cap, and register with circumferentially elongated holes in each half-liner. A disadvantage of this particular method of location is that the presence of the dowel hole disrupts the oil film in the bearing.

### Main bearings installation

The fitting precautions to be observed are basically similar to those mentioned earlier for the big-end bearings (Figure 1.73a). It is, of course, necessary to install the crankshaft thrust washers prior to fitting the appropriate main bearing cap, taking care that the oil grooves face the thrust surfaces on the crankshaft. The bearing caps are then torque tightened in sequence to their specified value, starting from the centre and working alternately towards each end (Figure 1.73b). Any undue resistance to rotation of the crankshaft by hand should be checked after finally tightening each bearing cap. The actual main bearing clearances may be checked using the Plastigage method, as previously described for the big-end bearings.

### Crankshaft thrust bearings

Before this particular bearing application is discussed, an engineering distinction must be made between two types of bearing. A bearing that is intended to resist a load applied perpendicular to the axis of the shaft is termed a *radial* or *journal* bearing. In contrast, a bearing that is intended to resist a load applied along the axis of the shaft is termed an *axial* or *thrust* bearing.

The crankshaft is located axially in the crankcase by plain (as opposed to rolling) thrust bearings which restrain it against endwise movement from loading imposed mainly by the transmission system. This loading may be in a forwards direction during release of a friction clutch and in a rearwards direction when a fluid coupling is in operation (Sections 13 and 15).

Crankshaft thrust bearings are generally of the same composite construction as that employed for the half-liners of the big-end and main bearings, as described earlier, although bronze bearings may also be used. They often take the form of separate semicircular thrust washers, which are installed either in pairs or singly on each side of one of the main bearing housings. Where thrust washers of smaller size are used in pairs, each lower half is keyed to the bearing cap, thus preventing both lower and upper halves from rotating once the cap is fitted (Figure 1.74a). If lower-half thrust washers only are used, rotation is prevented by their upper ends abutting the joint faces of the crankcase bearing saddle (Figure 1.74b). Since the thrust surfaces of the bearings must be separated by an oil film, the washers are provided with grooves or pockets to distribute the oil reaching them from the main bearings they embrace.

To accommodate both the thickness of the oil film and the thermal expansion of the parts concerned, a small clearance is provided on assembly between the thrust bearing surfaces and the adjacent contact faces of the crankshaft. This clearance must, of course, always be less than that existing between the crankweb faces and the sides of all the other main bearings, in order to relieve them of any thrust loads. The required clearance typically falls within the range 0.05–0.15 mm (0.002–0.006 in) and is termed the *crankshaft end float*.

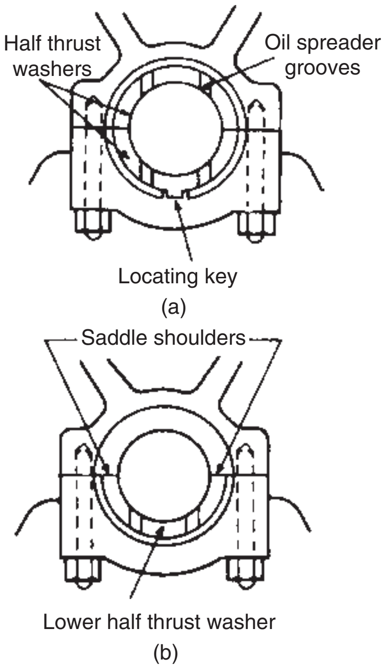


**Figure 1.74** Crankshaft thrust bearings: (a) paired (b) single

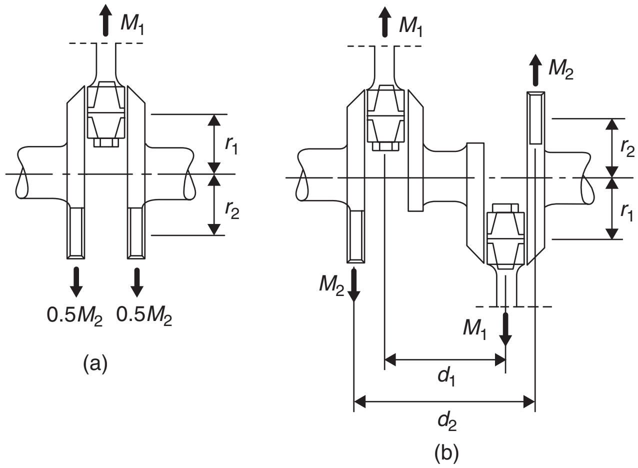


**Figure 1.75** Simple examples of force and couple balance in crankthrows:

(a) force balance in single crankthrow when $M_2 = M_1r_1/r_2$
(b) force balance in opposed crankthrows when $M_1r_1d_1 = M_2r_2d_2$
$M_1$ rotating mass at crankpin
$M_2$ rotating mass of counterbalance weight
$r_1$ radius at which $M_1$ acts
$r_2$ radius at which $M_2$ acts
$d_1$ distance between offset cylinders
$d_2$ distance between counterbalance weights

## Crankshaft balancing

In Section 1.5 our attention was directed in simple terms to those factors which make for acceptable overall balance of a reciprocating piston engine. Similarly, we have now to consider the basic principles involved in balancing the crankshaft itself, which compared with such items as the cooling fan and the flywheel is the most difficult part of an engine to correct for imbalance. At the heart of crankshaft balancing is the requirement for counterbalance weights. These are incorporated by the engine designer for one or more of the following purposes.

Static balancing of the rotating parts
Partial balancing of the reciprocating parts
Dynamic balancing of the rotating parts
Reducing bearing pressures.

*Static balancing of the rotating parts*
As with all rotating parts used in engineering, the unbalanced rotation of a crankshaft becomes more significant as speed increases. Indeed, the centrifugal forces arising from any such imbalance increase with the square of the engine speed – that is, doubling the speed makes the vibration effect four times worse. It is therefore perhaps stating the obvious that a crankshaft, like a road wheel, must at least be in a condition of *static balance*. This is achieved when the centre of gravity lies on the axis of rotation of the crankshaft, which implies that there is an equal distribution of mass about this axis. To verify this in practice requires the crankshaft to be supported with two of its main journals resting on knife edges; then whatever position the crankshaft is rolled to it should show no tendency to oscillate when coming to rest.

Of all the various arrangements of cylinders used in automotive practice only the single and in-line twin ones require the addition of counterbalance weights to correct for inherent lack of crankshaft static balance. This arises simply because the mass of their crankthrows is not distributed symmetrically about the axis of crankshaft rotation. It therefore becomes necessary to provide counterbalance weights on the opposite side of the axis of rotation to the unbalanced crankthrow(s). Since a single counterbalance weight cannot be placed in the same plane as that occupied by the crankpin and connecting rod, the crankwebs on each side are extended in the opposite direction to the crankpin to form a divided counterbalance weight (Figure 1.75a). In actual practice this weight is calculated so as to counterbalance not only the crankthrow but also the lower portion of the connecting rod, which may be considered as revolving with the crankpin.

For all other multicylinder arrangements, where the crankthrows are symmetrically spaced about the crankshaft axis, there is no inherent static imbalance and any correction needed is related to the machining tolerances on the crankshaft assembly. Counterbalance weights may still be required, however, but from other considerations of balance, as we shall presently find.

*Partial balancing of the reciprocating parts*
A force must be applied to accelerate a piston over the first half of its stroke, and similarly a force is developed by the piston as it decelerates over the second half of its stroke. As explained in Section 1.5, these reciprocating forces are termed primary inertia forces. Again, it is only with single- and in-line twin-cylinder engines that some attempt must be made to compensate for these forces by adding counterbalance weights to the crankshaft, since in all other multicylinder engines they

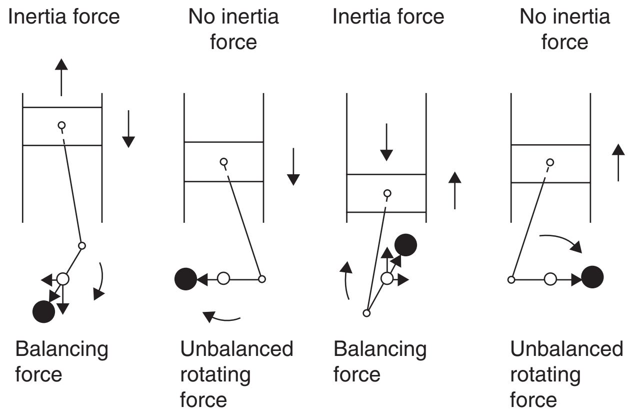


**Figure 1.76** Partial balance of reciprocating and rotating parts

are inherently balanced. This is because the arrangement and number of cylinders are so contrived that the primary forces generated in one cylinder are directly in opposition to, and therefore cancelled by, those in another cylinder.

With regard to the rogue single- and in-line twin-cylinder engines, their lack of primary balance can be partially alleviated by increasing the counterbalance weighting referred to in the previous section. The words 'partially alleviated' have been deliberately chosen, because a reciprocating mass can never be completely balanced by a rotating mass on the crankshaft owing to the resultant forces acting along different paths (Figure 1.76). Any attempt to counterbalance the reciprocating forces is always made at the expense of introducing an unbalanced rotating force, the centrifugal effect of which becomes the greatest nuisance at right angles to the cylinder axis. A compromise solution is therefore usually sought by increasing the mass of the divided counterbalance weights such that they produce an opposing force equal to one-half the maximum reciprocating force to be cancelled. In other words, the counterbalance weights added to the crankshafts of single- and in-line twin-cylinder engines generally over-correct for static balance and under correct for reciprocating balance.

*Dynamic balancing of the rotating parts*

Even though a multicylinder engine crankshaft might be in a state of 'perfect' static balance, it by no means follows that it is also in a similar state of *dynamic balance*. The reason for this is that in static balancing we are dealing with *forces*, whereas in dynamic balancing we are neutralizing *couples* – that is, opposing forces which are not acting in the same plane. An unbalanced couple can impose a rocking effect on the rotating crankshaft and therefore on the engine as a whole.

A ready impression of the manner in which an unbalanced couple acts may be gained by considering the crankshaft layout of a horizontally opposed twin-cylinder engine. Here the crankthrows lie on opposite sides of the crankshaft rotational axis, so that the symmetrical distribution of their masses allows static balance to be achieved without the addition of counterbalance weights. As a result, however, of each piston connecting to an individual crankthrow, the corresponding cylinders cannot be made truly coaxial. It is the presence of this offset distance between the nearly opposite cylinders which causes

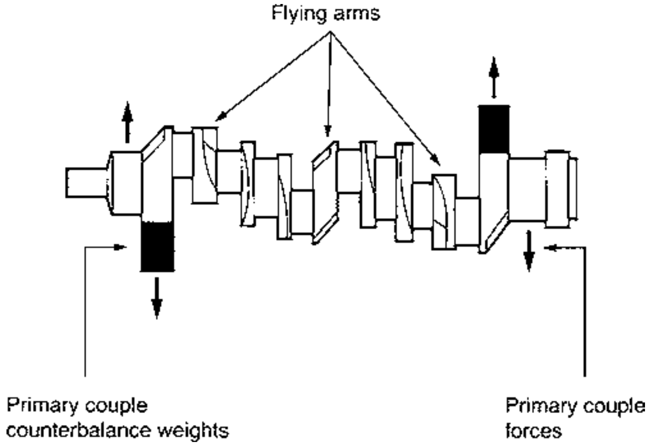


**Figure 1.77** Balancing arrangements for a 60° V6 engine crankshaft with four main bearings

the otherwise statically balanced crankthrows to introduce a rocking movement or unbalanced couple on the rotating crankshaft, arising from the centrifugal forces generated.

To attain dynamic balance it is therefore necessary to impose an equal but opposite couple lying in the same plane as the unbalanced one. In practice this can be arranged by extending the end crankwebs in opposite directions to their crankpins to form counterbalance weights (Figure 1.75b). Note that the addition of these counterbalance weights does not destroy the static balance of the crankshaft, so that whereas a statically balanced crankshaft is not necessarily in dynamic balance, a dynamically balanced crankshaft is always in static balance! The requirement for counterbalance weights to achieve dynamic balance of the crankshaft can also arise in other multicylinder engines, notably those with V cylinder arrangements. Taking for example the popular 60° V6 engine, there is an unbalanced primary couple that has to be neutralized by two widely separated and relatively large counterbalance weights, which act in opposition to each other (Figure 1.77). In actual practice a fairly thick counterbalance weight forms part of the foremost web of the crankshaft and a thinner one forms part of the rearmost web, the action of the latter being supplemented by an auxiliary counterbalance weight in the flywheel. A modest unbalanced secondary couple is also present, but this can be comfortably absorbed by the resilience of the engine mountings.

*Reducing bearing pressures*

This would not be the important balancing consideration that it is but for the fact that a crankshaft cannot be made perfectly rigid. So although a crankshaft might give every appearance of being in 'perfect' dynamic balance, the centrifugal forces developed at what may otherwise be inherently balanced crankthrows are still encouraging the shaft to bend as it rotates. In reality, of course, the crankshaft is being restrained from such bending deflection by the main bearings, but in performing this duty it will be clear that their loading and hence working pressures are being increased.

Another reason for adding counterbalance weights to a crankshaft is therefore to counteract bending deflection and thereby reduce bearing pressures. The crankshaft layout of

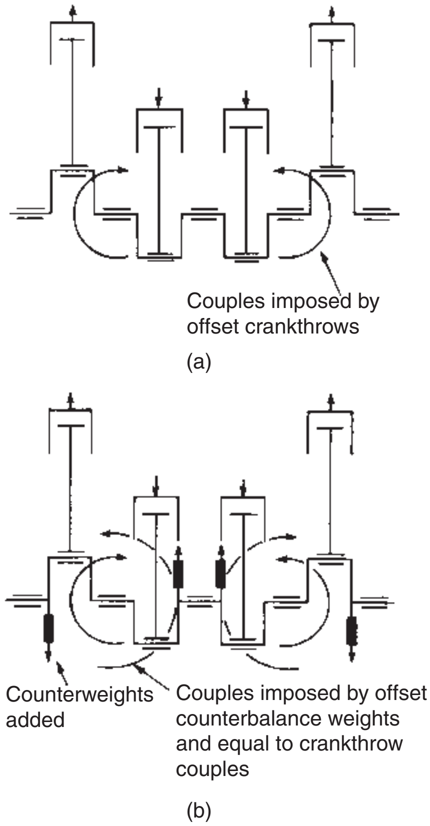


**Figure 1.78** Balancing conditions for an in-line four-cylinder engine: (a) dynamic balance (b) dynamic balance with reduced bearing pressures

modern in-line four-cylinder engines affords a good example of this principle being applied in practice. First, it is useful to consider the state of balance of such a crankshaft without the addition of counterbalance weights. Then we find that it possesses inherent static balance, because the two outer crankthrows lie on one side of the crankshaft rotational axis and the two inner ones on the other side. Also we find that although an unbalanced couple acts about the front half of the crankshaft in the same manner as that described earlier for the two crankthrows of a horizontally opposed twin-cylinder engine, this couple is counteracted by a similar one acting about the rear half of the crankshaft (Figure 1.78a). Therefore on the basis of providing a dynamically and by the same token statically balanced crankshaft, the need for counterbalance weight is non-existent; in earlier practice this was how the in-line four-cylinder crankshaft was designed.

With increasing engine speeds, however, the magnitude of the couples acting about each half of the crankshaft, even though self-cancelling, become such that they are trying very much harder to bend the crankshaft. Since this is being resisted by the main bearings, their working pressures are increased. For this reason it is now necessary to provide counterbalance weights for each half of the crankshaft and thereby cancel the individual couples so that bending deflection is minimized (Figure 1.78b).

A disadvantage of adding counterbalance weights to any crankshaft, especially if it is of the longer and more tortuously shaped six-cylinder kind, is that they lower the resistance to torsional vibration. This topic will be dealt with in Section 1.9.

During manufacture, final imbalance correction in a crankshaft is performed by drilling into the counterbalance weights.

*Harmonic balancers*

In Section 1.5 it was mentioned that there has been a revival of interest in the use of twin counterbalancing shafts, to cancel out the unbalanced secondary inertia forces of in-line four-cylinder engines. It was the highly esteemed automobile (and aeronautical) pioneer Dr F. W. Lanchester (1869–1946) who first applied this classic solution to the problem when in 1911 he patented what was originally known as an 'anti-vibrator', then later as a 'harmonic balancer' and in more recent years variously as 'counterbalancing shafts' and 'silent shafts'.

Technically speaking, though, we should perhaps prefer to use the description 'harmonic balancer'. This is because if we were to consult a treatise on engine balancing, it would become evident that the secondary inertia forces of an in-line four-cylinder engine result from the fact that the common centre of gravity of the four pistons, instead of remaining stationary as would be the case if the connecting rods were infinitely long, has a small harmonic motion of double frequency. It is this unwanted small harmonic motion that must be cancelled or balanced out, and explains the designation 'harmonic balancer'. The term harmonic may be thought particularly apt, because as in music when two notes sound right and pleasing together (Dr F. W. Lanchester was also an authority on music theory!), so we can contrive agreement between two moving forces.

However, it also seems to follow that a harmonic balancer would be unnecessary if only we could go out and buy ourselves a set of infinitely long connecting rods! Since the notion of infinitely long connecting rods may seem rather obscure to the motor vehicle service engineer, let us now try to understand its significance in simple terms. Referring to Figure 1.79a it will be seen that instead of using a conventional hinged connecting rod between the piston and crank, we have substituted a crank and slotted-bar or 'Scotch yoke' mechanism, which was sometimes employed in early steam and pumping engines. Comparing this arrangement with the conventional one shown in Figure 1.79b, it will be noticed that there is no inequality of piston travel towards the dead centres for corresponding angular movements of the crank. In fact by conferring a straight-line, instead of a swinging, motion on the connecting rod we have arrived at the practical equivalent of an infinitely long connecting rod. As a result the piston acquires a simple, as opposed to non-simple, harmonic motion. That is, it now oscillates about its equilibrium or mid-stroke position in such a way that its acceleration (positive or negative) towards this position is directly proportional to its displacement therefrom. So in the case of our in-line four-cylinder engine, the common centre of gravity of the four pistons would remain stationary and there would be no unbalanced secondary inertia forces.

Unfortunately a crank and slotted-bar mechanism would be quite impractical for a high-speed internal combustion engine, because of its unwieldy nature and the difficulty in arranging effective lubrication. We are therefore left with little choice but to use a hinged connecting rod, which imposes

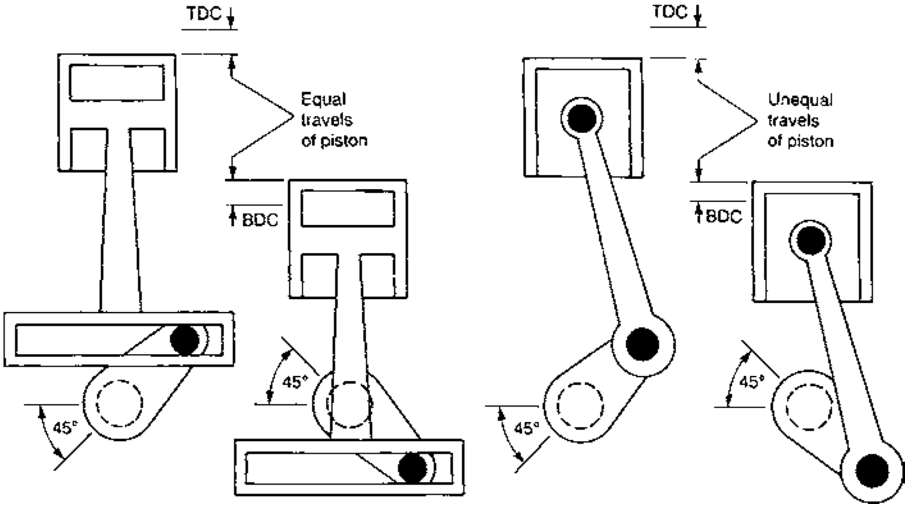


**Figure 1.79** Comparison between (a) slotted-bar and (b) hinged connecting-rod and crank mechanisms

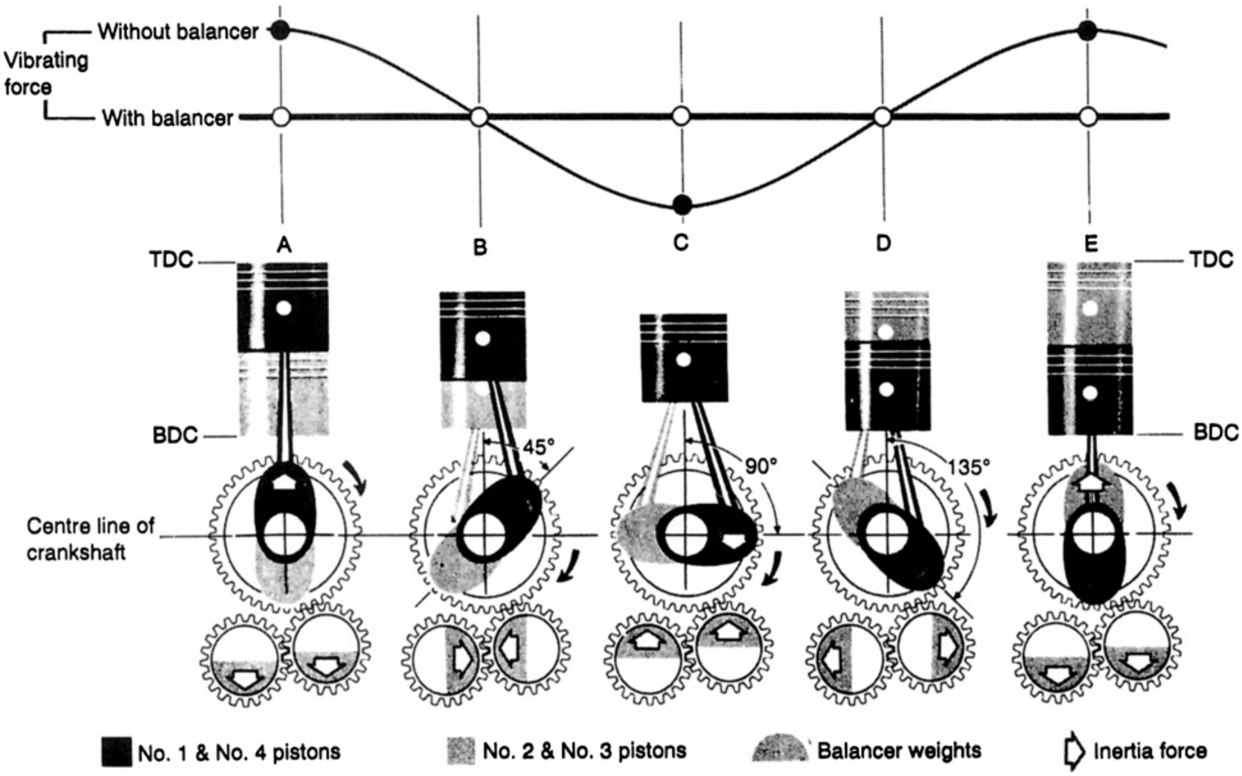


**Figure 1.80** Operation of four-cylinder engine harmonic balancer (*International Harvester*)

a non-simple harmonic motion on the piston and is responsible for creating the unbalanced secondary inertia forces with an in-line four-cylinder engine. The purpose of a harmonic balancer is therefore to cancel out these forces by introducing equal and opposite ones.

In Lanchester's original application of the harmonic balancer, two parallel shafts were fitted below and equidistant from the crankshaft axis. These shafts were geared together to run in opposite directions, with one of them being chain driven at twice the crankshaft speed, and they were provided with bob-weights in or about the plane of symmetry of the engine. The effects of these bob-weights were such that they neutralized each other in respect of their lateral motion, but combined in respect of their vertical motion to give a small harmonic one whose phase was contrary to that created by the error in motion of the pistons. Therefore when the pistons were at their top and bottom dead centres, the bob-weights were always at their lowest position. This sequence of events will be made clear in Figure 1.80, which applies to a heavy-duty diesel engine. In this installation the geared-together bob-weights are in turn driven from a gear shrunk on to the crankshaft and so positioned that they align with the exact centre of the engine. The balancer gears are matched and timed to each other by suitable markings and also to the crankshaft driving gear. Lubrication for the assembly is provided from the engine main oil gallery.

Despite its ingenuity, few designers of a past era in motoring thought it worthwhile to incorporate a harmonic balancer in their four-cylinder engines for passenger cars, their objections being the mainly practical ones of the additional faster working parts and weight and also the theoretical one of creating two otherwise useless lateral forces. It was not until the mid 1970s, when Mitsubishi Motors in Japan introduced their refined version of the harmonic balancer, that interest in its application was revived. Mitsubishi had found that by minimizing engine vibration in this manner, they were able to achieve a reduced level of booming in the body interior and also less intrusive noise at high engine speeds, together with improved durability for the engine auxiliaries and equipment.

The Mitsubishi installation of harmonic balancer, or 'silent shafts' as they call it, is of particular interest because of the way in which the balancer shafts are disposed at different levels on either side of the engine to perform an additional function. Namely, this arrangement provides a vertical separation about which the lateral forces generated by the bob-weight can be utilized to create a resisting couple, which assists in opposing sympathetic rocking vibrations of the engine on its mountings. Both balancer shafts are chain driven at twice engine speed from a sprocket on the nose of the crankshaft. The upper shaft is directly driven from its chain sprocket and therefore rotates in the same direction as the crankshaft, while the lower shaft is indirectly driven from its chain sprocket through a pair of gears, so that its direction of rotation can be opposite to that of the crankshaft (Figure 1.81). Each shaft with its bob-weights positioned in the centre plane of the engine is rigidly supported in front and rear bearings, the latter receiving lubrication under pressure from the former via holes through the centre of the shafts.

## 1.9 CRANKSHAFT TORSIONAL VIBRATION DAMPERS

### The causes of crankshaft vibration

When, say, a heavy and very rugged-looking engine crankshaft is being physically handled in the workshop, the impression given of rigidity is such that the possibility of it being set into a state of vibration might seem unlikely. However, the crankshaft is viewed rather differently by an engine designer, who recognizes that it does in fact behave as an elastic body. This is because in practice the crankshaft cannot be made perfectly rigid, and during engine operation it can be subject to three modes of vibration: torsional, axial and bending.

*Torsional vibration*

A ready impression of what is meant by torsional vibration may be gained by considering the antics of the simple torsional pendulum used in physics experiments. This piece of apparatus consists essentially of a metal disc suspended in the horizontal plane by a length of wire, the upper end of which is rigidly anchored to a fixed beam (Figure 1.82a). If a disturbing force momentarily turns the disc from its position of rest, then not only will the spring restoring force of the twisted wire return the disc to its original position, but also the inertia of the disc (or natural reluctance to change its state of rotary motion) will cause it to overshoot. This in turn sets up restoring forces in the again twisted wire that cause the process to be repeated in the opposite direction, and so on. The disc will continue to oscillate with diminishing amplitude or rotary movement either side of its original position of rest, until what we can now identify as torsional vibration of the pendulum dies away or decays (of which more will be said later).

If the disturbing force is applied repeatedly, so that the pendulum is excited into a continuous state of torsional vibration, then the amplitude of disc oscillation will depend upon the frequency of application of the disturbing force and the *natural frequency* of vibration of the pendulum. Here it

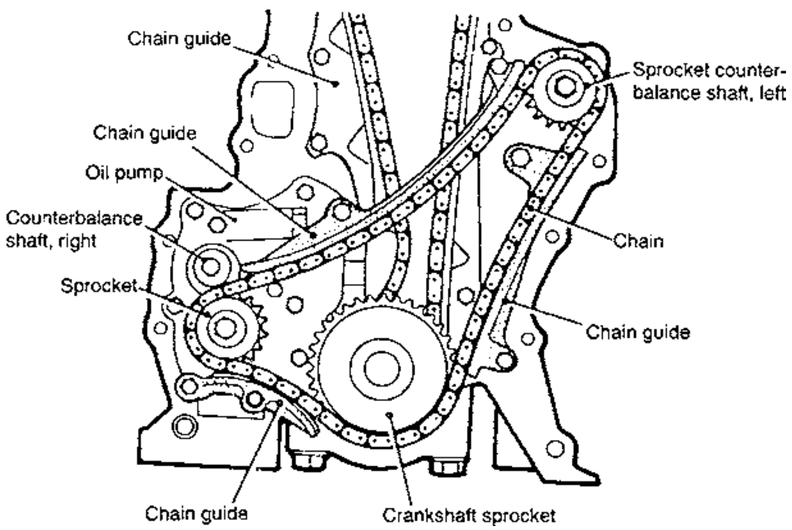


**Figure 1.81** Driving arrangements for 'silent shafts' harmonic balancer (*Colt-Mitsubishi*)

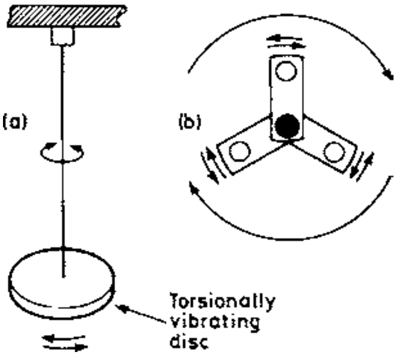


**Figure 1.82** Crankshaft torsional vibration: (a) simple torsional pendulum (b) torsional vibration superimposed upon crankshaft rotation

should be appreciated that every body possesses one or more natural frequencies of vibration (the term *frequency* means a certain number of vibrations performed per second) at which it will continue to vibrate when disturbed, until the energy it receives has been dissipated. A few simple experiments conducted with our torsional pendulum would also yield the further useful information that its natural frequency of vibration could be raised by any one or all of several methods. These are to make the wire shorter, to use a thicker wire, and to reduce the weight of the metal disc.

By applying our knowledge of the manner in which the simple torsional pendulum behaves, we are now better able to visualize how torsional vibrations can arise with the much more complex engine crankshaft and how they can be minimized. The disturbing forces applied to the crankshaft are, of course, derived from the pulsating gas and inertia torques acting via each piston, connecting rod and crankthrow combination, as explained in Section 1.4. If the crankshaft were perfectly rigid, then the only effect of these pulsating torques would be to cause some irregularity in its speed of rotation, which could be smoothed out by the action of the flywheel. However, as mentioned earlier, the crankshaft cannot in reality be made perfectly rigid, with the result that the torque pulses are capable of twisting it and therefore of exciting it into a state of torsional vibration. This vibration is superimposed upon the continuous rotation of the shaft (Figure 1.82b).

If the frequency of the disturbing vibrations should coincide with one of the natural frequencies of crankshaft vibration, then a condition known as *resonance* will occur. A danger of resonant vibration is that the energy of the disturbing vibrations may be greater than that lost by the twisting and untwisting of the crankshaft, so that the amplitude of torsional vibration builds up to such a degree that the crankshaft can be over-stressed and eventually suffer a fatigue fracture.

In practice, of course, the design of the crankshaft system is contrived such that its natural frequency of vibration is raised as high as possible. On a similar principle to our simple torsional pendulum, the natural frequency may be raised by making the shaft as short as possible, increasing its diameter and using a lighter flywheel. A resonant or *critical order* vibration would therefore only be expected to occur beyond the normal speed range of the engine, or in other words if the engine is over-revved. However, it may still be necessary in the interests of both engine smoothness and satisfactory operation of the timing drive to suppress the less critical orders of torsional vibration, which do occur within the normal speed range. For this purpose the crankshaft can be fitted with some form of torsional vibration damper.

*Axial and bending vibrations*

Axial vibration of a crankshaft is of lesser significance than torsional and bending vibration and is generally regarded as being a by-product of the former. That is, the twisting and untwisting of the crankshaft is accompanied by alternate decreases and restorations in length, which would of course apply to any other body subjected to the same treatment.

Bending vibration of the crankshaft when considered as a beam also results from the pulsating gas and inertia forces imposed upon it via the piston and connecting rod assemblies. As might be expected, this form of vibration is strongly reacted against at the main bearings. In high-compression-ratio engines, the maximum rate of pressure rise during combustion can be such as to cause undesirable vibration of the engine structure and mechanism, especially bending vibration of the crankshaft if it and its bearing supports are not rigid enough.

### Types of torsional vibration damper

When describing the vibration of a torsional pendulum, it was implied that in the absence of further disturbing forces the vibration eventually dies out or decays. It does so, of course, because of the natural damping effects of air friction on the surfaces of the oscillating parts and of internal friction in the spring material itself. The internal friction or *damping capacity* of a material is that property which relates to vibration energy being absorbed and subsequently dissipated in the form of heat. Apart from external air and internal material friction there is a further 'apparent' damping effect present in the case of crankshaft operation, this being the energy absorbed and again dissipated as heat by the shearing of the oil films that lubricate the crankshaft, connecting rod and piston bearing surfaces.

Even when the total effect of these natural and apparent damping forces is taken into account, it is by no means sufficient to suppress resonant vibrations of a crankshaft. For this reason is may be necessary to include in the crankshaft assembly a device known as a torsional vibration damper or simply crankshaft damper. The concept of the torsional vibration damper as a means of introducing additional friction to damper vibration of a crankshaft is quite old; C.H. Bradbury, a one-time leading diesel engine specialist, described it humorously as 'a corrective device for engines designed to run at one speed and sold to run at some other speed'.

Two types of torsional vibration damper may be encountered in petrol engine practice. These are generally classified as the slipper damper that was used in earlier engine designs, and the rubber damper that is in current use.

### Construction and operation of the slipper damper

This type of damper is basically a device for separating two masses of different inertia by frictional means, one being the crankshaft and the other usually a pair of small flywheels. The latter are not keyed to the crankshaft, but are simply spring loaded apart against friction surfaces on the damper hub, which is rigidly attached to the nose of the crankshaft (Figure 1.83a). So although the damper flywheels normally rotate in unison with the crankshaft, it is also possible for them to slip – hence the description slipper damper – relative to the crankshaft, once the friction torque that restrains their angular movement is overcome. Loosely fitting dowel pins are used to prevent relative angular movement between the flywheels themselves, whilst at the same time permitting them axial freedom under the influence of their spring loading.

The concept of the slipper torsional vibration damper is generally credited to Dr F. W. Lanchester, who developed such a device for the Daimler Company just before World War I, although Henry Royce arrived at very much the same solution to the problem also during this period. Early versions of what was once generally referred to as the Lanchester damper tended to be of elaborate construction, since they virtually amounted to a small multiplate clutch assembly with lubricated metal-to-metal rubbing surfaces. Later examples were of much simpler construction, as originally described and in the first instance pioneered by the American Chrysler Corporation in 1925. Non-metallic friction discs were incorporated in these later dampers, their rubbing surfaces being unlubricated.

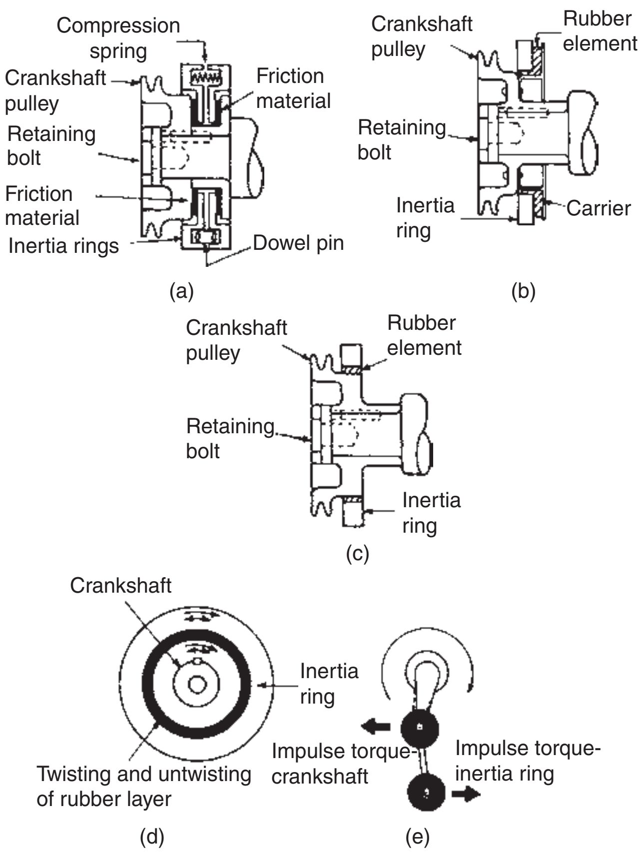


**Figure 1.83** Types and action of crankshaft torsional vibration damper: (a) slipper (b) bonded rubber (c) unbonded rubber (d) damping (e) detuning

At the onset of a critical vibration period, the inertia of the damper flywheels is such that they become increasingly reluctant to follow the torsional oscillations superimposed upon the rotating crankshaft, and ultimately slippage occurs between the faces of their webs and the friction discs. That is, the friction torque of the damper that normally restrains the flywheels from angular movement relative to the crankshaft is exceeded, so that the damper is in effect behaving like a slipping clutch. As a result the energy absorbed in overcoming friction at the rubbing surfaces and dissipated in the form of heat is abstracted from the energy stored in the vibrating crankshaft. A serious amplitude of torsional vibration is therefore never allowed to build up. It should be realized, however, that a slipper damper can only really be effective in dealing with the particular frequency of torsional vibration for which its spring loading and hence slipping torque has been predetermined.

### Construction and operation of the rubber damper

One of the shortcomings of the slipper damper was that its performance did not always remain consistent over an extended length of service, as a result of deterioration of its friction surfaces. The ensuing engine vibration periods could also be accompanied by noisy operation of the slipper damper, or what was sometimes referred to by service engineers as slipper roar. Put another way, a misbehaving slipper damper can have a worse effect on crankshaft torsional vibration than no damper at all. Recognition of these difficulties led to the development of a much simpler type of torsional vibration damper in which the two masses of different inertia, represented by the crankshaft and in this case a single small flywheel, were separated by both frictional and elastic means through the medium of rubber.

In long-established practice the rubber damper essentially comprises three concentric parts, these being a carrier cum hub assembly that is rigidly attached to the nose of the crankshaft, a ring-shaped flywheel or inertia ring that may be grooved to accept a V-belt, and a layer of rubber which is either bonded to each of these components (Figure 1.83b) or sandwiched between them under precompression, as in later designs (Figure 1.83c). It will be appreciated that there is no other connection between the carrier and the ring apart from that established by the intervening layer of rubber. The advantages of the later non-bonded version are that it is less costly to manufacture and allows a wider choice of rubber specification. Coincidentally, it was also Dr F. W. Lanchester who first patented a rubber type of crankshaft torsional vibration damper in 1928, and again the American Chrysler Corporation who, in the mid 1930s, developed the idea in the general form that we know it today.

Although the construction of the rubber damper is simpler than that of slipper and viscous dampers (Section 2.5), its operation is rather more complex because it acts as both a damper and a detuner. So far as the former duty is concerned, the onset of a critical vibration period causes the inertia ring to behave in a manner similar to the flywheels of a slipper damper, except that in this case the rubber layer is continually being twisted back and forth (Figure 1.83d). Since the rubber possesses a useful amount of internal friction, the energy it absorbs and rejects as heat is abstracted from the vibrating crankshaft.

So much for its friction damping function. The *detuning* function of a rubber damper is perhaps not so readily visualized until it is appreciated that, unlike the slipper and viscous dampers, its inertia ring forms part of a torsional spring drive system with its own natural frequency of vibration. It is, in fact, analogous to that of the torsional spring drive used in clutch centre plates described later, except that the wind-up action of the circumferential coil springs is replaced by the rubber layer being deflected in torsional shear. By careful tuning of the damper vibration characteristics, its inertia ring can be induced to move in anti-phase with the torsional oscillations of the crankshaft and thus oppose their build-up. As a further simplification, the action of the damper as a detuner may be compared to that of a double-jointed pendulum, where the application of a disturbing force to the upper half can cause both halves to swing in opposite directions; in other words, their masses will move in anti-phase (Figure 1.83e).

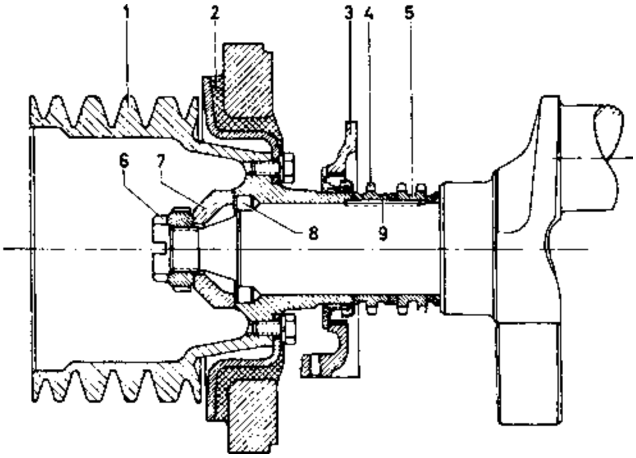


**Figure 1.84** Example of attention to detail in rubber type vibration damper installation (*Mercedes-Benz*)

1 V-pulley
2 vibration damper
3 cover with radial sealing ring
4 oil pump sprocket
5 camshaft sprocket and distributor idling gear
6 clamp nut
7 pressure flange
8 dowel pin
9 Woodruff key

### Methods of mounting torsional vibration dampers

For maximum effect a torsional vibration damper is always mounted on the front end of the crankshaft (Figure 1.84), since it is this end that suffers the greatest amplitude of vibration as a result of being furthest away from the *nodal point* (position of no vibration) of the crankshaft. The fact that the nodal point lies much closer to the rear end of the crankshaft can be explained by the greater inertia of the nearby flywheel and hence its reluctance to vibrate. If flywheels were mounted at both ends of the crankshaft then the nodal point would lie midway along the length of the shaft. It has sometimes been observed that it is one matter to design a damper of adequate capacity, but quite another to make its hub stay put on the nose of the crankshaft, because any joint transmitting a pulsating torque is inherently trying to fidget itself loose. A mechanically efficient joint is therefore essential for securing the damper and may be achieved by several methods, which most conveniently lend themselves to illustration (Figure 1.85). Split wedge rings and flange fittings are usually associated with heavy-duty diesel engine practice. An interference-fitting hub is sometimes favoured in American passenger car engine design.

## 1.10 VALVE TRAIN

### General background

Chiefly for reasons of accessibility, the inlet and exhaust valves in very early motor vehicles were arranged in two separate rows, one on either side of the cylinders and operated from beneath by similarly positioned camshafts. This long since obsolete arrangement of side valves provided what was known as a *T-head engine* (Figure 1.86a). Its combustion characteristics were later recognized as being poor, and the contrasting hot and cold (exhaust and inlet) sides of the engine could lead to cylinder distortion problems.

The T-head engine was then superseded by the *L-head engine* (Figure 1.86b), in which the inlet and exhaust valves were arranged in a single row on one side of the cylinders and again operated from beneath by a similarly positioned camshaft. Engines with this particular arrangement of side valves underwent considerable development and for many years provided a power unit which was generally cheap to produce. It became obsolete during the mid 1950s because its power output was somewhat limited by space restrictions on the usable size of inlet valves and by difficulties encountered in adequately cooling the exhaust valves.

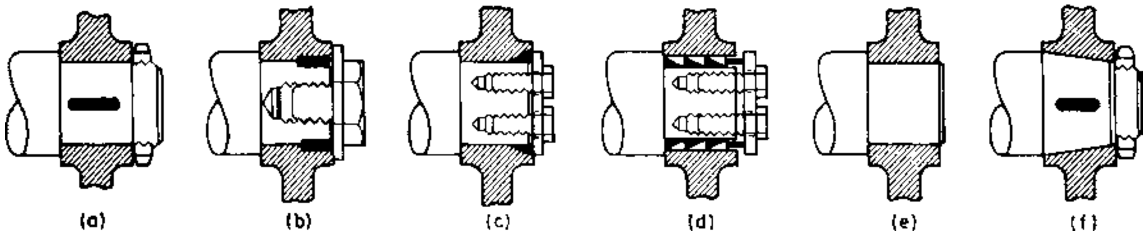


**Figure 1.85** Flangeless methods of hub mounting for crankshaft vibration dampers: (a) parallel with key (b) parallel with dowel pins (c) parallel with wedge ring (d) parallel with wedge rings (e) parallel interference (f) taper with key

An early compromise between positioning the valves alongside the cylinders and in the head above them was a combination of these two locations. This resulted in the occasional *F-head engine* being built with overhead inlet and side exhaust valves (Figure 1.86c), both sets of valves being operated from a single camshaft mounted in the crankcase. Each overhead inlet valve was operated by a push-rod-and-rocker system. The main advantage of this type of layout was that larger inlet valves could be used, but being heavier they also placed limitations on maximum allowable engine speed. F-head engines were relatively expensive to produce and have been obsolete since the early 1960s.

While these various developments of the side valve engine had been applied to many touring cars, the designers of racing and sports cars had fairly early recognized that better engine performance would be more readily obtained by placing both the inlet and the exhaust valves over the cylinders, albeit with a certain amount of mechanical complication and less quiet operation. In varying degrees such arrangements allowed for more efficient shapes of combustion chamber and for a less tortuous and therefore faster route to be taken by the ingoing mixture and the outgoing exhaust gases.

Furthermore, the inlet and exhaust valves could either be arranged in a single vertical or near-vertical row, or be separated into two rows and mounted at an included angle to each other. These two arrangements of in-line and inclined valves are thus said to provide *I-head* and *V-head engines*, respectively (Figures 1.86d and e). Their mode of operation can be either directly from a single or a pair of cylinder-head-mounted or *overhead camshafts*, or indirectly through a push-rod-and-rocker system acting upon one, and in some earlier cases two, crankcase-mounted or *side camshafts*.

The present-day requirement for high performance from a medium-power engine has resulted in generally higher maximum crankshaft speeds, typically in the region of 5500–6000 rev/min. This can be explained by recalling the factors governing the power output of an engine since, if neither piston displacement nor mean effective pressure can be further increased, then the only other way to raise maximum power is to permit higher crankshaft speeds. For this reason, it becomes increasingly more important to avoid erratic operation of the valves at high engine speeds. As a result, about 80 per cent of car manufacturers now offer models with the engine valves operated directly from an overhead camshaft, rather than by the less rigid push-rod-and-rocker system.

Commonly used abbreviations in connection with these valve layouts are as follows:

| | |
|---|---|
| SV | side valves |
| IOE | inlet-over-exhaust valves |
| OHV | overhead valves |
| SOHC | single overhead camshaft |
| DOHC | double overhead camshafts |

## Side camshaft, push-rods and rockers

*Camshaft*

This serves to open the engine valves positively and to control their closing against the return action of the valve springs. In motor vehicle practice, a one-piece construction is almost invariably used for the shaft and its cams (Figure 1.87). Camshafts are generally produced from hardenable cast iron, which has replaced the case-hardened forged steel material used formerly. The angular spacing of the integral cams is such as to impart the required motion, in correct sequence, to the inlet and exhaust valves in each cylinder. To preserve accuracy of valve motion, the camshaft must be rigid enough to resist deflection under the alternating torsional and bending loads imposed upon it by the valve operating mechanism.

The camshaft journals are supported radially in the crankcase by a series of plain bearings. Since the loading on the camshaft bearings is generally not heavy and the operating speed of the camshaft is only one-half that of the crankshaft, the bearings are sometimes machined direct in the engine structure, especially where this is of aluminium alloy. Otherwise, established practice is to use separate steel-backed bushes that are pressed into machined bores. These bushes are lined with either a white metal alloy or one of the other bearing materials referred to in connection with the big-end

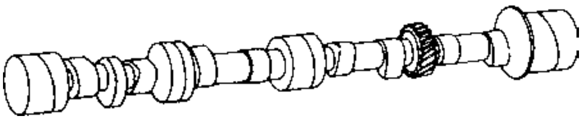

**Figure 1.87** Camshaft for a four-cylinder engine

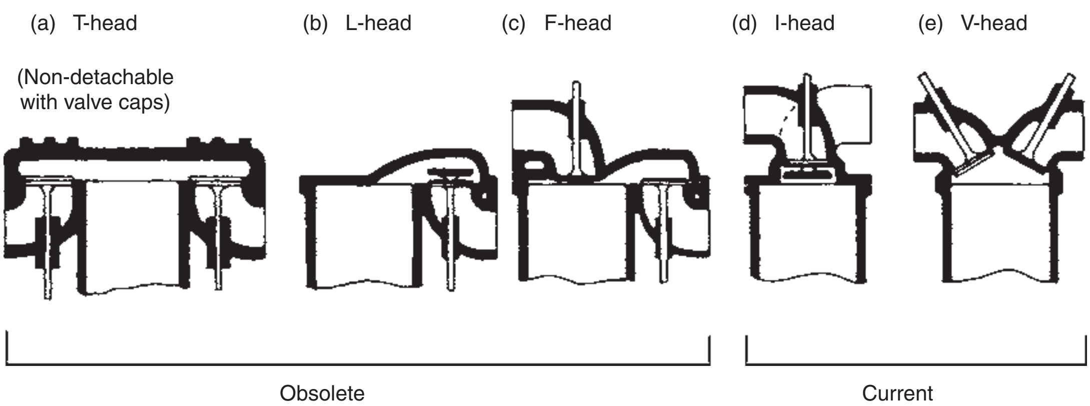


**Figure 1.86** Identification of basic valve arrangements

and main bearings (Section 1.7). To enable a camshaft to be inserted endwise through its bushes, the radius of its bearing journals is made slightly larger than the operating radius of its cams. There may also be a progressive, but albeit very small, decrease in radii for successive journals and bushes in the direction of camshaft insertion, so that it does not have to be threaded through close-fitting bearings of all the same size when being inserted (or withdrawn).

*Cam followers*

To convert the radial motion of the cams into the reciprocating motion necessary for opening and closing the valves, cam followers or tappets must be used (Figure 1.88a). This is because the force exerted by a cam acts perpendicular to its contact surface and therefore does not remain in the direction of follower travel. In other words, whatever mechanism bears directly on the cam is subject to a certain amount of side thrust. Sliding followers are almost invariably used in conjunction with crankcase-mounted camshafts. This type of follower is termed a tappet barrel (Figure 1.88a), and is typically produced from hardenable cast iron. However, since certain combinations of camshaft and tappet materials behave better than others, great care has to be exercised by the engine designer to ensure their compatibility and avoid scuffing. Various surface treatments are also applied to the cams and tappet barrels, in order to assist the running-in process of their highly stressed contacting surfaces.

*Push-rods*

These are required to transmit the reciprocating motion of the cam followers to the valve rockers (Figure 1.88a). Both ends of the push-rod form part of ball-and-socket joints which accommodate the angular movements of the push-rod arising from the straight-line motion of the tappet barrel on the one hand and the arcuate motion of the valve rocker on the other.

Since the push-rod is part of a valve train that constitutes a vibrating system, it must combine maximum rigidity with minimum weight. Push-rods are generally produced from steel and may be of either solid or tubular construction. In comparing solid and tubular push-rods of the same strength, the latter (Figure 1.88b) usually offer some reduction in reciprocating weight and can also serve as an oil conduit in the valve train lubrication system.

*Valve rockers*

The function of these is to cause both a reversal and a magnification of the motion imparted by the cam and follower to the valve. The valve rocker, or rocker arm, is a short rigid beam that oscillates about an offset pivot of either the journal bearing or the ball-and-socket type. An advantage of the latter type is that it makes the rocker inherently self-aligning, but it is necessary to introduce some means of restraining lateral movement. In cross-section the depth of the rocker arm greatly exceeds its width, since the bending loads imposed upon it are mainly within the plane of oscillation, with very little side loading.

Valve rockers may be of either solid or hollow construction (Figure 1.100a and b), and may be produced from steel forgings, iron castings, steel pressings, or in some later applications from aluminium alloy. Solid rockers oscillate about a stationary hollow shaft known as the rocker shaft. This is supported by a series of pedestals mounted on the top deck of the cylinder head, each pair of rockers generally being separated from the next pair by one of the pedestals (Figure 1.89). Sideways location of individual rockers is against adjacent rocker shaft pedestals, the rockers in each pair being held apart by means of a compression spring, spring clips or spacer tube. For some designs a separate bronze bushing is pressed into the rocker bearing bore, while in others the rocker is hardened all over with a plain bore bearing directly on the rocker shaft. A curved pad is machined on the end of the rocker where it contacts the valve stem tip, so as to allow the partly

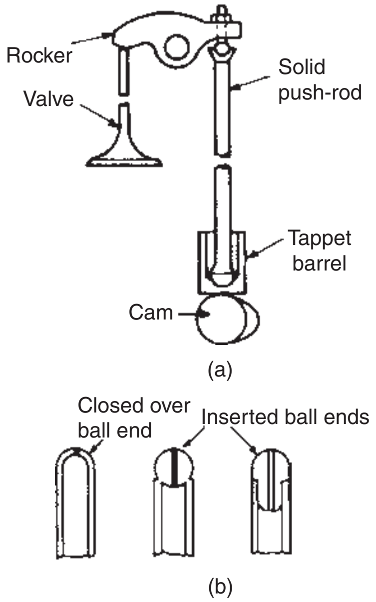


**Figure 1.88** (a) Push-rod valve operation (b) types of pushrod

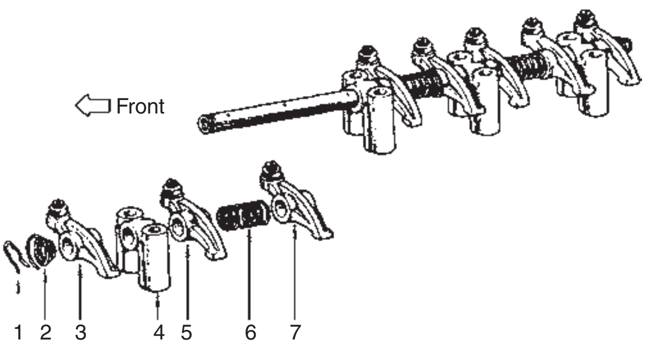


**Figure 1.89** Rocker shaft assembly for a four-cylinder engine (*Toyota*)

1 retainer clip
2 conical spring
3 valve rocker arm, no. 1 exhaust
4 rocker shaft pedestal
5 valve rocker arm, no. 1 inlet
6 compression spring
7 valve rocker arm, no. 2 inlet

rolling and partly sliding motion that occurs between them. Although they have been used with conventional rocker shafts, hollow rockers are now usually mounted on individual pivot posts with either hemispherical or part-cylindrical seatings (Figure 1.100b), the latter being currently favoured in American practice. The pivot posts may be pressed and in some cases screwed into bosses on the top deck of the cylinder head.

### Overhead camshaft and sliding followers

*Camshaft*
For a cylinder-head-mounted camshaft, modern practice tends to favour an increase in the number of intermediate bearings to one between each pair of cams. The bearing installations for overhead camshafts are currently similar to those of crankcase-mounted camshafts. An exception is where the bearings are supported in pedestals with detachable caps, which permit the camshaft to be inserted from above the engine. Also, this particular arrangement enables a reduction to be made in the diameter of the journal bearings, since the cams no longer have to be assembled through them (Figure 1.90).

*Cam followers*
The use of direct-acting sliding followers with a cylinder-head-mounted camshaft demands relatively large-diameter tappets (Figure 1.90). This increase in their base area is dictated by the absence of a multiplying leverage (otherwise provided by rocker arms) in the valve train, so that larger cams have to be used to give the desired amount of valve lift. Since an overhead camshaft is in even closer proximity to the valves than it was the obsolete side valve layout, the tappets must necessarily be both hollow to fit over the valve springs and short in length. For this reason they are aptly termed ‘inverted-bucket tappets’, although they may sometimes be referred to as Ballot-type tappets, having originally been introduced by the French Ballot company for its straight-eight engine in 1919.

In a recent Toyota development of the inverted bucket tappet, the tappet body is cold forged from aluminium alloy with a sprayed-on iron coating for its sliding surface. A tappet setting steel shim is recessed into its head portion and a stiffening steel disc is caulked beneath its head where contact is made with the valve stem tip (Figure 1.36). This construction promotes better fuel economy by virtue of generally lightening the valve train, because the valve spring load can be reduced so that less torque is required to drive the camshaft. Furthermore, the valve train is less prone to chatter, since the tappet is lighter in weight and not so hard as steel and is therefore quieter in operation.

### Overhead camshaft and pivoting followers

*Camshaft*
Similar remarks apply as in the previous system regarding the number of installation of the camshaft bearings.

*Cam followers*
These are known as the pivoting type because they swing between the camshaft and the valve tips and act as either a simple (straight) lever arm (Figure 1.100g and h) or ‘bell-crank’ (angled) lever (Figure 1.100i).

Modern applications of the former type generally feature a forged steel lever arm, or finger rocker as it is often called, which is self-aligning on a ball-and-socket pivot mounting (Figure 1.91), although journal bearing pivots have been used in the past. To restrain lateral movement of the ball-and-socket-mounted finger rocker, the valve stem tip may be recessed into the curved contact pad on the end of the arm (Figure 1.100g). To reduce valve train friction in some more recent applications, the sliding contact between the cam and finger rocker is replaced by a rolling one, the rocker being slotted in its central portion to embrace a roller mounted on needle bearings, the modern Ford V6 engine being an example of this practice.

The bell-crank type of pivoting rocker is supported on either plain journal or, more recently, needle roller bearings from a conventional rocker shaft. This is mounted above the camshaft, so that the downwards-extending lever arms of the bell-cranks bear on the cams and the outwards-extending arms contact the valve tips (Figure 1.100i)

### Comparison of different systems

*Side camshaft, push-rods and rockers*
The advantages of this method of valve operation are generally that the timing drive to the camshaft is uncomplicated and therefore less expensive to produce, and that tappet clearance

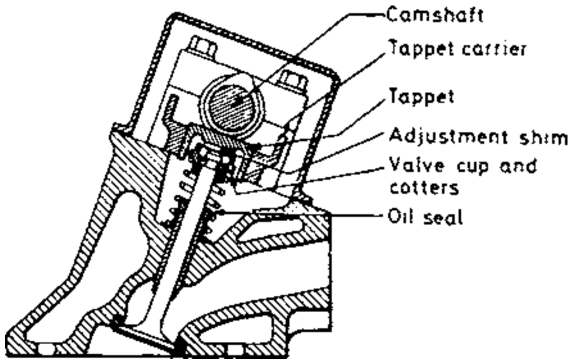


**Figure 1.90** Overhead camshaft and sliding followers

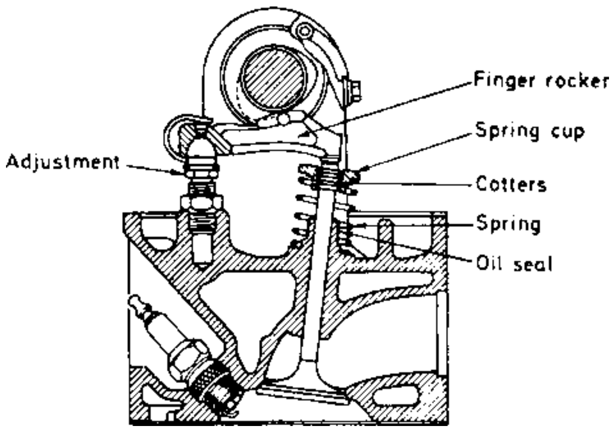


**Figure 1.91** Overhead camshaft and pivoting followers

adjustment can be conveniently performed. Disadvantages are concerned with its greater tendency towards vibration, which is due to flexing of the push-rods and rockers and creates false motions of the valves. There may also be the need for frequent tappet clearance adjustment because of the number of points subject to wear.

*Overhead camshaft and sliding followers*
For this method of valve operation the advantages are chiefly its lower inertia and greater rigidity in the interests of high-speed operation and a reduced requirement for tappet clearance adjustment. Its disadvantages include the need for a more elaborate and therefore expensive timing drive to the camshaft(s); the engine in general tends to be more difficult to service, although tappet clearance adjustment in particular need not be laborious in modern versions, and in any event should seldom be required; and the engine height is increased.

*Overhead camshaft and pivoting followers*
Here the advantages are that it can provide a further reduction in inertia over that of sliding followers, together with some magnification of the motion imparted by the cam to its valve. It is also simpler to incorporate a ready means of tappet clearance adjustment. Disadvantages are generally that finger rockers do not possess the same degree of rigidity as sliding followers; there must inevitably be some side thrust transmitted to the valves, and effective lubrication can be more difficult to attain for minimum wear between the cams and followers.

## Further developments in valve actuating systems

The advent of four-valve combustion chambers for passenger car engines, as discussed in Section 3.3, has necessarily led to some revision of the actuating mechanisms that have traditionally been used with two-valve installations. An interesting comparison between the various mechanisms that may be used, and which were investigated by the Toyota company during the development of their own four-valve engines, is shown in Figure 1.92. It will be noticed that in the second and third examples of SOHC layouts, a single cam and forked rocker is used to actuate one of the two pairs of valves. The final choice made by Toyota on the basis of valve train rigidity and cost effectiveness was the direct-acting second example of DOHC. This less complex system also minimized friction losses in the valve train, which contributed to improved fuel economy in the engine speed range most used during everyday driving. Furthermore, this particular system had already proven virtually maintenance-free in high-performance engines.

A new approach to camshaft manufacture has been adopted by Ford for their V six-cylinder engine, which was introduced in 1995 and features DOHC for each bank of cylinders. As a weight saving measure and also to reduce inertia loading on the timing drive, each camshaft is of tubular construction with pressed-on sintered metal cam lobes. In the continuing pursuit of engine refinement with multiple valve cylinders an interesting development by BMW has been to counterbalance the eccentricity of the paired cam lobes by offsetting, in the opposite sense, the shaft sections between adjacent cams.

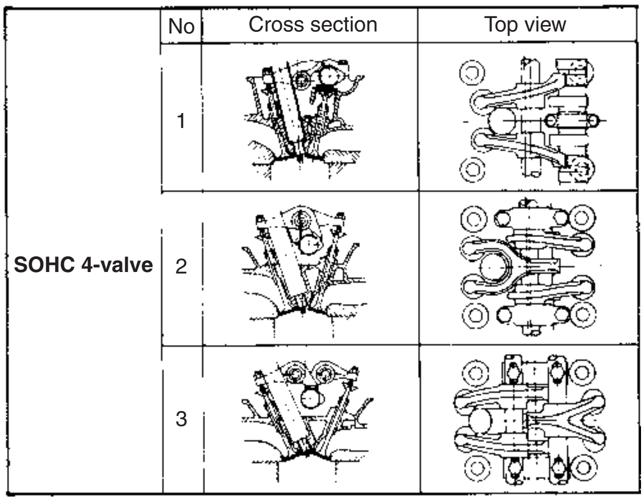


**Figure 1.92** Examples of actuating mechanisms for four-valve layouts (*Toyota*) HLA, hydraulic lash adjuster (hydraulic tappet)

## Identifying the parts of a cam profile

At this stage we must consider some of the whys and wherefores of what is probably the most versatile of all machine elements, namely the cam itself. According to their construction, cams used in engineering are usually classified into three types, known as disc, face and cylindrical cams. In the valve train of the automotive engine we are concerned only with the first-mentioned type. To convert rotary motion into a reciprocating movement, the basic parts of a simple disc cam profile comprise the base circle, the flanks and the nose (Figure 1.93). The last two parts constitute the cam lobe or lifting portion of the cam, since they determine the extent to which the follower can be lifted clear of the base circle. The period of valve opening, or valve event, is derived from the angular relationship between those points on the cam lobe where the opening flank begins and the closing flank ends. It therefore includes any interval during which the valve dwells in the fully open position, the governing factor here being the length of the nose arc. Conversely, that portion of cam rotation during which the base circle is operative with respect to the follower determines the valve closed period.

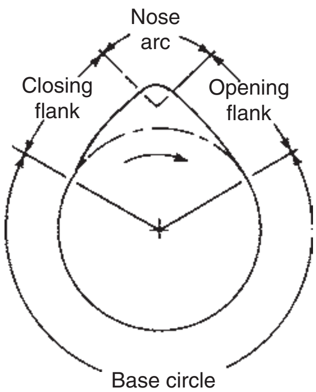


**Figure 1.93** Basic parts of a simple cam profile

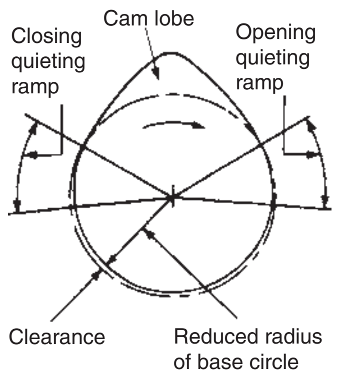


**Figure 1.94** Simple cam profile modified by addition of quieting ramps

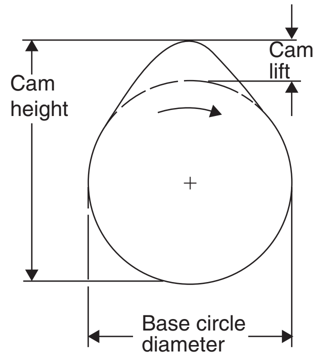


**Figure 1.95** Extent of cam lift

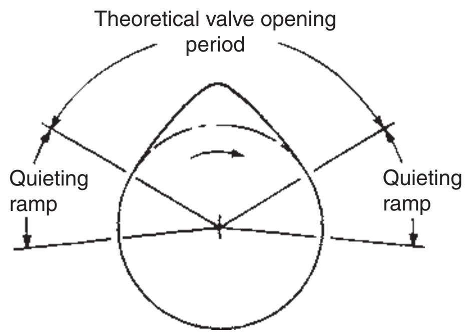


**Figure 1.96** Cam valve opening period

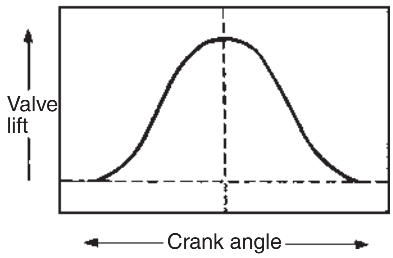


**Figure 1.97** Valve opening diagram

The cam profile must also take into account the effects of valve or tappet clearance, so as to minimize noise from this source. For this purpose the simple cam profile is modified to include quieting ramps between the base circle and the flanks (Figure 1.94). These quieting ramps take the form of inclines that occupy 15 to 30° of camshaft rotation with a rise equal to the tappet clearance. Geometrically they are blended into the cam profile by reducing the radius of the base circle to correspond with this clearance and then connecting the undercut base circle to the opening and closing points on the flanks by means of the ramps. Dynamically these ramps serve to reduce the velocity at which the valve initially leaves and finally returns to its seating. Impact loading is thus reduced throughout the valve train, which contributes both to quieter operation and to extended valve life. Ramp heights are reduced for engines that feature hydraulic tappets because these virtually maintain a zero clearance throughout the valve train as explained later.

## Valve lift and open period

From the description of the basic parts of a cam profile just given, it should be evident that a complete rotation of the cam momentarily lifts its follower by an amount equal to the overall cam height minus the base circle diameter dimension, the amount of lift being known as the cam lift (Figure 1.95). Here we must be careful to distinguish between cam lift and valve lift, since they are not necessarily one and the same thing. This is because the cam lift may be either less than, or equal to, the actual valve lift, depending upon whether or not a multiplying leverage is provided by valve rockers. The various arrangements of valve rockers were described earlier in this section, and it only remains to add that they usually confer a multiplying leverage in the region of 1.5:1. A further point is that valve lift will always be reduced in the presence of tappet clearance, so in the absence of valve rockers it would be more correct to the state that valve lift is nearly equal to cam lift.

The maximum lift of a valve generally approximates to one-quarter of its head diameter, since any greater opening than this gives diminishing returns in gas flow capacity. That is, the restriction to flow in the port and through the valve opening becomes equal when optimum lift occurs.

The period of valve opening or valve event has already been defined as the angular relationship between those points on the cam lobe where the opening flank begins and the closing flank ends (Figure 1.96). Whilst this is fine in theory, it is rather more difficult in actual practice to determine the precise points at which the valve begins to open and close. This is because the tappet clearance is usually taken up somewhere near the middle of the quieting ramp when follower lift for each degree of cam rotation is very small indeed. It is for this reason that when verifying the nominal valve timing of an engine, the checking tappet clearance is normally specified to be greater than the running clearance, so as to ensure that valve lift does not occur until the beginning of the cam opening flank when the initial movement of the valve will be more readily detected. Taking matters a step further, if the valve lift for every say 5° of crankshaft rotation is measured and plotted on a graph then we arrive at a *valve opening diagram* (Figure 1.97). The significance of this is that the larger the area under the curve, the more gas will flow through the valve during its open period. A valve opening diagram is therefore of immediate interest to an engine designer and also anyone involved in preparing or supertuning competition engines. It is of much less interest to an automotive service engineer, however, who is more likely to be concerned with measuring cam height during engine overhaul to check whether it lies within the wear limits laid down by the manufacturer.

## Types of cam profile

If each valve could be opened to its full lift instantaneously, dwell in the fully opened position for the desired period of opening and then be closed instantaneously, then clearly the area under the curve of its opening diagram would be at a maximum. Equally clearly, too, these ideal conditions of valve operation are unattainable in practice, since the infinite acceleration and deceleration of the valve train components would subject them to mechanically destructive forces. In the real-life automotive engine, it is therefore necessary to employ a cam profile that will open and close the valves in a more considerate manner. Although the theoretical considerations underlying the design of modern cam profiles are far removed from the realms of automotive service engineering, it may nevertheless be of interest to at least identify and briefly describe the various types of cam profile that have been used past and present, as follows.

*Concave cams*
A roller follower has to be used in conjunction with this early type of cam profile, where each concave flank is an arc of a circle that is tangent to both the base circle radius and the nose radius (Figure 1.98a). Although this type of profile lifts the follower with constant acceleration and deceleration, the transition from one to the other occurs very suddenly. Another disadvantage lies in the need to use a roller follower, which being heavier in construction creates larger inertia forces than a simple flat-faced one and thus makes the concave cam unsuitable for application to high-speed engines.

*Tangent cams*
Either a roller or a curve-faced follower is required for this again early type of cam profile, which is very simple in construction since each straight-line flank is a tangent joining the base circle radius with the nose radius (Figure 1.98b). It is sometimes described as a fast cam, because so long as the follower remains on the flank it continues to be accelerated. Then follows a very sudden transition from acceleration to deceleration, which is confined to that period of cam rotation when the follower is being lifted by the nose radius and is consequently very high in value. This in turn demands the use of a much increased valve spring force and generally makes the tangent cam again unsuitable for high-speed operation of an engine.

*Convex cams*
An essentially flat-faced follower is used in conjunction with this type of cam profile, in which each convex flank is an arc of a circle that is tangent to both the base circle radius and the nose radius (Figure 1.98c). A characteristic of this form of cam is that the follower is lifted for only a short period at high acceleration, this being followed by a much longer period of relatively leisurely deceleration, so that the controlling force exerted by the valve spring need not be too

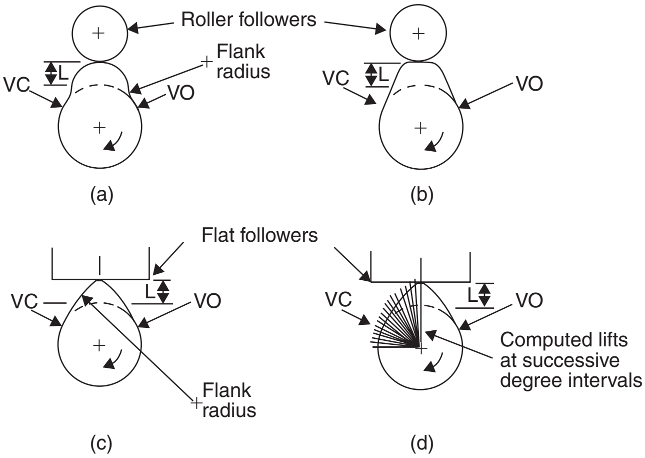


**Figure 1.98** Types of cam profile: (a) concave (b) tangent (c) convex (d) non-geometric convex. L lift; VO valve opening; VC valve closing

great. As with the previously described concave and tangent cams, the convex cam still suffers from the disadvantage that the transition from acceleration to deceleration occurs very suddenly. However, the three-arc convex cam was widely used prior to the advent of the modern convex cam of non-geometric type described next.

*Non-geometric convex cams*
Here again an essentially flat-faced follower is used with cam profiles of this type, where each convex flank is not simply an arc of a circle, but has been tailor-made from a series of computed lift values at successive degree intervals of cam rotation (Figure 1.98d). By this means it becomes possible to make less sudden the transition from acceleration to deceleration during lift of the follower and also, of course, likewise during its return. Furthermore, cam profiles of this type can be mathematically developed to compensate at certain critical operating speeds for the effects on valve lift caused by any lack of rigidity in the valve train components.

## Wear considerations related to cam form

The precise alignment required to avoid edge loading or digging in between nominally parallel cam and tappet contact faces is, in practice, very difficult to guarantee. Since edge loading could lead to excessive surface stresses, the contact faces of the tappet barrels may be made very slightly convex and are thus better able to accommodate any such discrepancies in alignment. This particular feature is usually combined with cams that are both tapered across their width and offset axially relative to the axes of the tappet barrels (Figure 1.99). The effect of this geometrical relationship between the cam and tappet contact faces is to induce rotation of the tappet barrel as it reciprocates, thus securing a more satisfactory distribution of wear over its working face. In modern OHC practice only the offsetting of the cams to induce rotation of inverted-bucket tappets is usually necessary, the cam and tappet contact surfaces being ground parallel.

In more recent American engine designs there has been a revival of interest in the roller type of sliding cam follower, the roller being used at the foot of a hydraulic tappet (this type of tappet is described later). The roller is provided with needle rolling bearings at its pivot and operates against a nodular iron camshaft with modern cam forms. Careful design is necessary to ensure that the roller is of optimum size to minimize contact stress. Also, the roller track may require a crowned profile, so as to compensate for any slight misalignment between the roller and its cam. Apart from conferring greater durability on the valve train, it is claimed by one manufacturer that this arrangement reduces friction by about 8 per cent and is therefore a contributory factor in improving the all-important fuel economy. A roller may similarly be applied to the pivoting rocker of an overhead camshaft installation, as described earlier.

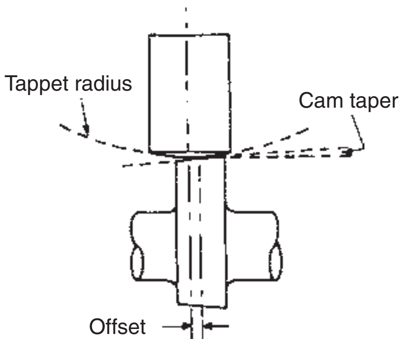


**Figure 1.99** Cam and tappet contact geometry

### Valve clearance: manual adjustments

The design of an engine must take into account the effects of thermal expansion and contraction of the valve train components, relative to that of the engine structure. Since these effects could result in the valves being held off their seats when they should be firmly closed, a small operating clearance has to be introduced into the reciprocating parts of the valve train. This is generally termed the valve or tappet clearance, and provision for either its manual or automatic adjustment is incorporated in all engines. The latter form of adjustment will be dealt with later.

For engines with push-rod overhead valves and solid rockers, valve clearance is usually adjusted at the rockers by means of a locking nut and hardened screw, the ball end of which registers in the push-rod socket (Figure 1.100a). The clearance may be checked by inserting a feeler gauge blade of appropriate thickness between the rocker pad and the valve stem tip. (A useful practical hint is not to slacken completely the locking nut while turning the adjusting screw.)

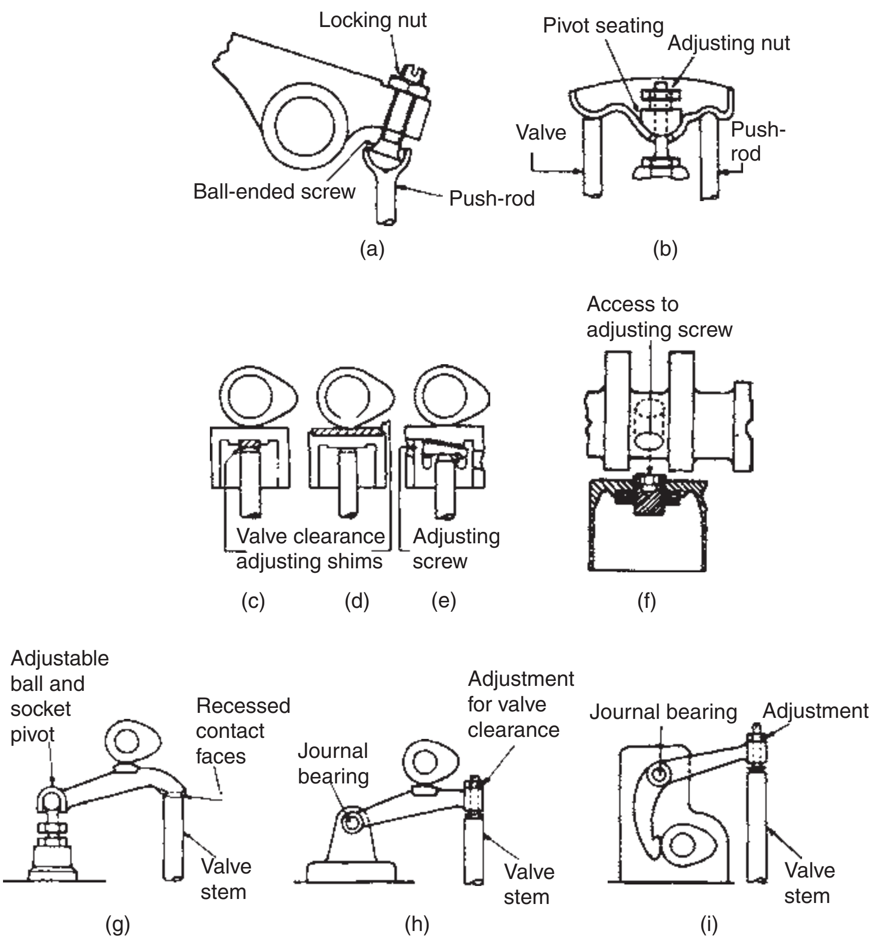


**Figure 1.100** Valve clearance adjustments

Where manual adjustment of the valve clearance is employed for push-rod overhead valves and hollow rockers, a screw-threaded portion at the upper end of the pivot post receives a self-locking nut, against the underside of which the pivot seating abuts (Figure 1.100b). Turning the self-locking nut thus allows the valve clearance to be set by virtue of either lowering or raising the rocker pivot. In the case of an overhead camshaft and inverted-bucket tappets, the adjustment can be a time-consuming operation in those installations where valve clearance is set by the selective assembly of graduated-thickness shims, which are inserted between the valve stem tip and the underside of the tappet head (Figure 1.100c). Other less inaccessible means of adjustment have, however, been introduced in recent years for inverted-bucket tappets.

In one now widely used system discs of appropriate thicknesses are located in recesses in the tappet heads and are directly contacted by the cams, so that if a valve is held depressed after being operated by its cam the disc may easily be changed (Figure 1.100d). Another system utilizes the wedging action of a taper-faced adjuster cum locking screw, which is located transversely in the tappet and forms an abutment for the valve tip (Figure 1.100e). Yet another and particularly ingenious system incorporates a self-locking screw in the head of the tappet. Access to this screw, which bears directly against the valve stem tip, is gained through a divided cam and cross-drilling in the camshaft (Figure 1.100f).

For manual adjustment of valve clearance with an overhead camshaft and self-aligning finger rocker, the ball pivot is usually screwed into a boss in the top deck of the cylinder head, so that raising or lowering the ball pin alters the valve clearance accordingly (Figure 1.100g). In the case of journal bearing pivots, a screw-type adjustment for valve clearance is generally provided at the valve tip end of the rocker (Figure 1.100h). The same applies to the bell-crank type of pivoting rocker (Figure 1.100i). An interesting point to note in connection with these various methods of valve clearance adjustment is that those where the screw adjuster remains stationary serve to reduce the inertia of the valve train.

*The effects of incorrectly setting tappet clearances*

Accurate setting of the valve or tappet clearances in accordance with the manufacturer's specification is of vital importance, otherwise at least the valve timing of the engine can be affected and at worst valve failure may result. In giving incorrect valve timing, should the clearance be set smaller than specified, the valve will tend to open earlier and close later than intended, and vice versa should the clearance be greater.

The effect of setting the tappet clearance much too close, perhaps in a misguided attempt to gain quieter running, can result in the valve not fully closing on to its seating. Apart from loss of cylinder pressure, this will allow the high-temperature combustion gases to blow past the valve face and, since no heat can escape to the valve seat, the valve head temperature becomes so high that rapid burning and destruction of the face occurs. This situation is usually aggravated by charring of the oil film on the valve stem, which causes the valve to stick in its guide. The effect of setting the tappet clearance much too wide is to produce excessively noisy operation and also wear caused by pounding of the valve against its seating. Furthermore, the amount by which the valve is lifted from its seating will be reduced.

**Hydraulic tappets**

Since their introduction to the American motor industry on the Pierce-Arrow car of 1932 (a make of car that we shall be meeting again later in connection with early power-assisted steering), self-adjusting hydraulically operated tappets are often employed in preference to the simple mechanical type. Although the latter can periodically be manually adjusted to compensate for changes due to wear, they are unable to accommodate the effects of thermal expansion. Therefore a running clearance must be established in the valve train, which not only creates a potential source of noise, but also can be subject to error in its setting. Hence, the reasons for using hydraulic tappets include quietness of valve train operation, constant valve timing and the elimination of valve clearance adjustments in service. These advantages are obtained because tappets of this type automatically maintain zero clearances throughout the valve train, under practically all operating conditions, because their action is such as to compensate for the differences in thermal expansion of the valve train components and the engine structure. They also simplify assembly of the engine and therefore reduce cost of production.

In construction, the hydraulic tappet is essentially a telescopic device that receives a pressurized supply of oil from the engine lubrication system. The engine valve opening load is imposed upon a spring-loaded inner plunger, which has either a ball or a disc type of non-return valve that normally acts to close off the foot of the plunger. An exceedingly fine working clearance is provided between the plunger and the tappet body in which it is free to slide.

Its operating cycle begins each time an engine valve closes, at which point the tappet dwells upon the base circle of the cam and the plunger return spring extends the tappet assembly to absorb any clearances that exist in the valve train. As any extension of the tappet will create a depression beneath the non-return valve of the plunger, the valve opens so that oil under pressure from the engine lubrication system is admitted to the tappet compression chamber (Figure 1.101a). This is formed between the closed end of the barrel and the underside of the plunger. As soon as the cam begins to lift the tappet, the increased pressure on the trapped oil maintains the non-return valve closed, so that the column of oil behaves like a rigid strut to transmit the opening load developed between the cam and the engine valve (Figure 1.101b). There is an intentional slight leakage of oil from the compression chamber, which takes place between the plunger and its operating bore in the tappet barrel. This controlled leakage is known as tappet *leak-down*, and its purpose is to ensure that the engine valve always returns fully to its seating, as once again the tappet returns to the base circle of the cam ready for the next operating cycle.

Although once generally applied to the valve trains of push-rod-operated overhead valve systems, hydraulic tappets are now increasingly used to similar advantage as self-adjusting pivots for the finger rockers in modern overhead camshaft installations. That is, the hydraulic tappet body remains stationary and its plunger provides a pivot mounting for the finger rocker. This arrangement has the advantage of minimizing the inertia of the valve train, since the adjustment mechanism does in effect remain stationary. However, there has been a recent tendency in some multiple valve DOHC engines to incorporate a hydraulic self-adjusting facility in direct-acting

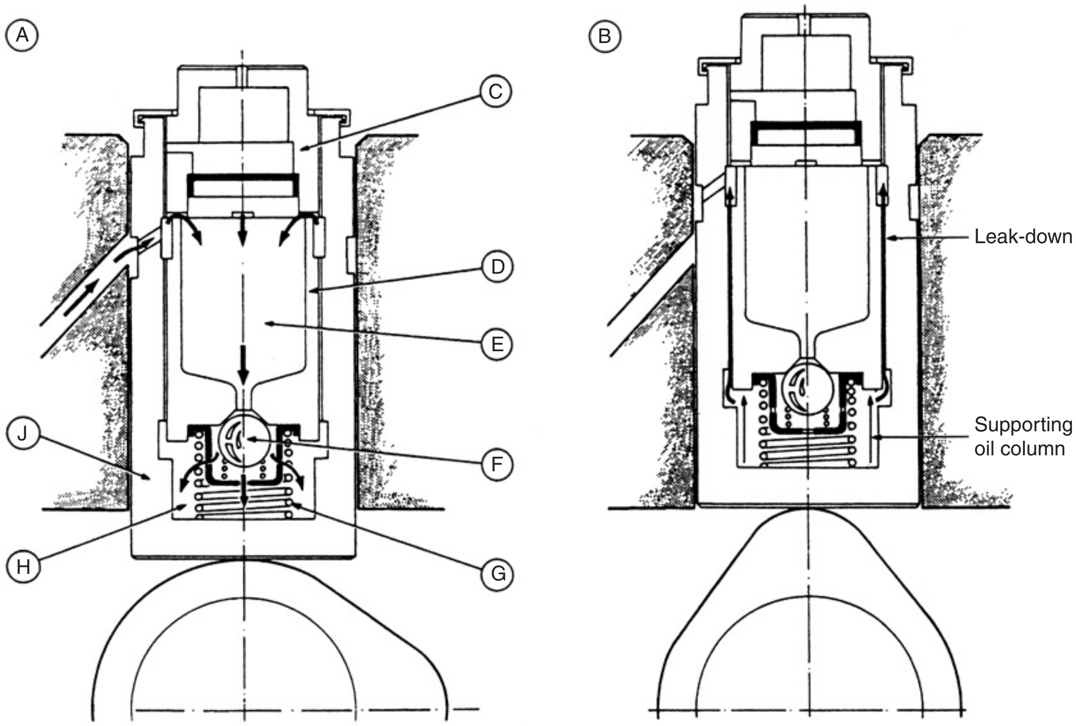


**Figure 1.101** Operating principle of hydraulic tappet: (a) tappet absorbing valve train clearances (b) tappet acting as rigid strut to open valve

A Valve closed
B Valve open
C Tappet plunger
D Tappet cylinder
E Feed chamber
F Non-return valve
G Tappet spring
H Pressure chamber
J Tappet body

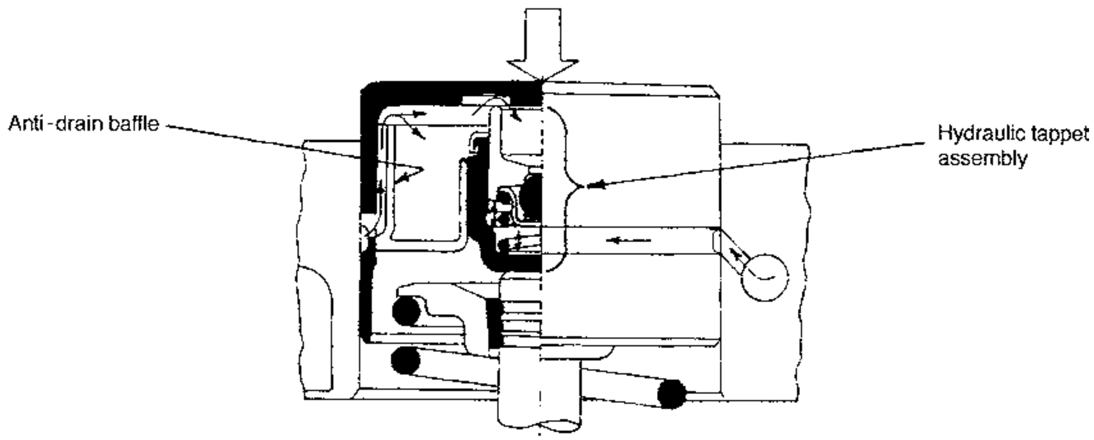


**Figure 1.102** Schematic arrangement of a hydraulic bucket tappet showing method of oil feed

sliding type followers, thereby producing a very compact 'hydraulic bucket tappet'. Its operating principle is identical to that already described for other hydraulic tappet applications, but an 'anti-drain' function may also be included. The relevance of the latter is that the nearer to the vertical that the tappet operates, the greater is the chance that air may be induced to enter its compression chamber when significant expansion of the tappet occurs. In other words, following the leak-down of a tappet under the spring load of an engine valve that remained partially open when the engine was switched off and then gradually closed, thereby leaving the tappet in a compressed state prior to the engine being started again. The purpose of an anti-drain baffle in the tappet (Figure 1.102) is therefore to ensure that, when the tappet expands on starting the engine, the level of oil retained in the tappet is high enough to exclude air from its compression chamber, which otherwise results in a noisy tappet until the air is eliminated. From similar considerations it must also be arranged that oil does not drain away from the hydraulic tappets supply gallery in the cylinder head when the engine is not running.