REEDS MARINE ENGINEERING AND TECHNOLOGY

MOTOR ENGINEERING KNOWLEDGE

FOR MARINE ENGINEERS

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REEDS MARINE ENGINEERING AND TECHNOLOGY

MOTOR ENGINEERING KNOWLEDGE FOR MARINE ENGINEERS

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PREFACE

The objective of this book is to prepare students for the Motor Engineering Knowledge part of the Certificates of Competency for marine engineering officers, issued by the Maritime Coastguard Agency (MCA) in the United Kingdom. The engineering certificates issued by the MCA also satisfy the International Maritime Organisation's (IMO) requirements for engineers which are detailed in chapter III of the Standards of Training, Certification and Watchkeeping for Seafarers (STCW). The latest edition of STCW includes the 2010 Manila amendments that are also included in this edition of Motor Engineering Knowledge as it is the most up-to-date information relating to the requirements of IMO's MARine POLution regulation MARPOL Annex VI which sets out the agenda for reducing the emissions from ship's engine exhaust gasses.

The text is intended to cover the ground work required for the examinations at the different levels of engineering officer of the watch, second engineering officer and chief engineering officer. The syllabus and engineering principles involved can be similar for both examinations but questions set for the chief engineering officer examination require a more detailed answer that those set at second engineering officer level. It is extremely important for the student preparing for the officer of the watch examination to concentrate on the safety procedures and practices of marine engineering. While it is not acceptable for the OOW to keep answering a question with 'I will ask the second or Chief', it should be remembered that responsibility does lie with the Chief and she/he is available to consult if all other options fail. The Chief on the other hand has no one to fall back upon although she/he can consult technical manuals.

The book can now also be considered as more than a specific examination guide and will be useful to superintendent engineers wishing to have a general guide to the latest trends from which they can seek more detail. Engineering knowledge is delivered via several different academic pathways from the Scottish Curriculum Authority's (SQA) Maritime Studies Qualification (MSQ) through Higher National Diploma's (HND) to Foundation Degrees and full Honours Degrees. The drawings are still intended to have direct relevance to the examination requirements but it is left to the student to practice his/her own versions.

The best method of study is to read carefully through each chapter, practicing the drawings, and when the principles have been mastered attempting the few examples knowledge as no model answer is available, nor indeed any one text book to cover all the possible questions. As a guide it is suggested that the student finds his/her information first and then attempts each question in the book in turn, basing their answer on either a good descriptive sketch and writing or a description covering about a page and a half of A4 paper. Try and complete this exercise in half an hour. I have found it particularly useful to use an artist's sketch pad, fill it with relevant drawings and practice them so that they can be reproduced as required in the examination.

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Paul A. Russell

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BASIC PRINCIPLES

Definitions and Formulae

Isothermal operation (PV = constant)

This is an ideal, reversible process carried out at constant temperature. It follows Boyle's law, requiring heat addition during expansion and heat extraction during compression. It is however impractical due to the requirement of very slow piston speeds.

Adiabatic operation (PV^{γ} = constant) (where γ = gamma = the adiabatic index Cp/Cv)

This is also an ideal and reversible process but with no heat addition or extraction and therefore the work done is equivalent to the change of internal energy. It is again impracticable due to the requirement of very high piston speeds.

Polytropic operation (PV^n = constant)

This is close to a practical process where the value of the index n usually lies between unity and gamma.

Volumetric efficiency

describe four-stroke engine and air compressor operation. Due to the restrictions of practical engine design typical values range between 86% and 92%.

Scavenge efficiency ('scavenging' is the term used to describe the air exchange process)

This is similar to volumetric efficiency but is used to describe two-stroke engines where some exhaust gas from the previous stroke may be included with the induction air at the start of compression. Both efficiency values are reduced by high revolutions (less time for the exchange process), and high ambient air temperature (less weight of incoming air). The introduction of exhaust gas recycling will change the values recorded on modern engines.

Mechanical efficiency

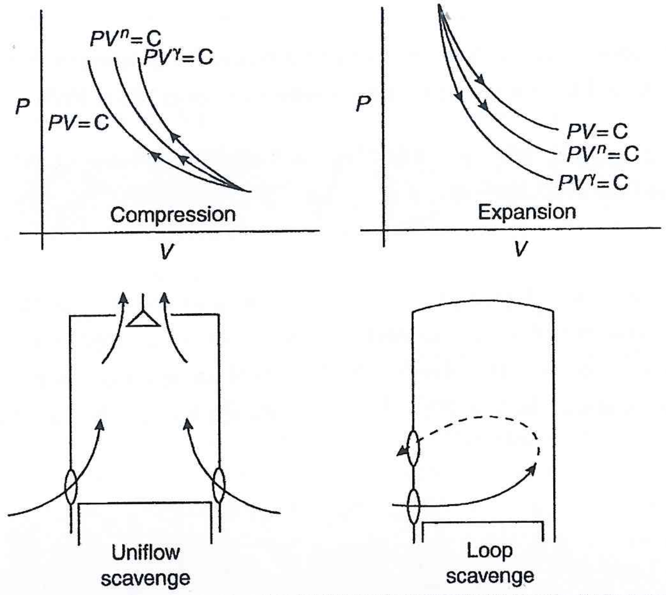
Mechanical efficiency is a measure of the mechanical perfection of an engine. It is numerically expressed as the ratio between the indicated power (power available from burning the fuel) and the brake power (power measured at the flywheel).

Uniflow scavenge

With uniflow scavenging the two-stroke engine is designed to have the exhaust at one end of the cylinder (top) and scavenge air entry at the other end of the cylinder (bottom) so that there is a clear flow traversing the full length of the cylinder (see figure 1.1) This design means that the scavenge air does not have to travel up the cylinder and down again, as with the other designs, to purge the exhaust gas from the previous cycle, hence the name UNI-flow. Due to the increased efficiency all modern engine designs are now based on this arrangement.

Loop scavenge and cross scavenge

This is the traditional two-stroke design where the exhaust gas exit and scavenge



▲ **Figure 1.1** *Compression, expansion*

sides of the cylinder with and without crossed flow loop (cross and transverse scavenge).

Brake thermal efficiency

This is the ratio between the energy developed at the flywheel, or the output shaft of the engine, and the energy supplied from burning the fuel. Traditionally this was measured by placing a 'load' or 'brake' on the output shaft, hence the term brake thermal efficiency.

Specific fuel oil consumption (SFOC)

SFOC is the fuel consumption per unit of energy at the cylinder or output shaft, kg/kWh (or kg/kWs), 0.38 kg/kWh would be normal for measurement at the shaft for kg/kWh. Therefore, a typical fuel consumption figure for a modern two-stroke diesel main engine would be quoted as being between 160 and 185 g/kWh.

Compression ratio (CR)

CR is a measurement of the ratio of the volume of air at the start of the compression stroke to the volume of air at the end of this stroke (measured between top dead centre (TDC) and bottom dead centre (BDC)). Usual value for a compression ignition (CI) oil engine is about 14:5 to 20:1, that is, clearance volume is 7.5% to 5% of stoke volume.

Fuel-air ratio

Depending upon the type and quality of fuel the amount of air required to give enough oxygen to completely burn all the fuel is about 14.5 kg for each kg of fuel. However, engines supply excess air to the combustion process and therefore the actual air supplied varies from about 29 to 44 kg/kg fuel. The percentage of excess air is about 150 (36.5 kg for each kg of fuel).

Performance curves for fuel consumption and efficiency

The initial design considerations for main engines powering merchant ships will be for optimised thermal efficiency (and minimum specific fuel consumption) to occur at the power conditions required to maintain the chosen service speed of the vessel. Marine practice is to quote the minimum specific fuel consumption at a given percentage of engine service load but maximum speeds are occasionally required when the specific fuel consumption will be much higher. Modern tonnage is often required to operate at speeds other than the design service speed. The practice of 'slow steaming' is now common and this means that the main engine will be required to operate at loads well below its service maximum continuous rating (SMCR). Engine manufacturers have responded and produced modern engines which have a much improved efficiency when running continuously at part load.

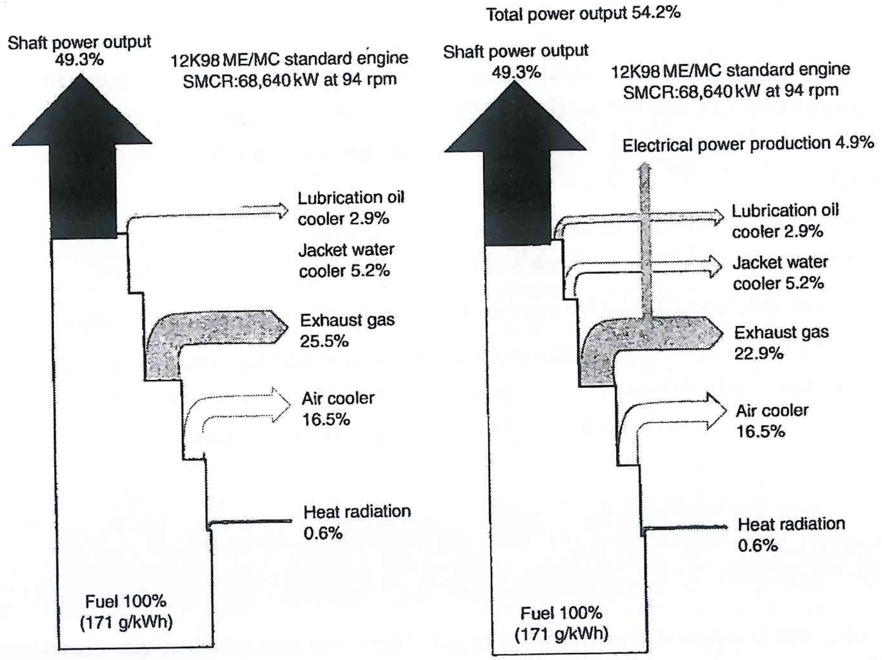
Manufacturers published performance curves (figure 1.2) that are useful in establishing principles, such as:

1. The fuel consumption (kg/s) increases steadily with load. However, the fuel consumption is not reduced by 50% if the load is reduced by 50% as certain essentials consume fuel at no load (e.g. heat for cooling water warming through, etc.).
2. Mechanical efficiency steadily increases with load as friction losses are almost constant and therefore become a smaller percentage of the total losses.
3. Thermal efficiency (brake for example) is designed to be at maximum at full load.
4. Specific fuel consumption is therefore a minimum at 100% power. Fuel consumption on a brake basis increases more rapidly than indicated specific fuel consumption as load decreases due to the friction losses being almost constant.

Heat balance

A simple heat balance is shown in figure 1.2. There are some factors not considered in drawing up this balance but as a first analysis this serves to give a useful indication of the heat distribution for the IC engine. The high thermal efficiency and low fuel consumption obtained by modern diesel engines is superior to any other form of engine in use at present.

1. The development of waste heat recovery systems gives the marine plant an efficiency gain as this is heat that would otherwise be lost to the environment.
2. The recent efficiency increases of exhaust gas-driven turbo-chargers not only contribute to high mechanical efficiency, by taking no mechanical power from the engine, but they also take a smaller percentage of the exhaust gas output to drive the charge air compressor. This means that more gas is left over to drive turbo generators, exhaust gas boilers and other waste heat recovery systems.
3. Cooling loss includes an element of heat energy due to generated friction.
4. Propellers do not usually have propulsive efficiencies exceeding 70% which reduces brake power according to the output power.
5. In the previous remarks no account has been taken of the increasing common

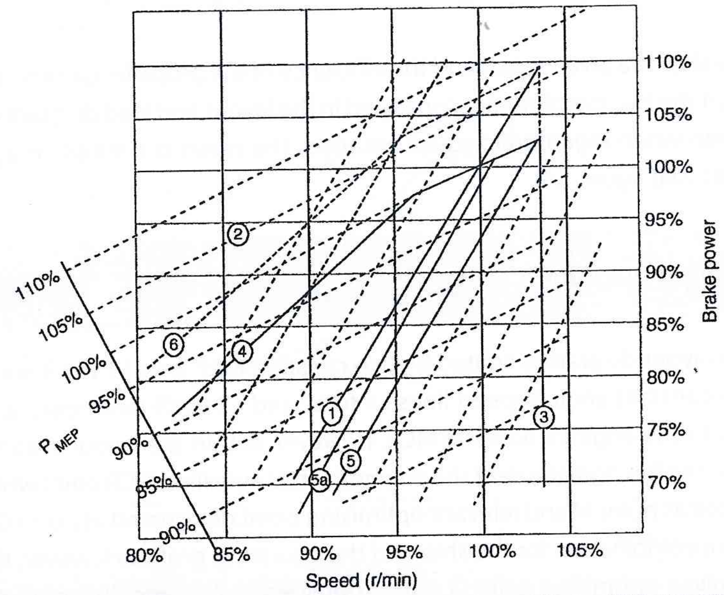


▲ **Figure 1.2** *Heat balance including waste heat recovery*

Analysis of the simplified heat balance shown in figure 1.2 reveals two important observations.

1. The difference between the indicated power and brake power is not only the power absorbed by the friction losses as some power is required to drive engine components such as camshafts, pumps, etc., which means a reduced potential for brake power.
2. Friction also results in heat generation which is dissipated by the various fluid cooling media, that is, oil and water, and hence the cooling analysis in a heat balance equation will include the frictional heat effect as an estimation.

Engine load diagram showing different combinations



▲ **Figure 1.3** *Standard engine load diagram*

speed, this will depend upon an adequate supply of charge air for combustion. Line 5 represents the power absorbed by the propeller when the ship is fully loaded and has a clean hull. The effect of a fouled hull is to move this line to the left as indicated by line 5a. In general a loaded vessel will operate between lines 4 and 5, while a vessel in ballast will operate in the region to the right of line 5. The area to the left of line 4 represents overload operation.

It can be seen that the fouling of the hull, by moving line 5 to the left, decreases the margin of operation and the combination of hull fouling and heavy weather can cause the engine to become overloaded, even though engine revolutions are reduced. Following on from this diagram the engine manufacturer will calculate the most efficient operating point for the engine. The operational requirements of the owner will determine the design speed and power for the normal running point of the engine (see layout information next).

Engine layout points

In designing engines for different types of duty the specific consumption minima may operate the ship at a given speed and the efficiency of the propeller can also be plotted on a graph. If the two functions are combined in the layout and load diagrams for diesel engines, then when logarithmic scales are used, the result is a simple diagram with straight lines (see figure 1.4).

Engine layout diagram

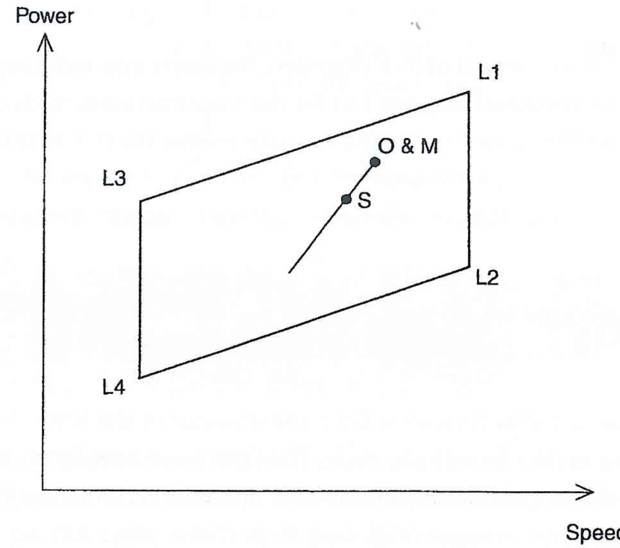
An engine's layout diagram is limited by two constant MEP lines L1 to L3 and L2 to L4, and by two constant engine speed lines L1 to L2 and L3 to L4 (see figure 2.1). The L1 point refers to the engine's nominal MCR. However, within the layout area the vessel designer has the freedom to select the engine's actual specified MCR point which would be designated as point M and relevant optimising point designated as point O, which is the optimum combination for the ship and the operating profile. However, the lowest SFOC for a given optimising point O will be obtained at 70% and 80% of the power at point O for both electronically and mechanically controlled engines, respectively.

Based on the best propulsion and engine running points, drawn up by the designer, the layout diagram of a relevant main engine may be drawn up. The specified MCR point M must be inside the limitation lines of the layout diagram. The optimised layout point of the engine is the rating at which the engine, timing and CR are adjusted to work most efficiently with the scavenge air pressure of the turbo-charger.

However, engines without variable injection timing (VIT) fuel pumps cannot be optimised at part-load. Therefore, these engines are always optimised at point L1.

Other information might also be included in these graphs by the engine manufacturers. Information such as:

- Propeller curve through an optimised point
- Layout curve for engine line
- Heavy propeller curve due to fouled hull and/or heavy seas
- Speed limit line
- Torque/speed limit
- MEP limit
- Light propeller curve clean hull and calm weather layout curve for propeller



▲ **Figure 1.4** *Engine layout points L1–L4*

Ideal Thermodynamic Cycles

Thermodynamic cycles are a series of operations carried out by a machine manipulating a substance. During the process heat and work are transferred by varying temperatures and pressures and eventually returning the system to its original state. The ideal thermodynamic cycles form the benchmark for reference against the actual performance of IC engines. In the cycles considered in detail all curves are regarded as frictionless adiabatic, that is, isentropic. The usual assumptions that are made are that constant specific heats and mass of charge are unaffected by any injected fuel, etc. and hence the expression *air standard cycle* may be used. There are two main classifications for reciprocating IC engines: (a) spark ignition (SI) such as petrol engines and, (b) CI such as diesel and oil engines. Liquid natural gas (LNG) is beginning to find favour with the main engine manufacturers due to its potential for producing less harmful emissions from the exhaust. Most engines designed to run on gas are currently using the Otto (SI) process, however, MAN Diesel & Turbo have recently conducted trials of a two-stroke engine operating on the Diesel process.

drawn using the usual method of P–V diagrams. Research into reducing the exhaust emissions from marine diesel engines has let the manufacturers to develop engines operating by using the 'Miller' cycle. The primary reason for this is that the highest temperatures of combustion are avoided and therefore the harmful nitrous oxides (NOx) are not produced to then be released into the atmosphere through the exhaust.

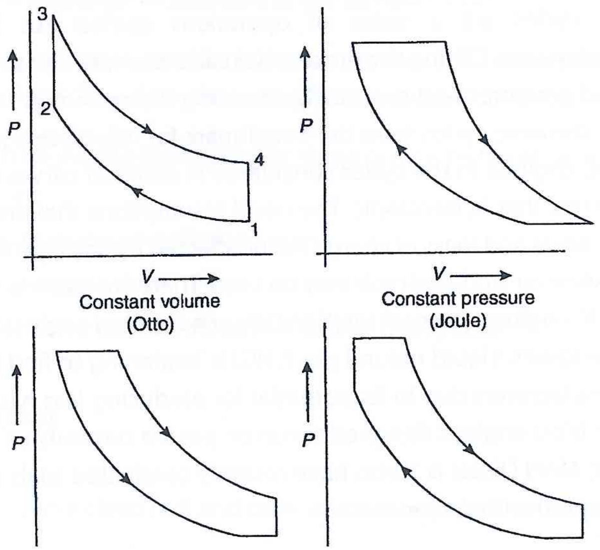
Otto (constant volume) cycle

Otto cycle was named after Nickolaus Otto, the inventor of the first efficient working IC engine working on the four-stroke cycle. The Otto cycle now forms the basis of all SI and high-speed CI engines. The four non-flow operations combined into a cycle are shown in figure 1.5.

$$\text{Air standard efficiency} = \text{work done/heat supplied}$$

$$= \frac{(\text{heat supplied} - \text{heat rejected})}{\text{heat supplied}},$$

referring to figure 1.5.



$$\begin{aligned}\text{Air standard efficiency} &= 1 - \text{heat rejected/heat supplied}\\ &= 1 - MC(T_4 - T_1)/MC(T_3 - T_2), \text{ where } M \text{ is the mass and}\\ &\quad\ C \text{ is the specific heat capacity of the substance}\\ &= 1 - 1/(r\gamma^{-1})\end{aligned}$$

[using $T_2/T_1 = T_3/T_4 = r\gamma^{-1}$, where r is the CR].

Note: Efficiency of the cycle increases with an increase in the CR. This is also true for the other four cycles.

Diesel (modified constant pressure) cycle

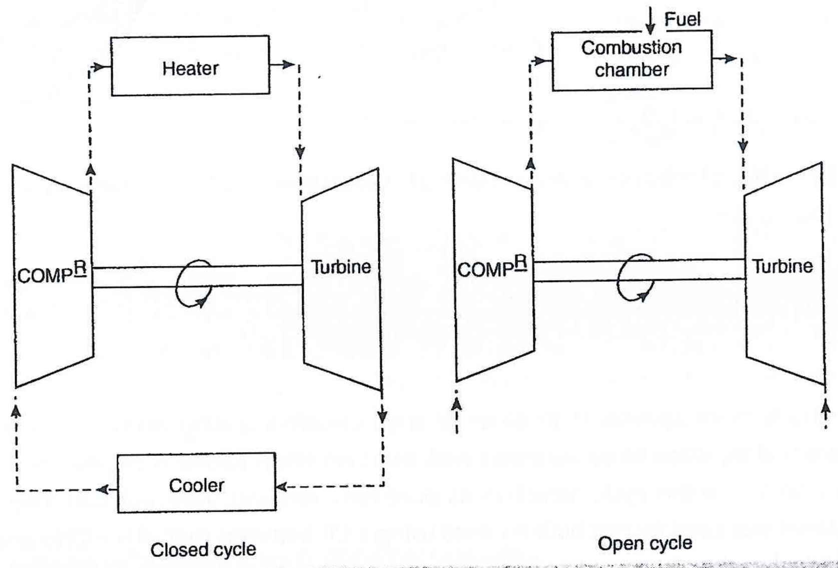
This cycle is more applicable to older CI engines utilising long periods of constant pressure fuel injection in conjunction with blast injection. Modern engines do not in fact aim to follow this cycle, which in its pure form requires very high CRs. The term semi-diesel was used for hot bulb engines using a CR between that of the Otto and the Diesel ideal cycles. Some very early Doxford engines utilised a form of this principle with low compression pressures and 'hot spot' pistons. The Diesel cycle is also sketched in figure 1.5 and it should be noted that heat is received at constant pressure and rejected at constant volume.

Dual (mixed) cycle

This cycle is applicable to most modern CI reciprocating IC engines. Such engines employ solid injection with short fuel injection periods fairly symmetrical about the firing dead centre. The term semi-diesel was often used to describe engines working close to this cycle. In modern turbo-charged marine engines the approach is from this cycle almost to the point of the Otto cycle, that is, the constant pressure period is very short. This produces very heavy firing loads but gives the necessary good combustion.

Joule (constant pressure) cycle

This is the simple gas turbine flow cycle. Designs at present are mainly of the open cycle type although nuclear systems may well utilise closed cycles. The ideal cycle P–V

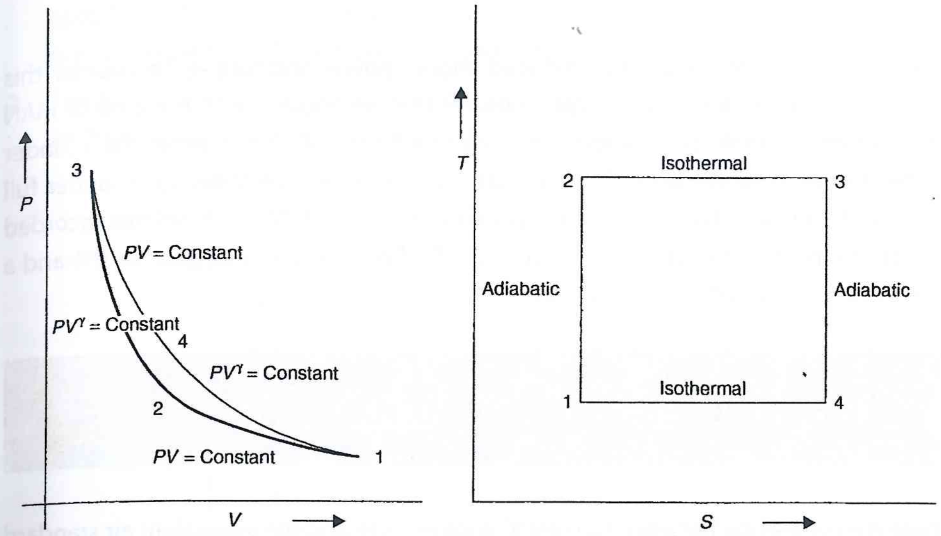


▲ **Figure 1.6** *Gas turbine circuit cycles*

Other cycles

The efficiency of a thermodynamic cycle is a maximum when the cycle is made up of reversible operations. The Carnot cycle of isothermals and adiabatics satisfies this condition and this maximum efficiency is, referring to figure 1.7, given by $(T_3 - T_1)/T_3$ where the Kelvin temperatures are maximum and minimum for the cycle. The cycle is practically not approachable as the MEP is so small and CR would be excessive. All the four ideal cycles have efficiencies less than the Carnot. The Stirling cycle and the Ericsson cycle have equal efficiency to the Carnot. Further research work is being carried out on Stirling cycle engines in an effort to utilise the high thermal efficiency potential. The Carnot cycle is sketched on both *P–Y* and *T–S* axes (figure 1.7). However, as with all these theoretical cycles the reality of producing a practical working engine running on one is very difficult. Therefore, actual engines are always a compromise.

Miller cycle



▲ **Figure 1.7** *Theoretical (ideal) cycles*

This engine cycle cannot be used unless an engine has full electronic control of both the fuel injection process and the ability to vary the valve operating timing. The temperature peaks during combustion are responsible for over 90% of NOx formation. Therefore, manufacturers use 'primary' combustion measures to eliminate the peak temperatures in the combustion chamber without incurring fuel consumption penalties or, if possible, at improved fuel efficiency. To achieve this, a range of engine modifications have been used, including:

- further cooling of the charge air
- improved re-entrant piston bowls
- low swirl inlet ports
- higher CRs
- higher fuel injection pressures and improved injector nozzle spray patterns
- revised fuel injection timing
- a combination of revised 'Miller cycle' valve timing and high efficiency, high pressure turbo-charging.

The Miller cycle involves the early closure of the inlet valve, causing the air entering the cylinder to expand ...

entering the cylinder and hence reduced engine power and torque. To counter this effect, higher pressure turbocharging ensures that an equal – or in the case of MAN Diesel's new technology package – or greater amount of air can enter the cylinder in the shorter time available. During trials using an intensive Miller cycle under full load conditions and turbocharger pressure ratios of 6.5–7, MAN Diesel has recorded reductions in NOx of over 30%, reductions in fuel consumption as great as 8% and a 15% increase in specific power output.

Actual Cycles and Indicator Diagrams

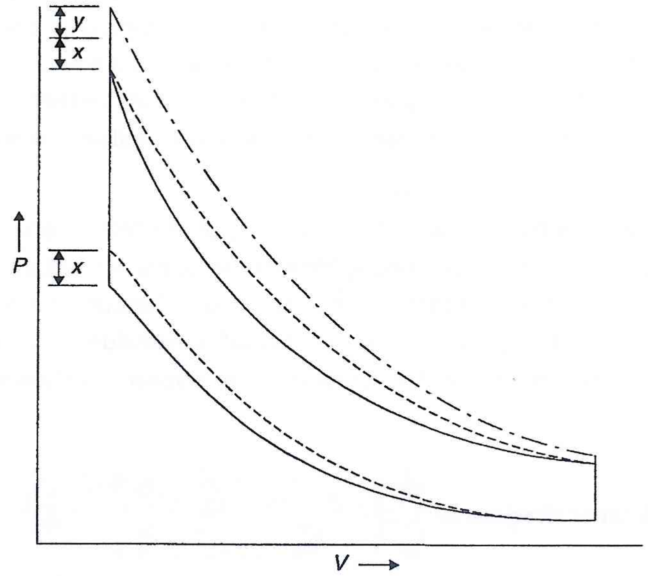
There is a correlation between the real IC engine cycle and the equivalent air standard cycle as is shown by the similarities in the *P–V* diagrams but how does this help the marine engineer working in remote places away from any substantial support.

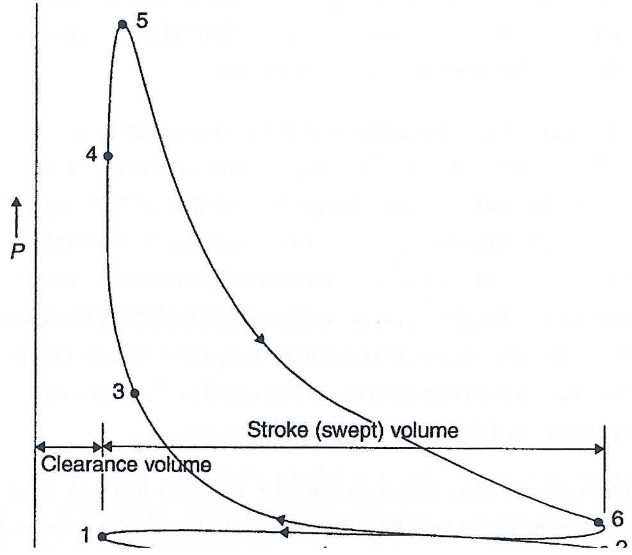
For many years engineers recorded the pressure in each cylinder by the use of a mechanical cylinder pressure indicator apparatus (fig 1.14). These were small hand-held units made up of a piston which was able to move up and down in a cylinder. The machine was connected to the engine via a cock on the engine called the indicator cock. Opening the cock allowed the full force of combustion through to the indicator equipment's cylinder. The pressure from the combustion acts upon the piston operating within the cylinder. Movement of the piston is restricted by a spring which can be changed to match the different combustion pressures of different engines. The vertical movement of the piston drives an arm, at the end of which is a pointer, that is used to draw the vertical line on a 'card' corresponding to the pressure in the cylinder. The horizontal movement of the card is achieved by rotating the drum that the card is attached to, in time with the movement of the piston. The movement was achieved by using a chord attached to a roller on the camshaft of the engine. Using the equipment shown in figure 1.14, an actual diagram was produced as shown in the lower drawing of figure 1.8. This allowed the engineer to assess the efficiency of the combustion process. Only one cylinder could be reviewed at a time but the different cards could then be compared alongside each other.

The differences between the cycles can now be considered and for illustration purposes the drawings given are of the Otto cycle. The principles are generally the same for most

specific heats, a reduction in γ due to gas-air mixing, etc. The resulting compression is not adiabatic and the difference in vertical height is shown as *x*.

2. The actual combustion gives a lower temperature and pressure than the ideal due to dissociation of molecules caused by high temperatures. These twofold effects





can be regarded as a loss of peak height of $x + y$ and a lowered expansion line below the ideal adiabatic expansion line. The loss can be regarded as clearly shown between the ideal adiabatic curve from maximum height (shown as chain dotted) and the curve with initial point $x + y$ lower (shown as dotted).

3. In fact the expansion is also not adiabatic. There is some heat recovery as molecule re-combination occurs but this is much less than the dissociation combustion heat loss in practical effect. The expansion is also much removed from adiabatic because of heat transfer taking place and variation of specific heats for the hot gas products of combustion. The actual expansion line is shown as a full line on figure 1.8.

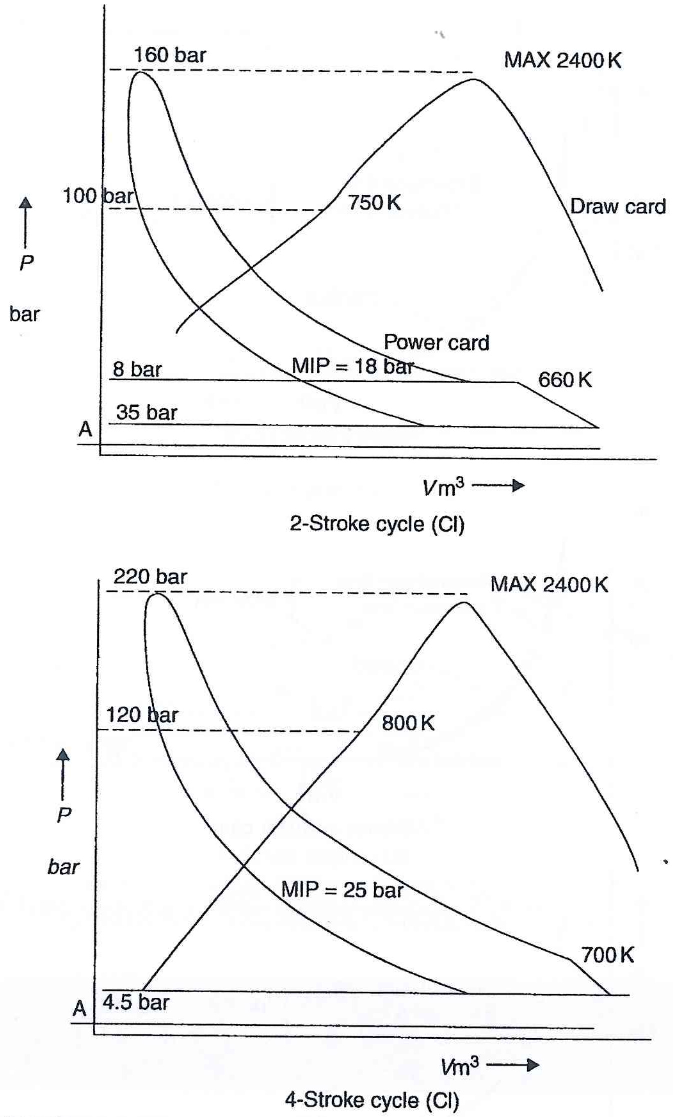
In general the assumptions made at the beginning of the section on ideal cycles are worth repeating, that is, isentropic, negligible fuel charge mass, constant specific heats, etc. plus the comments above such as for example on dissociation. Consideration of these factors plus practical details such as rounding of corners due to non-instantaneous valve operation, etc. mean that the actual diagram appears as shown in the lower sketch of figure 1.8.

Typical indicator diagrams

The power and draw cards are shown in figure 1.9 and should be studied closely. These examples are for two- and four-stroke CI engines and the typical temperatures and pressures are shown on the drawings where appropriate.

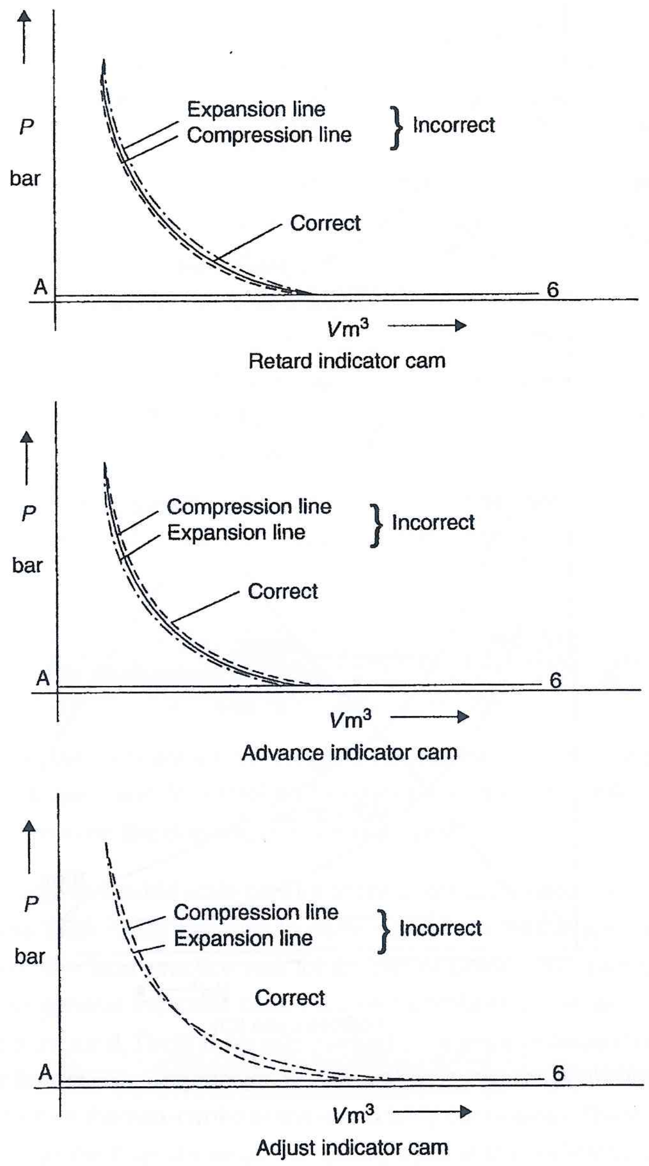
The draw card is an extended scale picture of the combustion process. They have been given the name 'draw cards' because in early marine practice the indicator card was drawn by hand. The later practice was for an 'out-of-phase' (90°) cam to be provided adjacent to the general indicator cam. Incorrect combustion details are highlighted by taking the draw card. There is no real marked difference between the diagrams for two-stroke or four-stroke. In general the compression point on the draw card is more difficult to detect on the two-stroke as the line is fairly continuous. There is no induction – exhaust loop for the four-stroke as the spring used in the indicator is too strong to discriminate on a pressure difference of say 1/3 bar only.

Compression diagrams are also given in figure 1.10; with the fuel shut off expansion and compression appearing as one line. Errors would be due to a time lag in the drive

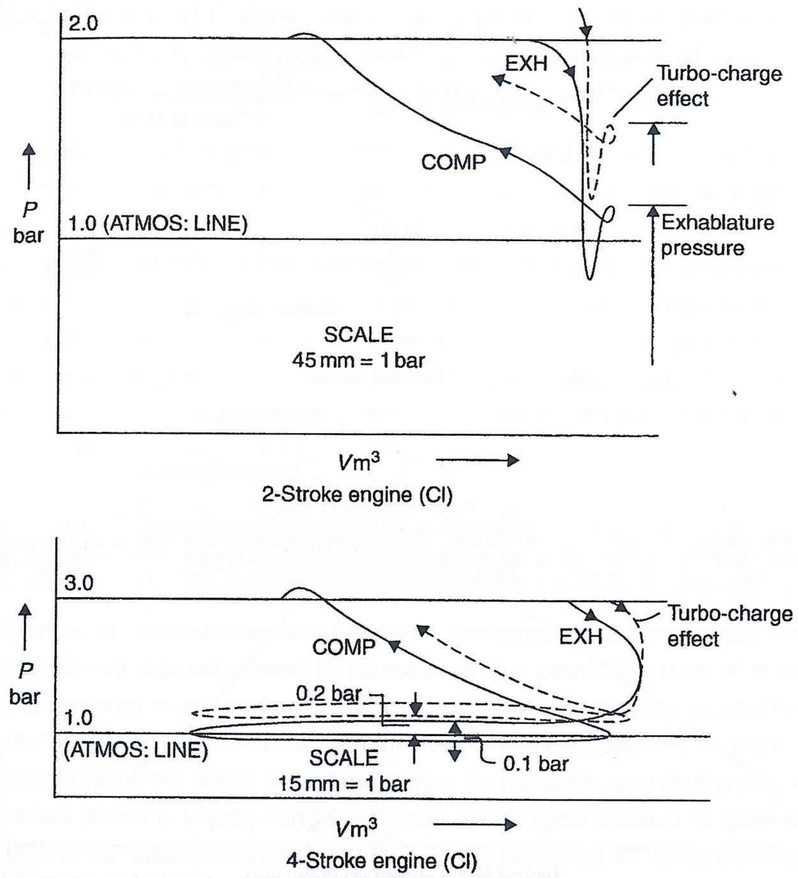


▲ **Figure 1.9** *Typical indicator (power and draw) diagrams*

Figure 1.11 shows the light spring diagrams for CI engines of the two- and four-stroke types. These diagrams are particularly useful in modern practice to give information about the exhaust – scavenge (induction) processes as all main engines are now turbo-charged. The turbo charge effect is shown in each



▲ **Figure 1.10** *Compression diagrams*

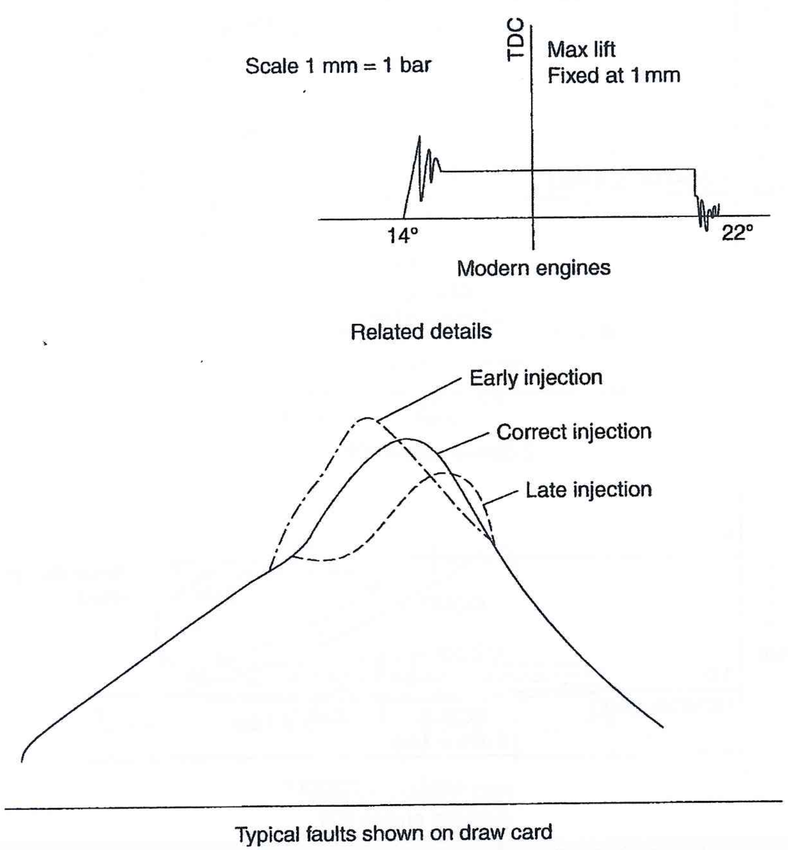


▲ **Figure 1.11** *Typical indicator (light spring) diagrams*

Other Related Details

Fuel valve lift cards are useful to obtain characteristics of injectors when the engine is running. A diagram given in figure 1.12 shows the outcome of such a card. Electronic system can easily be set up to obtain such a result even on high-speed engines. This is an example of the advantage of the electronic system over the older mechanical ones and is a common question asked by the flag state examiner.

Typical diagram faults are normally best considered in the particular area of study



▲ **Figure 1.12** *Fuel valve lift diagrams*

perhaps be stated that before attempting to analyse possible engine faults it is essential to ensure that the indicator itself and the drive are free from any defect.

CR has been discussed previously and with SI engines the limits are pre-ignition and detonation. Pinking (knocking) and its relation to octane number are important factors as are anti-knock additives such as lead tetra-ethyl, $Pb(C_2H_5)_4$. Factors more specific to CI engines are ignition quality, Diesel knock and Cetane number, etc. In general these factors plus the important related topics of combustion and the testing and use of lubricants and fuels should be particularly well understood and reference should be made to the appropriate chapter in Volume 8 of Reed's series. This is especially true for

measuring device used to determine the area, planimeter, are therefore significant. Multiplication by high spring factors makes errors in evaluation of mean indicated pressure (MIP) also significant and certainly of the order of at least ±4%.

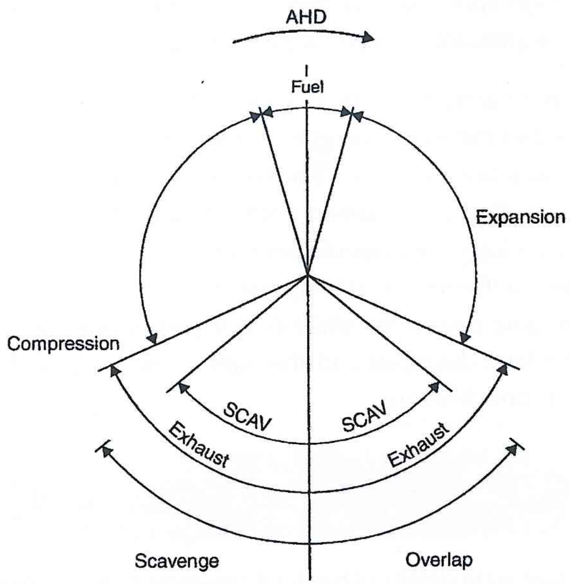
Further application of engine constants gives indicated power calculations having similar errors. Provided the inaccuracies of the final results are appreciated then the real value of the diagrams can be established. Power card comparison is probably the most vital information to be gained from indicator diagrams. However, modern practice using mechanical devices would perhaps favour maximum pressure readings, equal fuel quantities, uniform exhaust temperature, etc. for cylinder power balance and torsionmeter for engine power. The draw card is particularly useful for compression and/or combustion fault diagnosis and the light spring diagram for the analysis of scavenge – exhaust considerations.

Turbo-charging

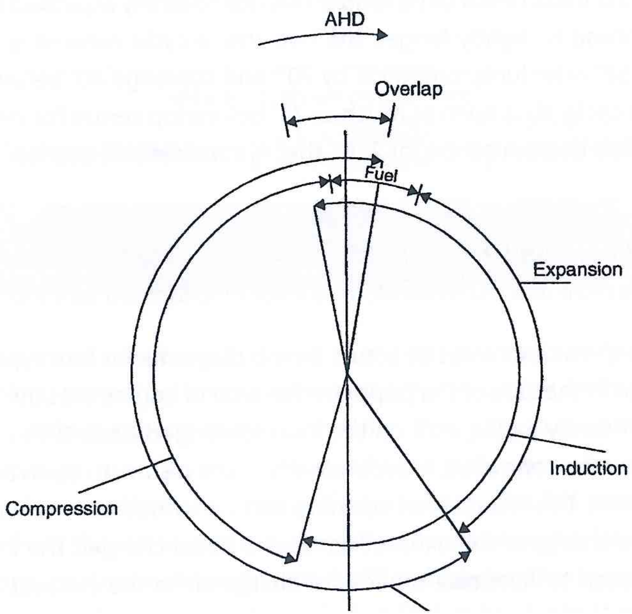
This subject is considered in detail in Chapter 4, however, one or two specific comments relating to timing diagrams need to be made at this point. The start of the exhaust process is required much earlier in turbo-charged engine to drop exhaust pressure quickly before the induction of the next air charge. The time allowed for the discharge of the greater gas mass needs to be longer than for naturally aspirated engines. As the air induction phase is slightly longer, the two-stroke cycle exhaust is open from 76° before BDC to 56° after (unsymmetrical by 20°) and scavenge 40° before and after. For the four-stroke cycle, air is open as much as 75° before top centre for 290° and exhaust is open 45° before bottom centre for 280°, that is, considerable overlap.

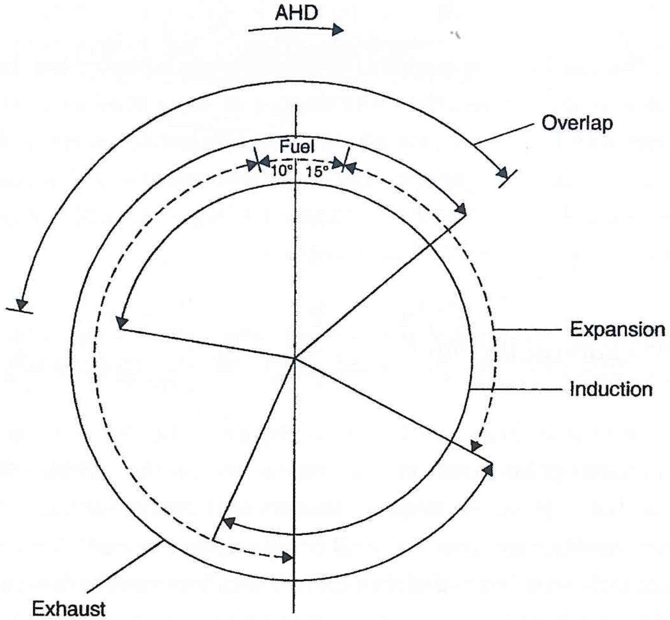
Actual timing diagrams

Figure 1.13a–d shows examples of actual timing diagrams for four types of engine. It will be seen that in the case of the poppet valve type of engine that the exhaust opens at a point significantly earlier than on the loop scavenged design. This is because the exhaust valve can be controlled, independently of the piston, to open and close at the optimum position. This means that opening can be carried out earlier to effectively utilise the pulse energy of the exhaust gas in the turbo-charger. The closing position can also be chosen to minimise the loss of charge air to the exhaust. With the loop

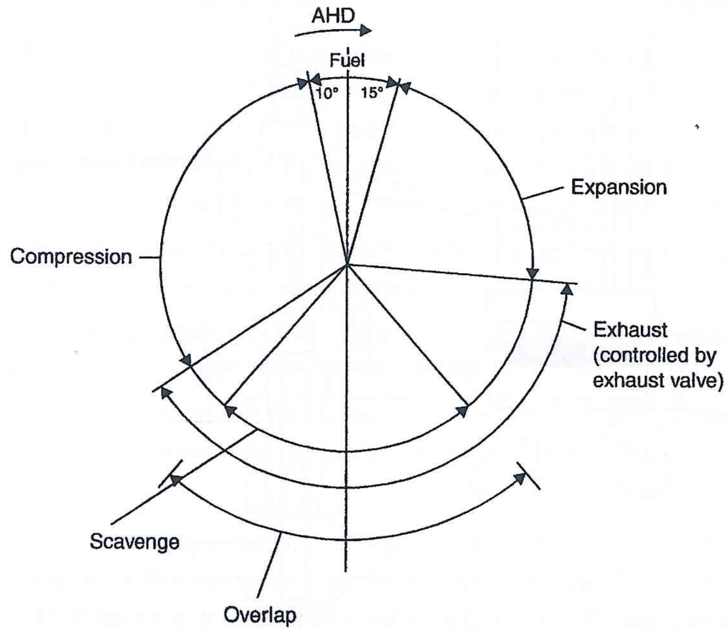


▲ **Figure 1.13a** *Crank timing diagram for two-stroke loop scavenged turbo-charged engine. Exhaust and scavenge symmetrical about bottom dead centre*





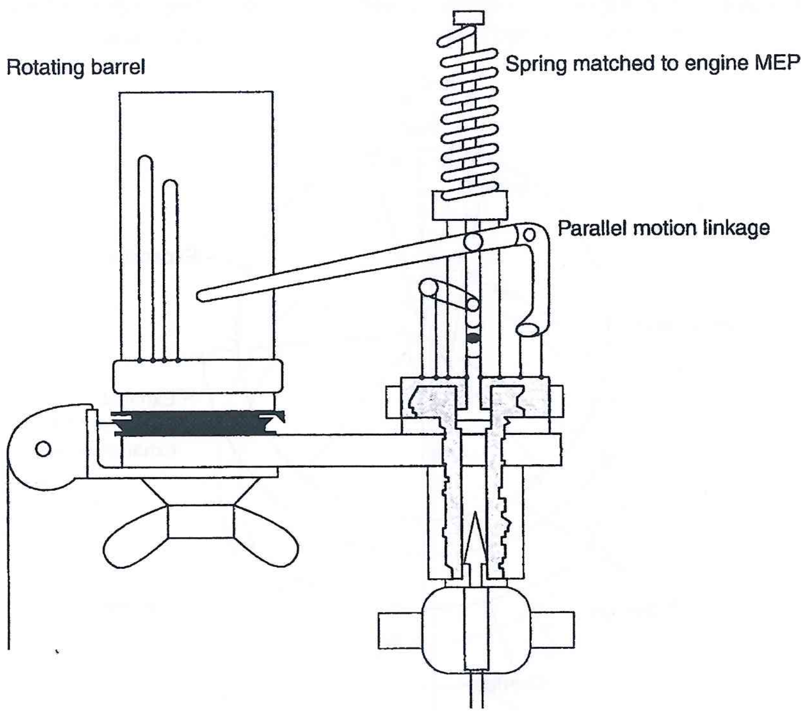
▲ **Figure 1.13c** *Four-stroke turbo-charged engine*



Comparison of the crank timing diagrams of the naturally aspirated and turbo-charged four-stroke diesel engine shows the large degree of valve overlap on the latter. This overlap together with turbo-charging allows more efficient scavenging of combustion gases from the cylinder. The greater flow of air through the turbo-charged engine also cools the internal components and supplies a larger mass of charge air into the cylinder prior to commencement of compression.

Types of indicating equipment

Conventional indicator gear is shown in figure 1.14; its precise details and manufacturer's descriptive literature can be found on the LEHMANN & MICHELS GmbH website. For high-speed engines an indicator of the 'Farnboro' type is often used. Maximum combustion pressure and compression pressure can be taken using a peak pressure indicator. The fuel is shut off to the cylinder being measured to obtain the compression pressure.



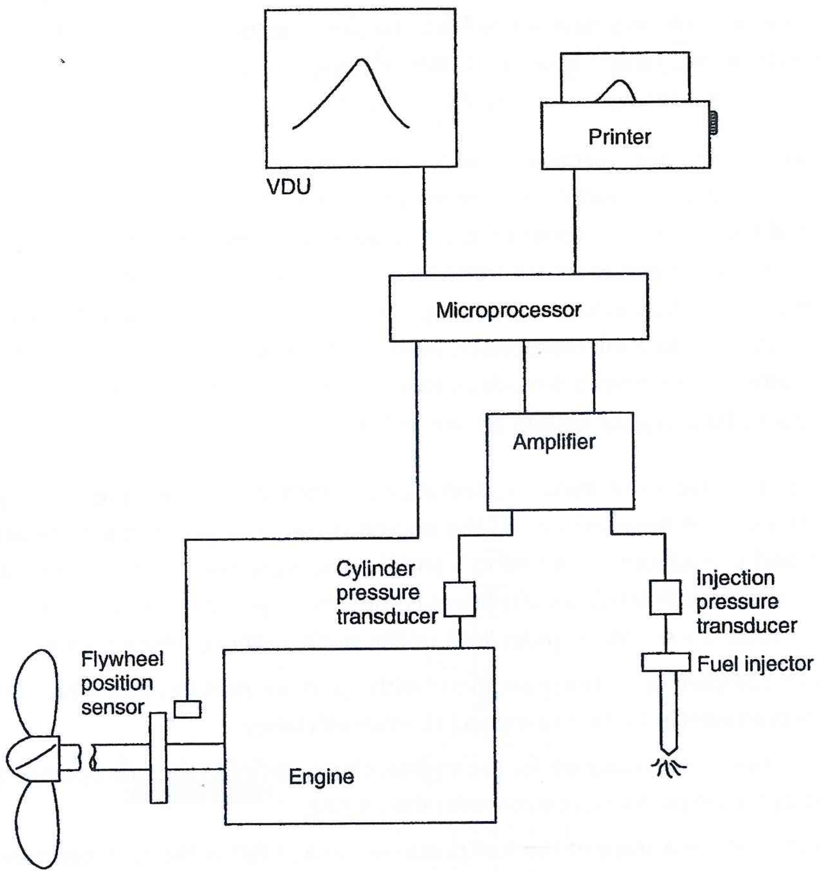
Electronic indicators

The limitations of mechanical indicating equipment have been overtaken by the need for greater accuracy and a higher level of detail as engine powers have risen and especially now as electronic combustion control has become normal practice. With outputs reaching 5500 hp per cylinder inaccuracies of ±4.0% will lead to large variations in indicated power and therefore attempts to balance the engine power by this method will only have limited success. The inaccuracies stem from the friction and inertia of mechanical equipment transmitting errors in recording as well as the inaccuracy of measuring the height of the power card.

Modern practice utilises electronic equipment to monitor and analyse the cylinder peak pressures and piston position and presents the information on a display which could be part of the analyser or could be the display on a computer. The cylinder pressure is measured by a pressure transducer which is attached to the indicator cock. Engine position is detected by a magnetic pick-up in close proximity to a toothed flywheel. The information is fed to a microprocessor, where it is averaged over a number of engine cycles, before calculations are made as to indicated power and MEP (figure 1.15). The advantages of this type of equipment are as follows:

1. It supplies real-time dynamic operational information. Therefore, the injection timing can be measured while the engine is running, which is a more accurate method of measuring the timing – and if sensors are placed on the crankshaft at each unit as with Wärtsilä's 'intelligent combustion' system – it allows for crankshaft twist while the engine is under load, unlike static methods which do not.
2. It can compare operating conditions with optimum performance which leads to improvements in fuel economy and thermal efficiency.
3. It produces a load diagram for the engine, clearly defining the safe operating zone and optimum performance zones for the engine.
4. It can produce a trace of the fuel pressure rise and fall in the high pressure lines, which is invaluable information when diagnosing fuel injection faults.

Early operational experience with this type of equipment pointed to problems of tunreliability with the pressure transducers when connected continuously to the engine. To overcome this problem manufacturers briefly experimented with cylinder pressure this information can be fed to the microprocessor. However, modern production techniques have improved the situation and there are now a number of different sensor technologies that are used for detecting and transmitting a high pressure that is also fluctuating at a high frequency but the most common technique is the resistive measurement method because it has the advantage of referencing the measured pressure to the ambient conditions.



▲ **Figure 1.15** *Basic electronic combustion indicator equipment*

One construction method consists of using two coating techniques. The first step is to electrically insulate the active sensor components from the body of the device by using a thin silicon dioxide (SiO_2) layer which is placed on the sensor using a chemical vapour

conventional lithographic techniques are used to create the conductor paths typically used in strain gauges.

Modern designs are helped by the use of finite element modelling (FEM). FEM analysis can be used to investigate sensors under different load conditions and their performance is evaluated so that designs can then be modified accordingly.

Electrical connection from the strain gauge on the cantilever beam to a printed circuit board (PCB) is made with the help of wire bonds in order to mechanically decouple the sensor element from the electronics and the output signal from the Wheatstone bridge, which is generated by the very slight stretching of the diaphragm under the action of the pressure that is being measured.

The accuracy of modern sensors are now well below 1% and in some cases the error amounts to a few bar over the operating range of 2500 bar and they also have to conform to the marine quality standards for the equipment.

The quality process could include highly accelerated lifetime testing such as subjecting the sensors to a series of tests which are designed to exceed the sensor specifications. The vibration test for example involves sensors with a specified vibration tolerance being tested outside their design conditions and at a high temperature range than expected in service, over a long time period.

By using the latest, economically priced sensors to monitor fuel injector needle lift and cylinder pressure the combustion process in each cylinder can be individually optimised, thereby allowing operators to reduce emissions, increase fuel efficiency and monitor engine health more closely.

The increased reliability of these systems allows combustion and fuel injection monitoring equipment to be permanently installed giving real-time data to inform engine performance and maintenance.

Variable valve timing (VVT)

With the modern electronic control and hydraulic/electrical actuation of components affecting engine combustion manufacturers can now accomplish so much more than that in the past. VVT is one of the areas of combustion control that has had a big influence on the operation of engines.

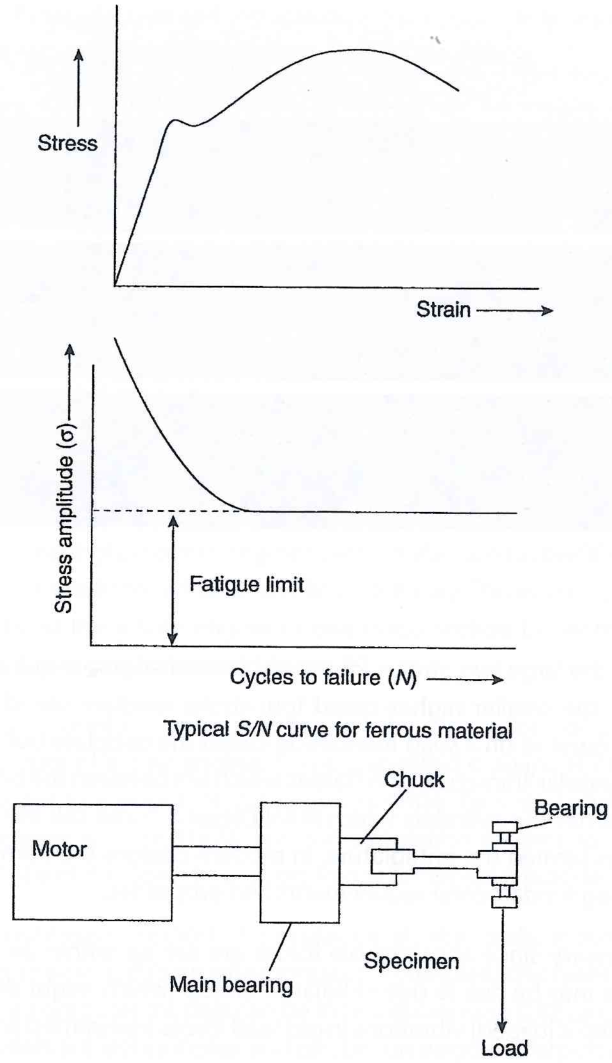
The problem with mechanical operation, using camshaft control of the inlet and

VVT enables the use of the Miller cycle, as described above, without the disadvantage of the very poor part load operation. If the Miller cycle is used at part load then the combustion is starved of air and a large amount of smoke it produced by the engine. The use of VVT technology overcomes this and allows the inlet and exhaust valve timing to be optimised for all engine loads. It also allows the Miller cycle to be switched on and off to match the engine operating conditions.

Fatigue

Fatigue is a phenomenon which affects materials that are subjected to cyclic or alternating stresses. Designers will ensure that the stress induced in a component is below the yield point of the construction material as measured on the familiar stress/strain graph (see Reed's series, Volume 2). However, if that component is subjected to constant cyclic stresses it may fail at a lower value due to fatigue. The most common method of displaying information on fatigue is the *S–N* curve (figure 1.16). This information is obtained from fatigue tests usually carried out on a Wohler machine in which a standard specimen is subjected to an alternating stress due to rotation. The specimen is tested at a particular stress level until failure occurs. The number of cycles to failure is plotted against stress amplitude on the *S–N* curve. Other specimens are tested at different levels of stress. When sufficient data have been gathered a complete curve for a particular material may be presented.

It can be seen from figure 1.16 that, in the case of ferrous materials, there is a point known as the 'fatigue limit'. Components stressed below this level can withstand an infinite number of stress reversals without failure. Stress is equal to the load divided by the cross-sectional area (CSA) of a given material; therefore, we can deduce that reducing the stress level on a component involves increasing the CSA resulting in a weight penalty or reducing the load. In marine practice the weight implications are generally regarded as secondary to reliability and long life and so components are usually stressed below the fatigue limit. This is not the case in, for example, aeronautical practice where weight is a major consideration. In this situation the component designer would compromise between weight and stress levels and from the *S–N* curve would calculate, with the addition of a safety margin, the number of cycles the component could withstand before failure occurs. A marine example is that the working life of four-stroke medium-speed diesel bottom end bolts are calculated



▲ **Figure 1.16** *Wohler machine for zero mean stress fatigue testing*

2 STRUCTURE AND TRANSMISSION

The structure of the large two-stroke, low-speed marine engine is quite different from the structure of the smaller higher-speed four-stroke medium-speed engines. Both types of engine must sit on a solid foundation called the bedplate but the two-stroke engine has a triangular 'framebox' or 'A' frames which sits between the bedplate and the structure that carries the cylinders which MAN Diesel & Turbo call the cylinder frame but is sometimes termed the entablature. In modern designs the frame section must have the following fundamental requirements and properties.

Strength is necessary since considerable forces are set up within an engine as it is operating. These may be due to out-of-balance effects (which might differ depending upon how the ship is loaded) vibrations in gas load forces transmitted and gravitational forces.

Rigidity is required to maintain correct alignment of the engine running gear. However, a certain degree of flexibility will prevent high stresses that could be caused by slight misalignment.

Lightness is important as it may enable the increase of power to weight ratio. Less material would be used, bringing about a saving in cost. Both are important selling points as they would give increased cargo capacity or reduced fuel consumption.

Simple design – if manufacture and installation are simplified then a saving in cost will be realised with reduced maintenance time and less down time.

Access – ease of access to the engine transmission system for inspection, maintenance and installation is a fundamental requirement.

Dimensions – ideally these should be as small as possible to keep engine containment to a minimum in order to give more engine room space. One of the drawbacks of the two-stroke engine is the empty space required above the cylinder head. This is due to the requirement for the piston rod to be withdrawn vertically at the same time as the piston when the piston needs maintenance, such as the replacement of the piston rings.

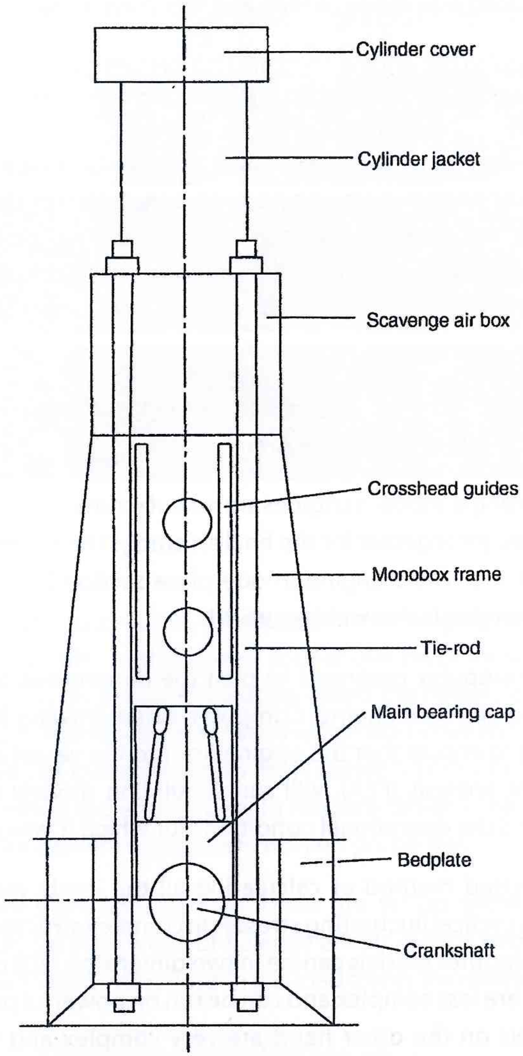
Seal – the engine and transmission system container must effectively seal in the oil and any vapours generated, from the engine room.

Manufacture – increasingly, modern engines are manufactured in prefabricated sections with the sections brought together for the final assembly. This is a much easier method than trying to build the whole engine in one place section-by-section. The general arrangement of the engine is shown in figure 2.1.

The designers use computer programs to plan the dimensions of the components and overall structure of a new engine. Computer-aided drawing (CAD) software will enable the designer to ensure that the engine fits into the vessel and other software called finite element analysis (FEA), will run simulation models to work out if the engine can withstand the operational conditions for which it was designed.

FEA – is a computerised method of calculating all the loads, mechanical stresses, thermal stresses and cyclical fluctuating stresses in complex structures. The algorithms that make up the computer models can be in two dimension (2D) or three dimension (3D). The 2D models are less complex and can be run on powerful personal computers (PCs). The 3D models on the other hand are very complex and would have to be run on mini or mainframe computers. These techniques have enabled a substantial increase in the designers' ability to design accurately stressed engines and indeed whole vessels.

The structure of the modern two-stroke engine must be rigid and able to withstand the tremendous forces imposed by the combustion pressures set up by these high performance engines. Full 3D stresses and bending moments are calculated on each engine design before it is put into production or a prototype is built.



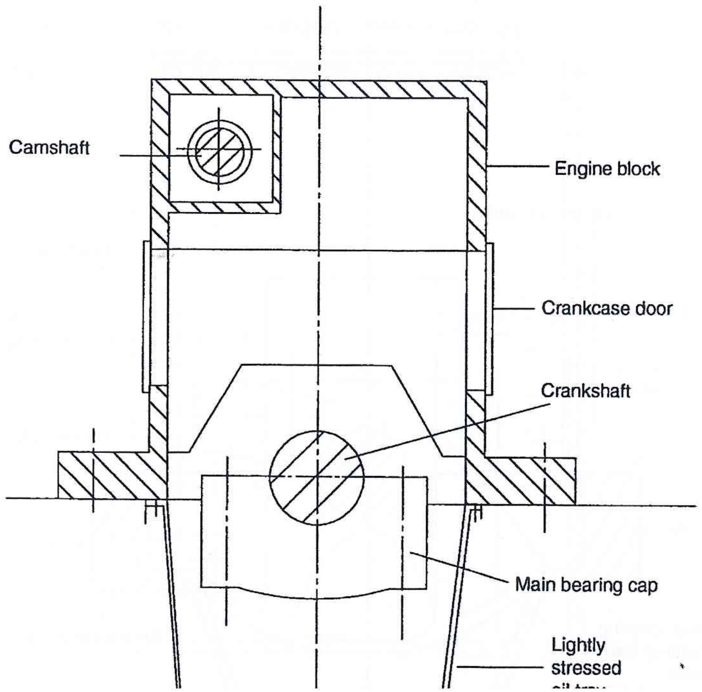
▲ **Figure 2.1** *General arrangement of engine structure*

Bedplate

The engine bedplate is a structure that was traditionally made of cast iron or cast steel

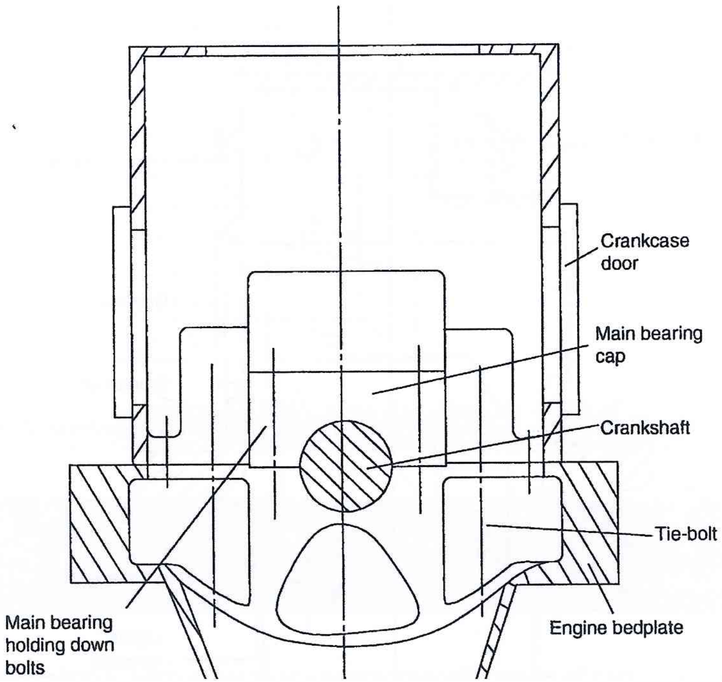
Cast iron one-piece structures are generally confined to the smaller high- or medium-speed engines rather than the larger slow-speed crosshead type engine. This is due to the problems that are encountered with the quality control of large castings. These problems include poor flow of material to the extremities of the mould, poor control of the grain size, which leads to a lack of homogeneity of strength and soundness, and poor impurity segregation. In addition to these problems cast iron has poor performance in tension and its modulus of elasticity is only half that of steel hence for the same strength and stiffness a cast iron bedplate will require to be manufactured from more material. This results in a weight penalty for larger cast iron bedplates when compared with a fabricated bedplate of similar dimensions. Cast iron does, however, enjoy certain advantages for the construction of smaller medium- and high-speed engines which are as follows:

- Castings do not require heat treatment.
- Cast iron is easily machined and is good in compression.
- The master mould can be re-used many times which results in reduced manufacturing costs for a series of engines.



- The noise and vibration damping qualities of cast iron are superior to that of fabricated steel.
- As outputs increase nodular cast iron, due to its higher strength, it is becoming more common for the manufacture of medium-speed diesel engine bedplates.
- Modern cast iron bedplates for medium-speed engines are generally, but not exclusively, a deep inverted 'U' shape, which affords maximum rigidity for accurate crankshaft alignment. The crankcase doors and relief valves are incorporated within this structure. In this design the crankshaft is 'underslung' and the crankcase is closed with a light unstressed oil tray (figure 2.2).
- As outputs of medium-speed engines increase some manufacturers choose the alternative design in which the crankcase and bedplate are separate components; the crankshaft being 'embedded' in the bedplate (figure 2.3).

Now that welding techniques and methods of inspection have improved and larger furnaces are available for annealing, the switch to prefabricated steel structures with their saving in weight and cost has been made where the advantages are realised. It

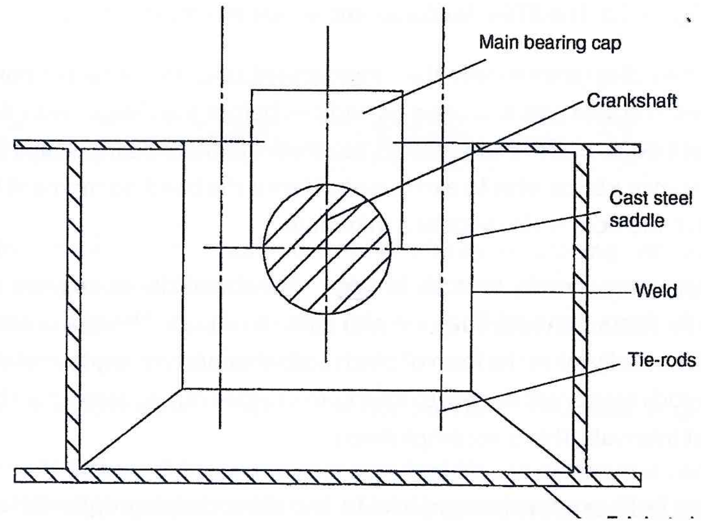


must be remembered that the modulus of elasticity for steel is nearly twice that of cast iron, hence for similar stiffness of structure roughly half the amount of material would be required when using steel (Young's modulus of elasticity is given the symbol '*E*' and is a measure of the stress to strain ratio of a material) – cold draw steel = 200 GPa and grey cast iron = 124 GPa.

Early designs were entirely fabricated from mild steel but radial cracking due to cyclic bending stress imposed by the firing loads was experienced on the transverse members especially in way of the main bearings. The adoption of cast steel, with its greater fatigue strength, for transverse members has eliminated this cracking. Modern large engine bedplates are constructed from a combination of fabricated steel and cast steel. Modern designs consist of a single walled structure fabricated from steel plate with transverse sections incorporating the cast steel bearing saddles attached by welding (figure 2.4). To increase the torsional, longitudinal and lateral rigidity of the structure suitable webbing is incorporated into the fabrication.

It is modern practice to cut the steel plate using automatic contour flame cutting equipment. Careful preparation is essential prior to the welding operation because:

- It is necessary to prepare the edges of the cut plate and therefore it is also necessary to make an allowance for this during the initial cutting phase.
- The equipment needs to be set correctly to ensure the smallest heat-affected zone (HAZ).



- Welding consumables are stored and used correctly to prevent hydrogen contamination of the HAZ which could lead to post-annealing hydrogen cracking.

Following the welding operation the welds are inspected for surface cracking and sub-surface flaws. The surface inspection is carried out by the dye-penetrant method or the magnetic particle method while the sub-surface flaws are inspected by ultrasonic testing. Any flaws found in the welds would be cut out, re-welded and tested.

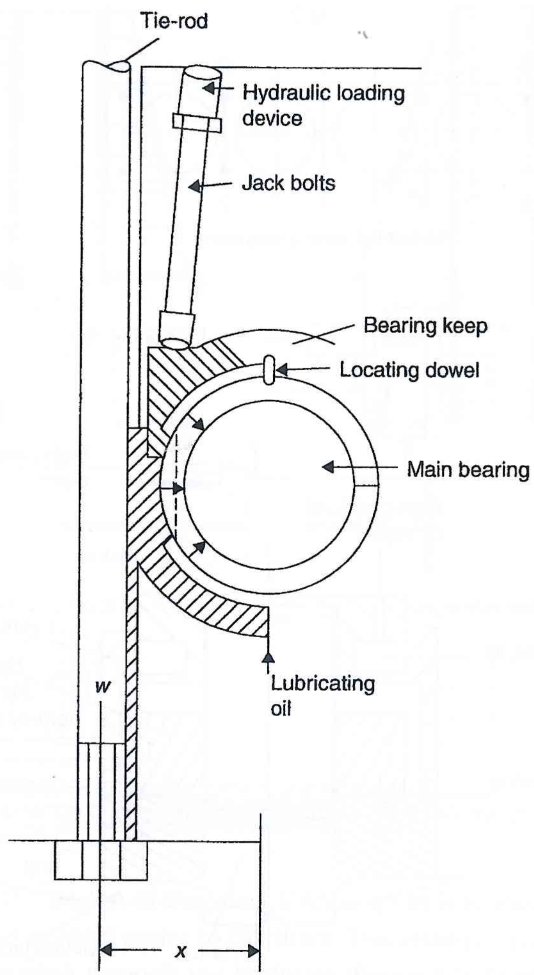
The bedplate is then stress relieved by heating the whole structure to below the lower critical temperature of the material in a furnace and allowing it to cool slowly over a period of days. When cooled, the structure is shot blasted and the weld is again tested before the bedplate is machined.

In order to minimise stresses due to bending in the bedplate, without a commensurate increase in material, tie-rods are used to absorb the combustion forces. Two tie-rods are fitted to each transverse member and pass, in tubes, through the entire structure of the engine from bedplate to cylinder cooling jacket (entablature). They are pre-stressed at assembly so that the engine structure is under compression at all times (including operation). Initially engines that utilised the opposed piston principle had their combustion loads absorbed by the running gear and therefore did not require tie-rods. Modern engines employ tie-rods or stay bolts as MAN prefer to call them and to minimise bending moment forces across the main bearing these are placed as close as possible to the crankshaft centre line. In some cases this has led to the use of 'jack bolts' as shown in figure 2.5. The RT84 flex for example uses this arrangement.

Figure 2.5 shows diagrammatically the arrangement used in the larger bore Wärtsilä Sulzer engines. The idea was first used on engines before the merger with Wärtsilä. By employing jack bolts, under compression, to retain the main bearing caps in position allows the distance x to be kept to a minimum. Hence the bending moment Wx, where W is the load in the bolt, is also kept to a minimum.

Owing to their great length, tie-rods in large slow-speed diesel engines may be in two parts to facilitate removal. They are also liable to vibrate laterally unless they are restrained. This usually takes the form of pinch bolts that prevent any lateral movement. Although tie-rods are per tensioned, to their correct value during assembly, they should be checked at intervals. This is accomplished by:

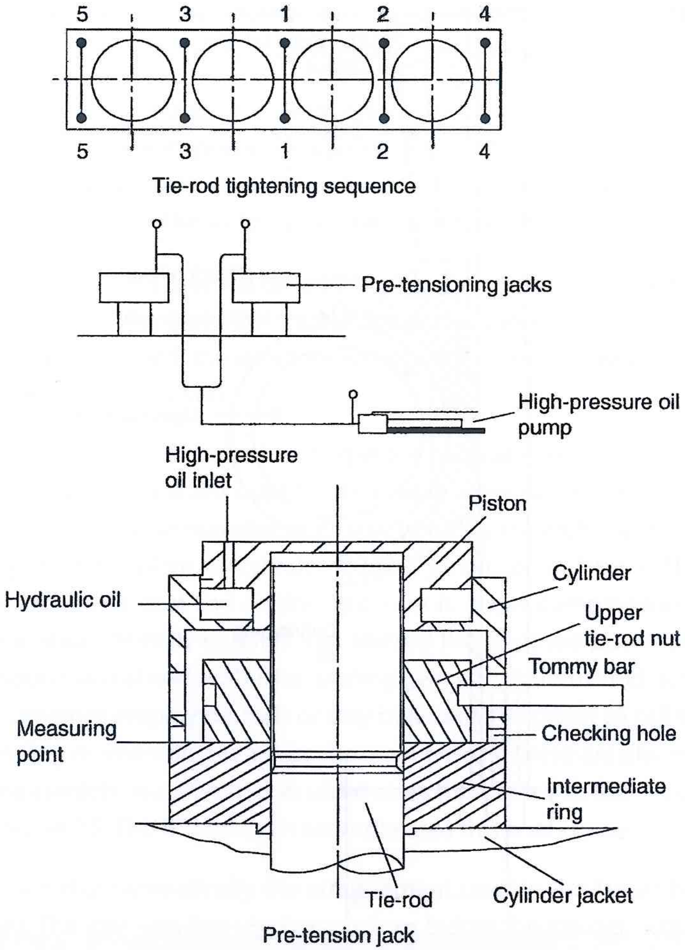
- Connecting both pre-tensioning jacks to two tie-rods lying opposite each other



▲ **Figure 2.5** *Use of main bearing 'jack' bolts*

- Checking the clearance between the nut and intermediate ring with a feeler gauge. If any clearance exists then the nut is tightened onto the intermediate ring and the pressure is released. If no clearance is found then the pressure can be released and the hydraulic jacks can be removed.
- Figure 2.7 shows the two different arrangements on the MAN MC engines.

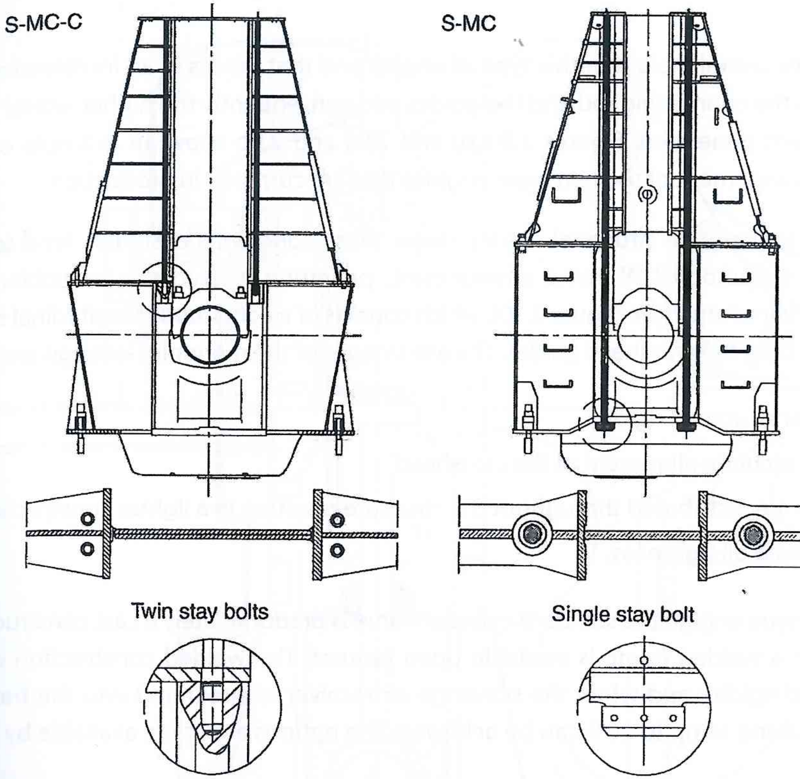
When hydraulic tensioning equipment is specified for use on the main engine bolts it is essential that all the equipment is maintained in good order and the accuracy of the



▲ **Figure 2.6** *Tie-rod tightening sequence and pre-tensioning jack*

from the rod as the nuts are round and therefore spanners will not fit. This is also true for the cylinder cover bolts.

If when inspecting the engine it is found that a tie-rod has broken, it must be immediately replaced. If the breakage that occurs is such that the lower portion is short and can be removed through the crankcase, the upper part can be withdrawn with relative ease from the top. If, however, the breakage leaves a long lower portion



▲ **Figure 2.7** *Arrangement of the tie-rods on the MAN S-MC series of engines*

the latest engines. The design of the latest MAN engines is to incorporate a shorter 'twin' tie or stay bold as MAN prefer to call them. This arrangement sees the tie bolt stop short of penetrating through the bedplate (figure 2.7). Students undertaking the flag state exams must have a good knowledge of both these systems and the reasons for their construction as the student may be required to serve on a vessel with these engines fitted following qualification.

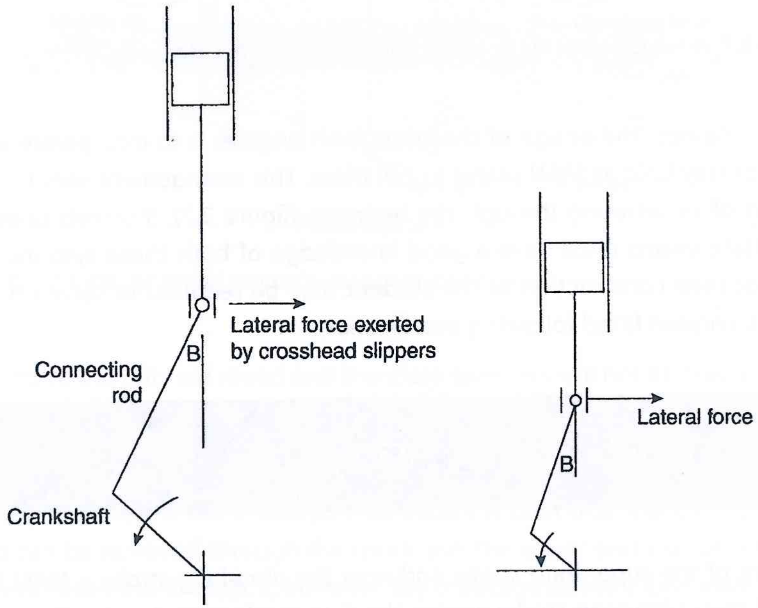
'A' Frames or Columns

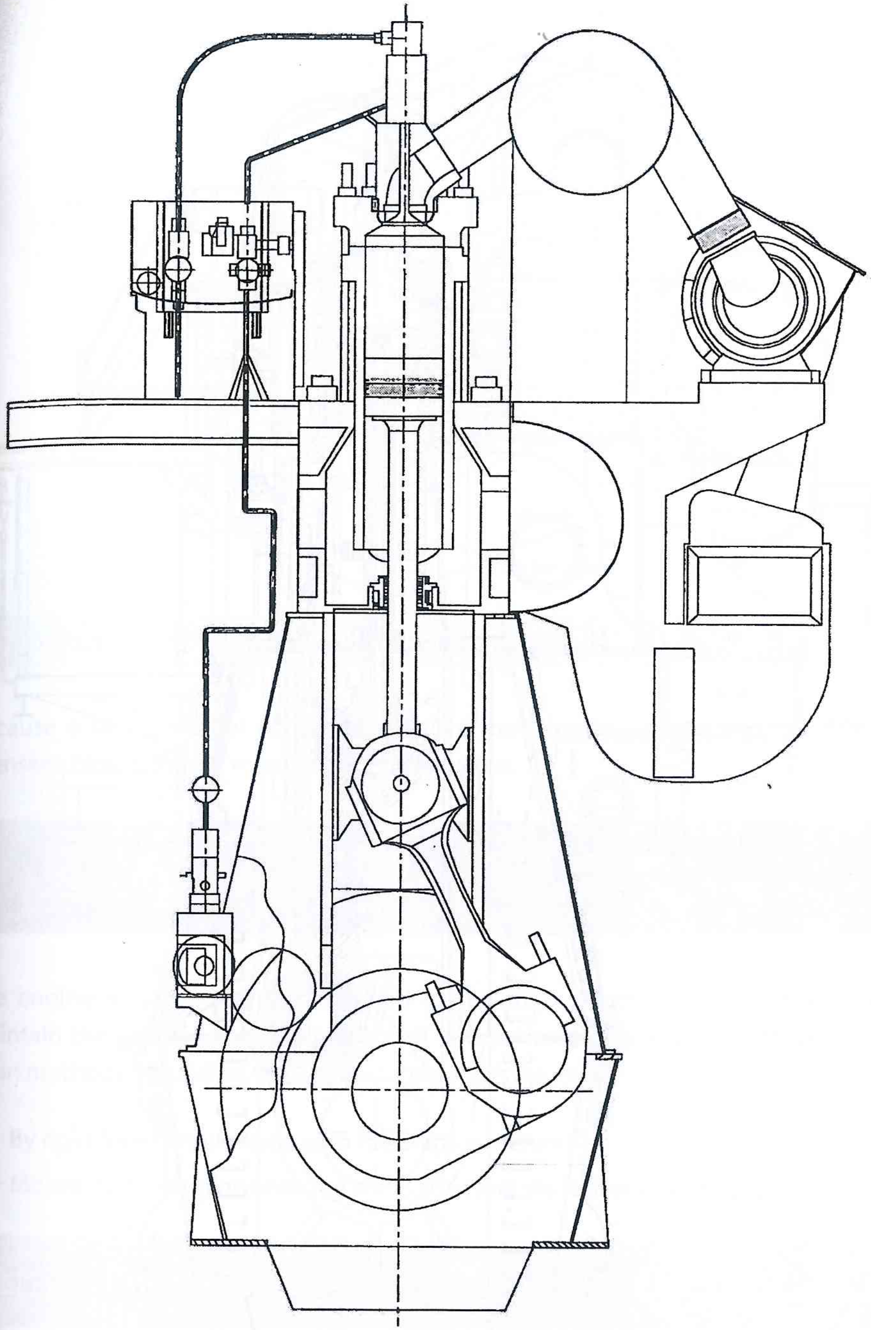
The advent of the super-long stroke and now the ultra-long stroke – MAN Diesel & Turbo call their ultra-long stroke engine the G-type where G denotes Green – slow-reduce the overall height of this type of engine and that results in an increased angle between the connecting rod and the guides and consequently the higher lateral force component generated. Figures 2.8(a,b) and 2.9a and 2.9b show an example of the actual arrangement of the two main engines that are currently in production.

In order to maintain structural rigidity under these conditions designers tend to not utilise the traditional 'A' frame arrangement, preferring instead the 'monoblock' or 'cylinder frame' structure (figure 2.10), which consists of a continuous longitudinal beam incorporating the crosshead guides. The advantages of the monoblock design are:

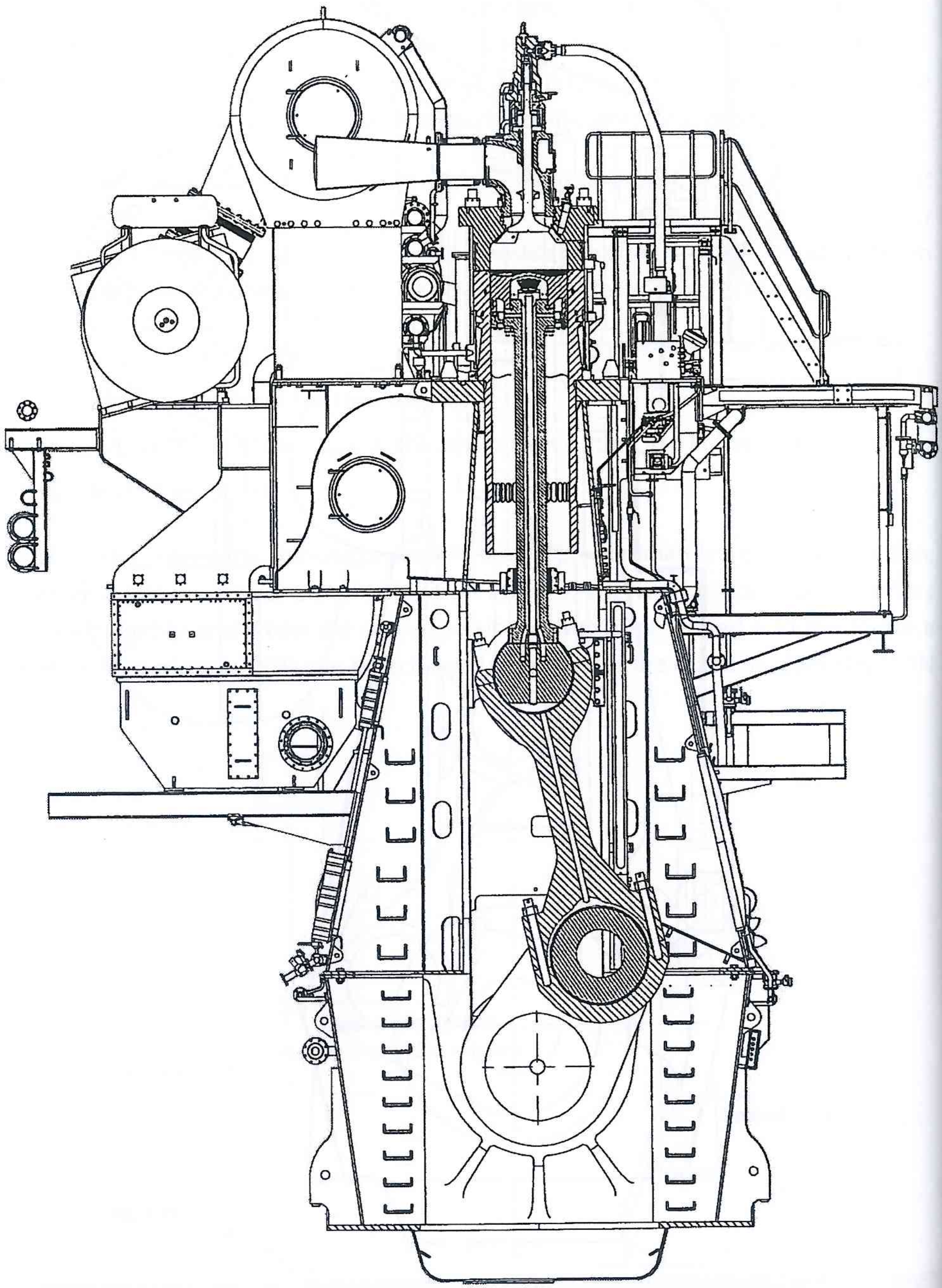
- greater structural rigidity
- more accurate alignment of the crosshead
- forces are distributed throughout the structure resulting in a lighter construction
- improved oil tightness.

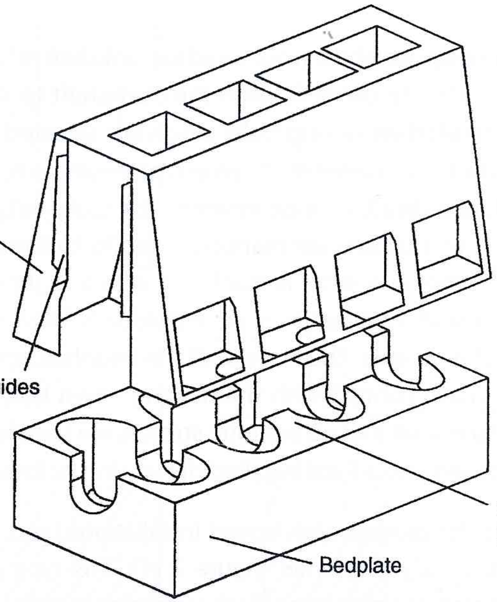
In the G-type engine the standard cylinder frame is predominately a cast construction, however, a welded frame is available upon request. The welded construction offers increased rigidity and when the scavenge air receiver is integrated into the frame a weight saving of up to 30% can be achieved. The options are made available by MAN





▲ Figure 2.9a *Wärtsilä RTA engine cross-section*





▲ **Figure 2.10** *Modern monobox construction*

because a large number of the engines are made under licence and the different licensees have different manufacturing capacities.

Holding Down Arrangements

The engine must be securely attached to the ship's structure in such a way as to maintain the alignment of the crankshaft within the engine structure. There are two main methods of holding the engine to the ship's structure.

1. By rigid foundations fixed onto the ship's structure.
2. Mounting the engine onto the ship's structure via resilient mountings.

Rigid foundations

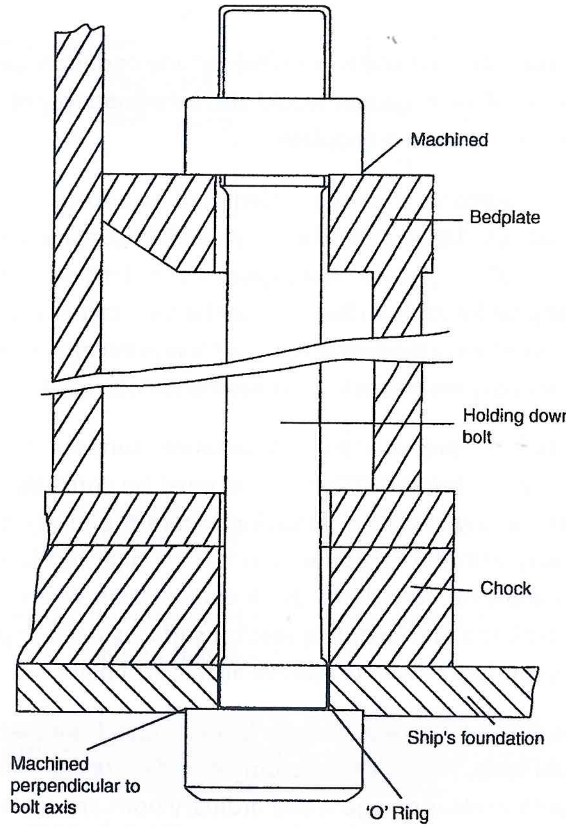
In this more traditional method, fitted chocks are installed between the engine bedplate no distortion of the bedplate which would lead to crankshaft misalignment. In addition, great care must be taken to correctly align the crankshaft to the propeller shaft. The engine is initially installed on jacking bolts which are adjusted to establish its correct location in relation to the propeller shaft. When the engine is correctly positioned, and crankshaft deflections indicate no misalignment, the space between the bedplate and seating is measured and chocks are manufactured. To facilitate the fitting of chocks the top plate of the engine seating is machined with a slight outboard-facing taper. The chocks, usually made from cast iron, are individually fitted and must bear the load over at least 85% of their area. The surface of the bedplate and the underside of the top plate that will make contact with the holding down bolt and the nut faces are machined parallel to ensure that no bending stresses are transferred to the bolt. As the holding down bolts and chocks are installed the jacking bolts are removed.

Holding down bolts for modern slow-speed installations tend to be the long sleeved type and are hydraulically tensioned (figure 2.11). This type of bolt, because of its greater length, has greater elasticity and is therefore less prone to fatigue cracking than the superceded short un-sleeved bolt. The bolts are installed through the top plate and a waterproof seal is usually effected with 'O' rings. Fitted bolts are installed adjacent to the engine thrust.

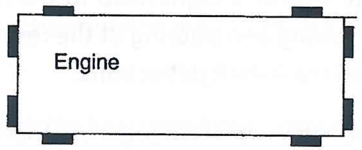
The holding down bolts should only withstand tensile stresses and should not be subjected to shear stresses. The lateral and transverse location is maintained by side and end chocking. The number of side chocks will depend upon the length of the engine (figure 2.12a,b).

It is extremely important that the engine is properly installed during building. The consequences of poor initial installation is extremely serious since it may lead to the fretting of chocks, the foundation and/or the bedplate, resulting in a slackening and breakage of holding down bolts and ultimately in misalignment of the engine. To maintain engine alignment it is important to inspect the bolts for correct tension and the chocks for evidence of fretting and looseness. There are a number of factors that can cause a holding down bolt to look as if it is tight when in fact it has become stretched. Corrosion or the nut binding on the threads may mean that the nut is held fast appearing to be firmly in place.

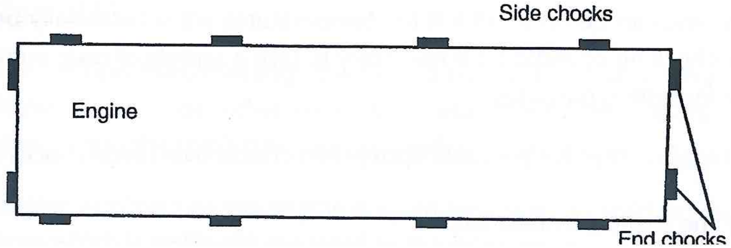
One method of testing the holding down bolts is to place your thumb or finger on one side of the nut, so that you can feel any movement between the nut and the surface of the bedplate that it is supposed to be firmly against. If you then carefully tap the



▲ **Figure 2.11** *Long-sleeved holding down bolt*



(a) Short engine: with two sets of side chocks



elongated beyond its elastic limit and is therefore outside its design specification. If you can reach, then the use of feeler gauges would also be a good way of checking for any clearance between the nut and the bedplate.

An alternative to the traditional chocking materials of cast iron or steel is epoxy resin. This material, originally used as an adhesive and protective coating, was developed as a repair technique to enable engines to be realigned without the need for the machining of engine seating and bedplate. It is claimed that the time taken to accomplish such a repair is reduced thus reducing the overall costs. Although initially developed as a repair technique the use of epoxy resin chocks has become widespread for new buildings.

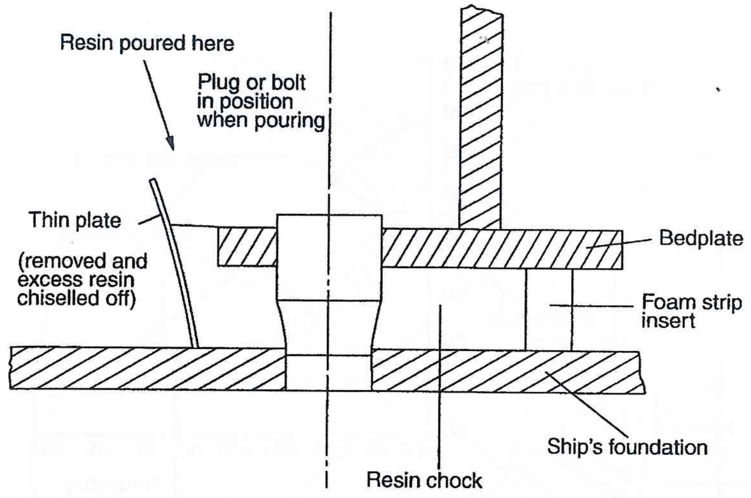
Resin chocks do not require machined foundation surfaces thus reducing the preparation time during fabrication. The engine must be correctly aligned with the propeller shaft without any bedplate distortion before the resin application. This is done in the usual way with the exception that it is set high by about 1/1,000 of the chock thickness to allow for very slight chock compression when the installation is bolted down. The tank top and bedplate seating surfaces must then be thoroughly cleaned with an appropriate solvent to remove all traces of paint, scale and oil.

As resin chocks are poured it is necessary for 'dams', made from foam strip, to be set to contain the liquid resin. Plugs or the holding down bolts are now inserted. Fitted bolts are sprayed with a releasing agent and ordinary bolts are coated with a silicone grease to prevent the resin from adhering to the metal. The outer sides of the chocks are now blocked off with thin section plate, fashioned as a funnel to facilitate pouring and 15 mm higher than the bedplate to give a slight head to the resin. This is also coated to prevent adhesion. Prior to mixing and pouring of the resin it is prudent to again check the engine alignment and crankshaft deflections.

The resin and activator are mixed thoroughly with the equipment that does not entrain air. The resin is poured directly into the dammed off sections. Curing will take place in about 18 h if the temperature of the chocking area is maintained at about 20–25°C. The curing time can be up to 48 h if the temperatures are substantially below this. During the chocking operation it is necessary to take a sample of resin material from each batch for testing purposes.

The advantages claimed for 'pourable' epoxy resin chocks over metal chocks include:

- Quicker and cheaper installation.



▲ **Figure 2.13** *Poured resin chocks*

steel the thrust forces are distributed to all chocks and bolts thus reducing the total stress on fitted bolts by about half (figure 2.13).

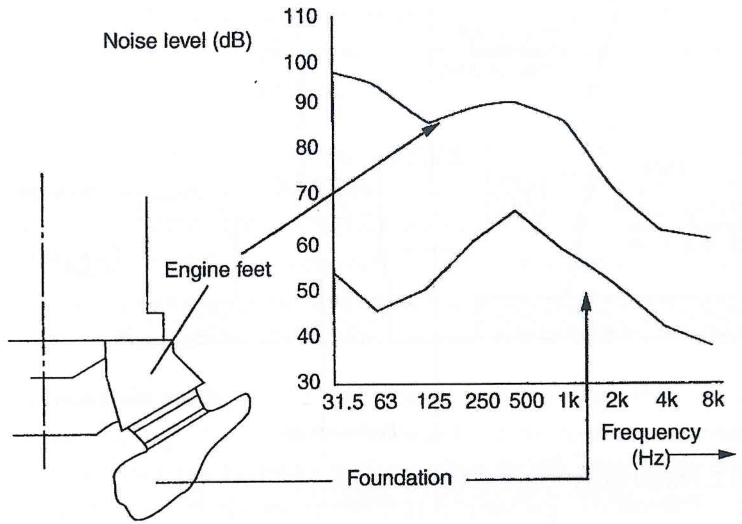
Resilient mountings

A possible disadvantage of rigidly mounted engines is the likelihood of noise being transmitted through the ship's structure. This is particularly undesirable on a passenger carrying vessel where low noise and vibration levels are necessary for passenger comfort and as a consequence manufacturers are now installing diesel engines on resilient mountings.

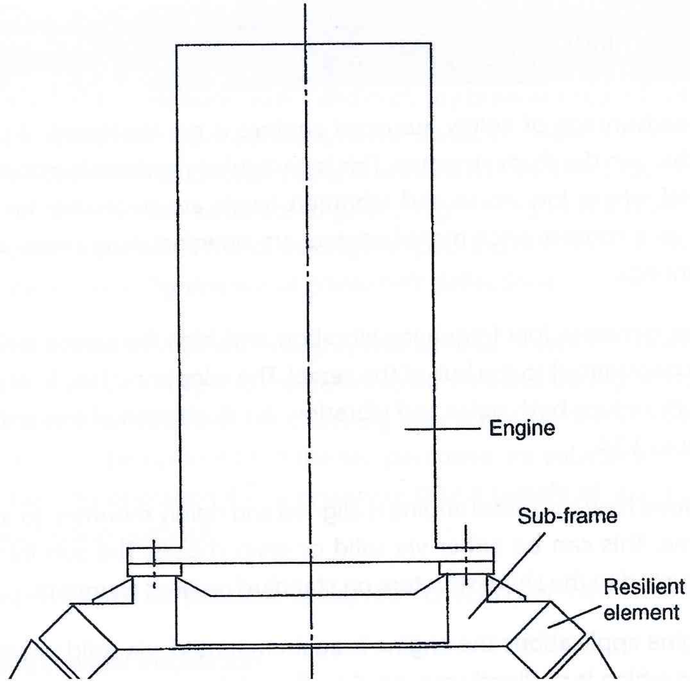
Diesel engines generate low frequency vibration and high frequency noise both of which can be transmitted to the hull of the vessel. The adoption of resilient mountings will successfully reduce both noise and vibration. An illustration of this reduction can be seen in figure 2.14.

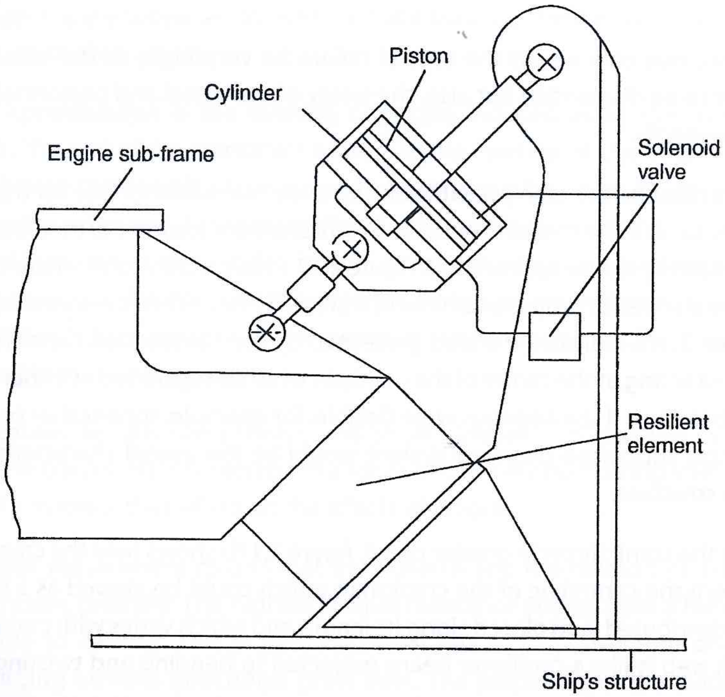
Figure 2.15 shows how the diesel engine is aligned and rigidly mounted to a fabricated steel sub-frame. This can be either via solid or resin chocks. The sub-frame is then resiliently mounted to the ship's structure on standard resilient elements.

In geared engine applications the engine is again mounted, via solid or resin chocks,



▲ **Figure 2.14** *Reduction in structure-borne noise achieved by resilient mountings*





▲ **Figure 2.16** *Hydraulic locking device for engine movement limitation during starting/stopping*

the ship's structure and this is accomplished by stopper devices built into the holding down arrangement.

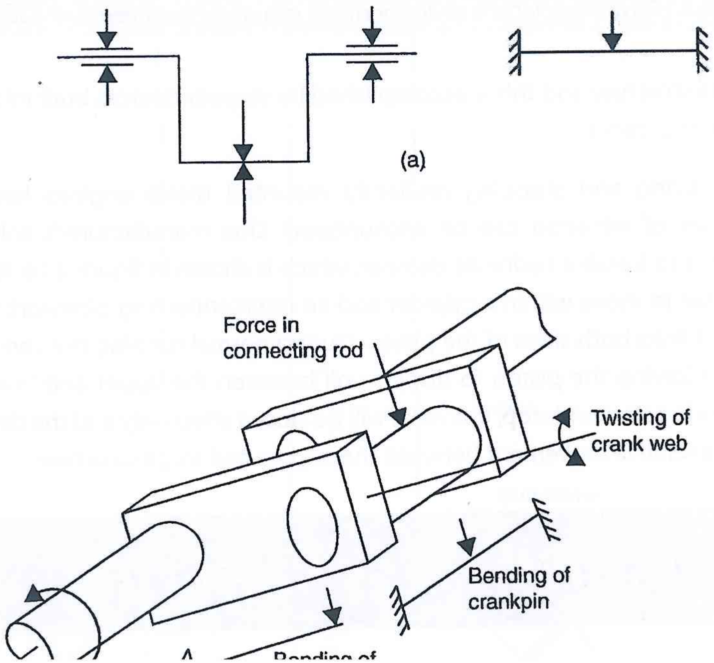
When starting and stopping resiliently mounted diesel engines large transitory amplitudes of vibration can be encountered. One manufacturer's solution to this problem is to install a hydraulic damper, which is shown in figure 2.16. It has a piston that is able to move within a cylinder and an interconnecting pipework via a shut off valve that links both sides of the piston. During normal running the connecting valve is open allowing the piston to displace oil between the upper and lower chambers freely. During start and stop, this valve will be closed effectively and the device will then prevent a relative movement between the engine and ship's structure.

Crankshafts

reliable since not only would the cost of failure be very high, as the whole engine would have to be dismantled but also, the safety of the vessel and personnel could be placed in jeopardy.

Crankshafts must be extremely reliable and virtually maintenance free for the effective life of the engine. If the stresses set up in a crankshaft are examined, then the need for extreme reliability will be appreciated. Figure 2.17 shows a crank unit where it could be regarded as a series of simple beams which are then subject to a number of different loads. Figure 2.17a indicates the load generated by the combustion pressure. This is a variable load acting at the centre of the crankpin which is supported at either end by its two main bearings. If the bearings were flexible, for example, spherical or ball shaped, then a simply supported beam equivalent would be the overall characteristic of this part of the structure.

Examining the crank throw in greater detail, figure 2.17b shows how the crankpin itself extends from the centreline of the crankshaft which could be viewed as a beam with an evenly distributed load placed along its length and which varies with crank position. Each crank web is like a cantilever beam subjected to bending and twisting. Journals

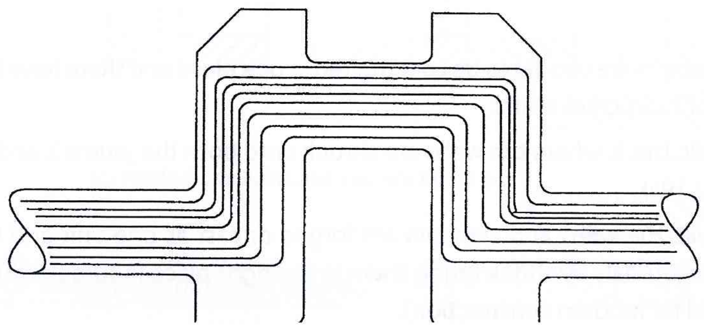


would be principally subjected to twisting, but a bending stress must also be present if we refer back to diagram (a).

A further consideration is the twisting or torque induced alone the length of the crankshaft. The end of the crankshaft nearest to the flywheel at the output end of the engine will have to transmit all the torque induced along the entire length of the shaft. Whereas at the other end, the crankshaft will only have to transmit the torque induced by one and then two of the cylinders in turn. A design feature of the G-type MAN Diesel & Turbo is that considerable weight saving has been realized by making the cylinders closer together at the forward end of the engine. This in turn means that the crankshaft can be shorter thus saving weight.

Bending causes tensile, compressive and shear stresses and twisting causes shear stress. As the crankshaft is subjected to a series of complex fluctuating stresses it must be built of a material that will resist the effects of fatigue.

This complex requirement means that the material and the method of manufacture must be chosen carefully. The highest fatigue resistance comes from a forging which is preferable to casting. This is because metal has a grain structure and unlike a casting, forging exhibits directional 'grain flow'. The properties of the material in the direction transverse to the grain flow are significantly inferior to those in the direction longitudinal to the grain flow. Under these circumstances the drop in fatigue strength may be as much as 25–35% with similar reductions in strength and ductility. Forging methods, therefore, ensure that the principal direction of grain flow is parallel to the major direct stresses imposed on the crankshaft (figure 2.18). Smaller crankshafts are drop forged from one piece of metal which means that the grain structure runs all along the length of the shaft. Larger crankshafts are just too big to be manufactured in this way.



The materials chosen for forged and cast crankshafts are essentially the same. The composition of the steel will vary depending upon the bearing type chosen. For a crankshaft with white metal bearings a steel of 0.2% carbon may be chosen; this will have a ultimate tensile strength (UTS) of approximately 425–435 MN/m^2. For higher output applications with harder bearing materials the carbon content is in the range of 0.35– 0.4% which raises the UTS to approximately 700 MN/m^2. To increase the hardness of the shaft still further alloying agents such as chromium-molybdenum and nickel are added. For smaller engines such as automotive applications the crankshafts are surface hardened and fatigue resistance is increased by nitriding.

There are two broad categories of crankshafts:

1. One-piece construction.
2. 'Built up' from component parts.

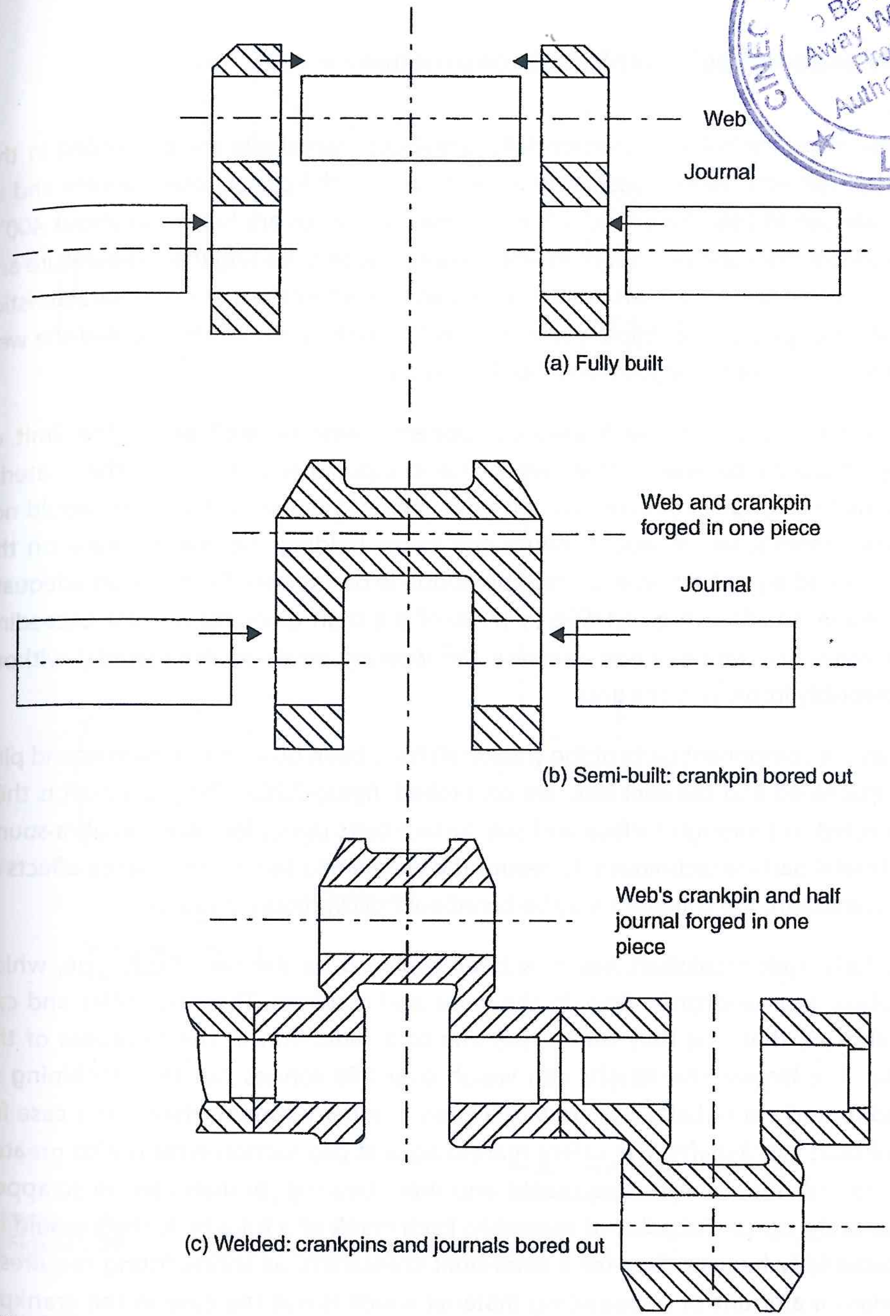
One-piece construction

One-piece construction, either cast or forged, is usually restricted to smaller medium- and high-speed engines. Following the casting or forging operation the component is rough machined to its approximate final dimensions and the oil passages are drilled. The fillet radius and crankpin are then cold rolled to improve the fatigue resistance and to reduce the micro-defects on the surface. Following machining the crankshaft is then tested for surfaced and sub-surfaced defects by using a combination of methods from dye-penetrant testing for surface defects to non-destructive testing for any defects that could be deeper within the structure.

'Built' crankshafts

These crankshafts are too big to be constructed in one piece and there have been three categories of 'built' crankshafts:

1. *Fully built*: this is where the webs are shrunk onto both the journals and crankpins (figure 2.19a).
2. *Semi-built*: the webs and crankpin are forged or cast as one unit and then fitted onto the journals by shrink fitting them in the right place (figure 2.19b) (the most favoured for modern construction).



▲ **Figure 2.19** *Three types of crankshaft assembly*

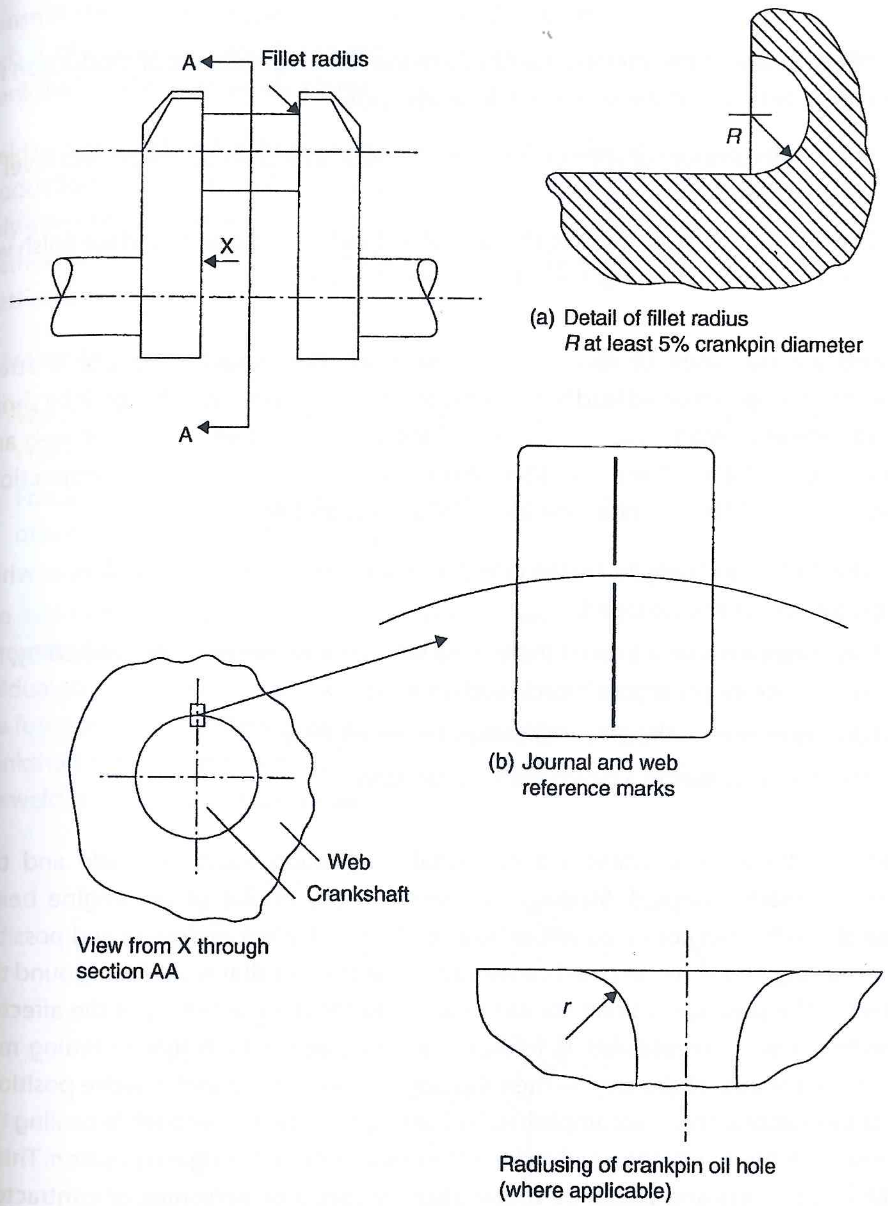
Fully and semi-built crankshaft construction

To minimise the risk of distortion fully semi-built crankshafts are assembled in the vertical position. Various jigs are required to ensure the correct crank angles and to provide support for the crankshaft as it grows. The webs are heated to about 400°C and the journals and pins are then inserted into position. Raising the temperature any higher would bring the steel to a critical temperature where the material's characteristics would change and the shape would deform. When the assembly has cooled the web material adjacent to the journal will be in tension.

The level of stress in the heated components must be well below the limit of proportionality to ensure that, when the components are cooled, the material returns to its original dimensions. If it went past its yield point the metal would not shrink properly, which would reduce the forces holding the web in place on the journal leading to fretting and probable slippage of the web. To ensure an adequate shrinkage, an allowance of 1/550 to 1/700 of the shaft diameter is usual. Exceeding this allowance would simply increase the working stress in the material without appreciably improving the grip.

When the component parts of the crankshaft have been built up the journals and pins are machined and the fillet radii are cold rolled (figure 2.20a). The crankshaft is then subjected to thorough surface and sub-surface tests using, for example, ultra-sound and metal particle techniques. To reduce the weight and the out-of-balance effects of the crankshaft, the crankpins may be bored out hollow (figure 2.19a–c).

The fully built crankshaft has now been replaced by the semi-built type, which displays improved 'grain flow' in the webs and crankpin. They are stiffer and can be shorter, than the fully built type, due to a reduction in the thickness of the webs. The largest crankshafts can weigh over 200 tonnes but the machining of solid forged crankshafts require larger production equipment than is the case for semi-built crankshafts. The safety margin against production error is also greater, as any individually defective cranks and main bearing journals can be scrapped separately, up to the point of assembly. Each crank of a fully built shaft would be substantially heavier than for a semi-built crankshaft, as shrink fitting requires a minimum amount of surrounding material which is not the case in the crankpin of the semi-built shaft. Fully built crankshafts were common in the past, but today there are no problems in casting, forging and machining the larger sections of the



▲ **Figure 2.20** *Crankshaft construction details*

The effectiveness of the interference fit due to the shrinkage process of production for both the fully built and the semi-built depends upon:

1. The correct amount of shrinkage which will result in setting up the correct level of stress in the web and journal.
2. The quality of surface finish of the journal and web. Good quality surface finish will give the maximum contact area between web and journal.

Dowels are not used to locate the shrink since this would introduce a stress concentration which could lead to fatigue cracking. When a crankshaft is built by shrink fitting, reference marks are made to show the correct relative position of web and journal (figure 2.20b). These marks should be inspected during crankcase inspections as slippage could occur under one of the following conditions:

- If the starting air is applied to the cylinders when they contain water or fuel, or when the turning gear is engaged.
- If an attempt is made to start the engine when the propeller is constrained by, for example, ice or any other obstacle such as a log.
- If during operation the propeller strikes a submerged object.
- If the engine comes to a rapid unscheduled stop.

Following these circumstances a crankshaft inspection must be made and the reference marks checked. Slippage will result in the timing of the engine being altered which if not corrected will at best result in inefficient operation and possible poor starting and at worse could cause additional stresses that would compound the defect. If the slippage is small, for example, up to 15° then re-timing of the affected cylinders may be considered. If, however, the slippage is such that re-timing may affect the balance of the engine then the original journal and web relative positions must be restored. This is accomplished by heating the affected web while cooling the journal with liquid nitrogen and jacking the crankshaft to its original position. This is a difficult process and would be undertaken by specialist personnel or contractors under controlled conditions.

Welded construction

The development of the large marine crosshead two-stroke engine has resulted in ever

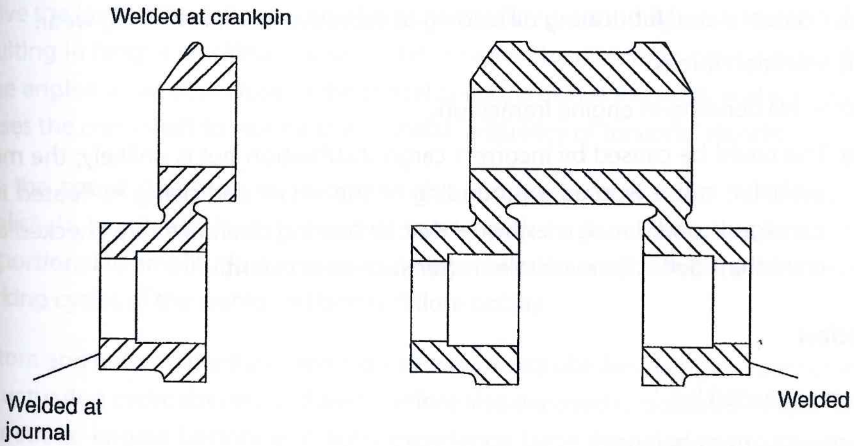
transmit the torques required the traditional shrink fitting method requires that the web is of a minimum width and radial thickness. This will inevitably lead to a larger crankshaft and consequently a larger engine.

Welded construction has been a possible solution in the future, however the cost of production and the difficulties of quality assurance (weld quality) mean that to date only a few of these have been built. These few have in fact given good service and this method of building a large crankshaft may return if there can be found a cost-effective method of production.

There are two methods of assembly:

1. Welding two crank-arms together and then making a crankshaft by welding the crank-arms together (figure 2.21a).
2. Forging a crank-throw complete with half journals and then welding them with others to form the crankshaft (figure 2.21b).

The welding technique chosen is submerged arc narrow gap. This technique is automated and produces a relatively small heat-affected zone, which produces minimal residual stresses and distortion. At the completion of welding the crankshaft is heated in a furnace to 580°C followed by a slow cooling period. Following heat treatment the crankshaft is tested using ultrasonic and metal particle techniques. If flaws are found the weld is machined out and re-welded.



(a) Half crank welded at journal and crankpin

(b) Crankthrow forged complete, welded at journal

The advantages claimed for welded crankshafts are:

1. Reduced principle dimensions of the engine.
2. Reduced web thickness results in a considerable reduction in weight.
3. Reduced web thickness allows journal lengths to be increased resulting in lower specific bearing loads.
4. Freedom to choose large bearing diameters without overlap restrictions.
5. Increased stiffness of crankshaft resulting in higher natural frequencies of torsional vibration.

Crankshaft defects and their causes

Misalignment

If we assume that alignment was correct at initial assembly then the possible reasons for misalignment are as follows:

1. Worn main bearings.
   (a) Caused by incorrect bearing adjustment, leading to overloading.
   (b) Broken, badly connected or choked lubricating oil supply pipes, causing lubrication starvation, leading to excessive or adverse bearing wear.
   (c) Contaminated lubricating oil leading to excessive adverse bearing wear.
   (d) Vibration forces.
2. Excessive bending of engine framework.
   (a) This could be caused by incorrect cargo distribution but is unlikely; the more probable cause would be grounding of the vessel and being re-floated in a damaged condition. It is essential that all bearing clearances are checked and crankshaft deflections are taken after such an accident.

Vibration

This can be caused by:

1. incorrect power balance
2. prolonged running at or near critical speeds

5. near presence of running machinery
6. excessive wear down of the propeller shaft bearing (this can lead to whipping of the shafting in bad weather conditions)
7. vibration-accentuated stresses, which can be increased to exceed fatigue limits and considerable damage could result. It can lead to fixings and fastenings working loose, for example, coupling bolts, bearing bolts, bolts securing balance masses to crank webs and lubricating oil pipes.

Other causes

On modern vessels with the aid of CAD and other mathematical modelling tools incorrect manufacture leading to defects is a rare occurrence but not unheard of. In the past, failure has been caused by:

- slag inclusions
- poor control of the heat treatment and machining processes, for example, badly radiised oil holes and fillets and
- careless use of tools resulting in impact marks on crankpins and journals can also lead to failure.

These defects all result in the creation of stress concentrations which, because of the cyclic loading of the crankshaft can raise the local level of stress in a given component to above the level of the fatigue limit, shown on the *S/N* graph, in figure 1.16 in Chapter 1, resulting in fatigue cracking and ultimately failure. The condition can be exacerbated if the engine is run at or close to the critical speed, which is the rotational speed which causes the crankshaft to vibrate at its natural frequency of torsional vibration.

It is the speed that induces resonance the consequence of which is to cause the crankshaft to vibrate in the torsional mode with large amplitudes. Stress, being proportional to amplitude, increases and may rise sufficiently to reduce the number of working cycles of the crankshaft before failure occurs.

Bottom end bolts on medium- and high-speed four-stroke diesel engines are subjected to fluctuating cyclic stresses and are therefore also exposed to potential fatigue failure. Four-stroke engine bottom end bolts experience large fluctuations of stress during the cycle. This is due to the inertia forces experienced in reversing the direction of the piston over TDC on the exhaust stroke. The forces experienced by bottom end bolts in maximum serviceability, stresses should be commensurate with a level below the fatigue limit. Since:

$$\text{stress} = \frac{\text{load}}{\text{area}}$$

it can be seen that for a given load the stress can only be reduced by increasing the area and therefore increasing the size and weight of the bottom end bolt. Designers opt for a compromise, they design a bolt that will experience a level of stress ABOVE that of the fatigue limit and specify the number of cycles the bolt should remain in service before it is replaced. It is therefore of vital importance that the running hours of four-stroke engines are known in order to monitor the safe working life of bottom end bolts.

In addition to this, designers will specify that bottom end bolts:

- are manufactured to high standards of surface finish
- have rolled threads
- be of the 'wasted' design with generous radii
- have increased diameter at mid shank to reduce vibration
- be tightened accurately to the required level.

As part of the maintenance programme bolts should be examined for mechanical damage which would cause stress concentration and damaged bolts should be replaced.

Fretting corrosion

This occurs where two surfaces forming part of a machine, which in theory constitute a single unit, undergo slight oscillatory motion of a microscopic nature.

It is believed that the small relative motion causes removal of metal and protective oxide film. The removed metal combines with oxygen to form a metal oxide powder that may be harder than the metal (certainly in the case of ferrous metals) thus increasing the wear. Removed oxide film would be repeatedly replaced, increasing further the amount of damage being done.

Fretting damage increased with load, amplitude of movement and frequency. Hardness

Oxygen availability also contributes to the attack, if oxygen level is low the metal oxides formed may be softer than the parent metal thus minimising the damage. Moisture tends to decrease the attack.

Bearing corrosion

In the event of fuel oil and lubricating oil combining in the crankcase, weak acids may be released which can lead to corrosion of copper lead bearings. The lead is removed from the bearing surface so that the shaft runs on nearly pure copper, which raises bearing temperature causing the lead rises to the surface when it is removed. The process is repeated until failure of the bearing takes place. Scoring of crankshaft pins can then occur. Use of detergent lubricating oil can prevent or minimise this type of corrosion because the detergency properties of the oil hold the small particles in suspension and the alkalinity of the oil will neutralise the acids produced.

Water in the lubricating oil can lead to white metal attack and the formation of a very hard black incrustation of tin oxide. This oxide may cause damage to the journal or crankpin surface by grinding action. Water also combines with any sulphur to form sulphuric acid which creates a further need for the oil to neutralise the acids.

Bearing clearances and shaft misalignment

The condition of the bearings and their correct position is very important to give feedback to the engineer about the condition of the engine. The new methods of Condition-Based Monitoring/Maintenance (CBM) utilise the monitoring of bearing clearance to give real-time data about the state of the engine. In the past bearing clearances were checked in a variety of ways:

- A rough check is to observe the discharge of oil, in the warm condition, from the ends of the bearings.
- Feeler gauges can be used, but for some of the bearings they can be difficult to manoeuvre into position in order to obtain readings.
- Clock (or as they are sometimes called, dial) gauges can be very effective and accurate providing the necessary relative movement of the crankshaft webs, can be achieved, this can prove to be difficult in short engines that have a stiff crankshart.
- Finally, the use of lead wire. This required the bearing keeps to be removed, the lead

equal to the bearing clearance. It was then just a matter of removing the keeps and measuring the lead wire thickness with a micrometer.

Engine manufacturers have been keeping records of the results produced by the techniques described above. MAN Diesel & Turbo found that as much as 7,000 ships each year have their engine's bearings viewed as part of 'open up' inspections but only 1% of defects are found during those inspections. Not only that but also as part of the growing trend of 'maintenance-induced failures' over 2% of bearing defects are caused by the inspection themselves. Proximity sensors have now been developed that will measure the bearing wear on a two-stroke main engine in real time. This has led to the recommendation not to open up main engine bearings if this monitoring equipment is fitted.

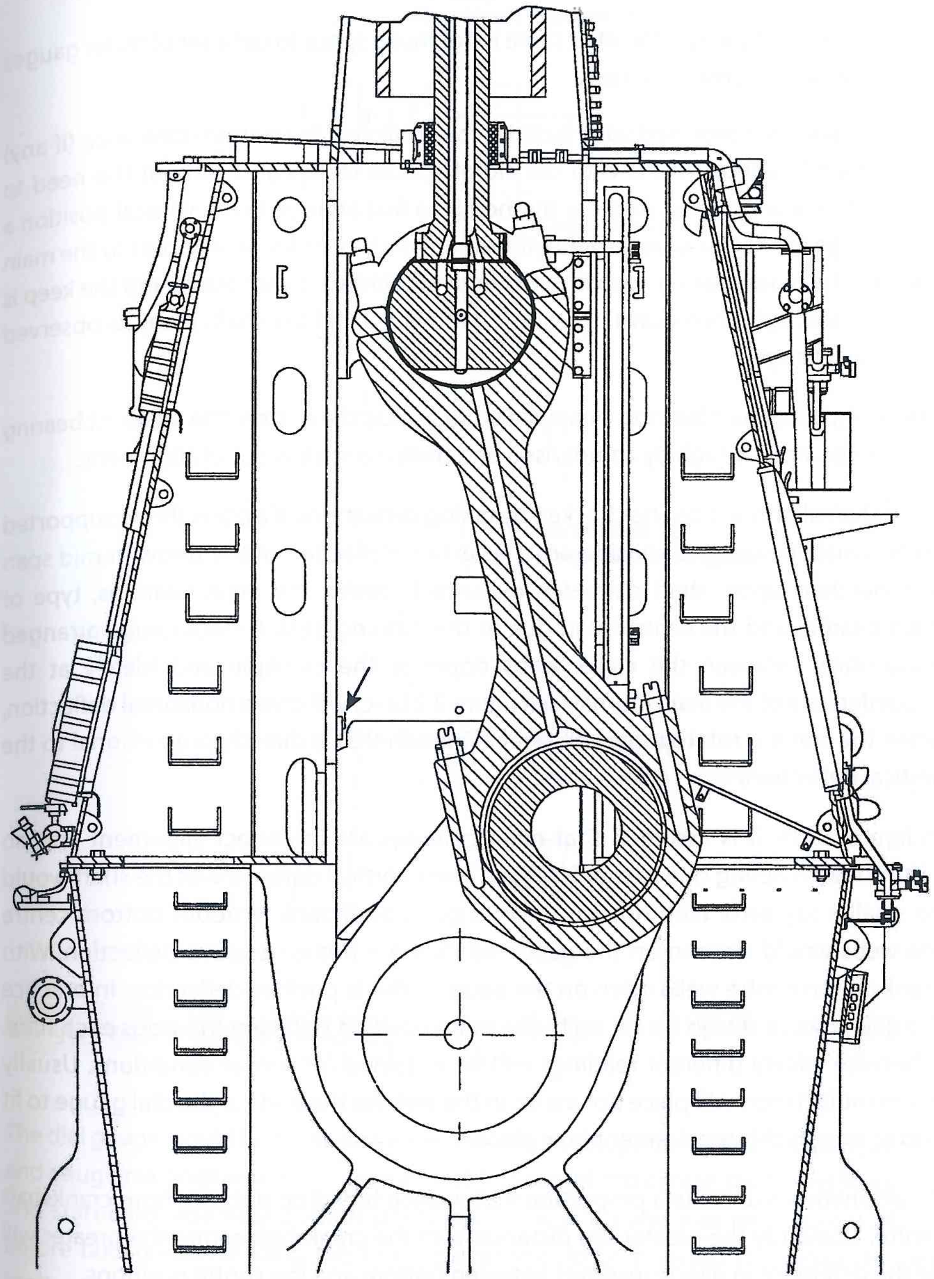
The equipment indicates wear in the crank train: main, crankpin and crosshead bearings. The temperature of the main bearing can also be monitored as can the extent of any water in the lubricating oil and the electrical potential between the propeller shaft and hull, all of which may have an adverse effect on bearing life.

The system is made up of analogue inductive sensors fitted to each cylinder and located on a bracket which is fixed to the engine frame so that the field effect from the sensor is interfered with each time the bottom of the crosshead guide reaches the bottom of its travel. There are several technologies available to achieve this but the effect on the generated field differs according to the physical position of the base of the crosshead guide. This position obviously takes into account the position of all the components in the crank train including the main, crankpin and crosshead bearings.

The proximity sensor interprets that results from the disturbance within the sensor field and transmits an electrical signal proportional to the disturbance. A signal processing unit (SPU) mounted outside the engine processes the signal and sends then to the interface unit mounted in the engine control room which forms the connection to the main Amot Monitoring System (AMS) and also allows local system access to the engineers in the control room (figure 2.22).

Further link can be made using an Ethernet connection via a PC on the ship's network. The calibrated SPU communicates wear data to the human–machine interface (HMI) which provides a clear graphic display of bearing wear.

If the crankshaft is aligned correctly and is straight in the engine then the main bearing



▲ **Figure 2.22** *Position of bearing wear sensor (shown by the arrow)* this value is usually very difficult as there is not much space to get a set of feeler gauges in place to take the measurement.

Some engines are provided with facilities for obtaining the bottom clearance (if any) of the main bearings. This is with the aid of special feelers and without the need to remove the bearing keep. Another method is to first arrange in the vertical position a clock gauge so that it can record the movement of the crank web adjacent to the main bearing. The main bearing keep is then removed, shims are withdrawn and the keep is replaced and tightened down. The vertical movement of the shaft, if any, is observed on the dial gauge.

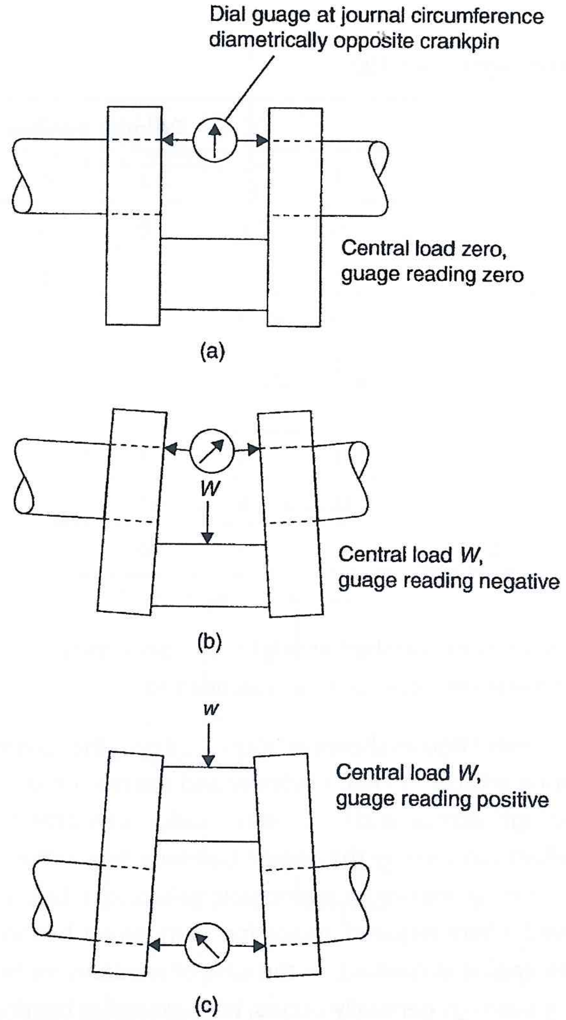
Obviously, if the main bearing clearance is not zero at the bottom the adjacent bearing or bearings are also high by comparison and then the shaft is out of alignment.

Crankshaft alignment can be checked by taking deflections. If a crank throw supported on two main bearings is considered, the vertical deflection of the throw in mid span is dependent upon: shaft diameter, distance between the main bearings, type of main bearing and the central load due to the running gear. A clock gauge arranged horizontally between the crank webs opposite the crankpin and ideally at the circumference of the main journal (see figure 2.23a–c) will give a horizontal deflection, when the crank is rotated through one revolution that is directly proportional to the vertical deflection.

In figure 2.23a, it is assumed that main bearings are in correct alignment and no central load is acting due to running gear. Then, vertical deflection of the shaft would be small – say zero. With running gear in place and crank at about bottom centre the webs would close in on the gauge as shown – this is negative deflection. With crank on top centre webs open on the gauge – this is positive deflection. In practice the gauge must always be set up in the same position between the webs each time, otherwise widely different readings will be obtained for similar conditions. Usually the manufacturers will place dot marks in the web for the ends of the dial gauge to fit into to ensure this requirement take place.

An alternative is to make a proportional allowance based on distance from crankshaft centre. Obviously the greater the distance from the crankshaft centre the greater will be the difference in gauge readings between bottom and top centre positions.

Since, due to the connecting rod, it is generally not possible to have the gauge



▲ **Figure 2.23** *Checking crankshaft alignment*

The dial gauge would be set at zero when crank is in, say, port side near bottom position and gauge readings would be taken at port horizontal, top centre, starboard horizontal and starboard side near bottom positions. Say *x*, *p*, *t*, *s* and *y* as per figure 2.24, but before taking each reading the turning gear should be reversed to unload the gear teeth, otherwise misleading readings may be obtained.

Any engines still with spherical main bearings will have greater allowances for crankshaft

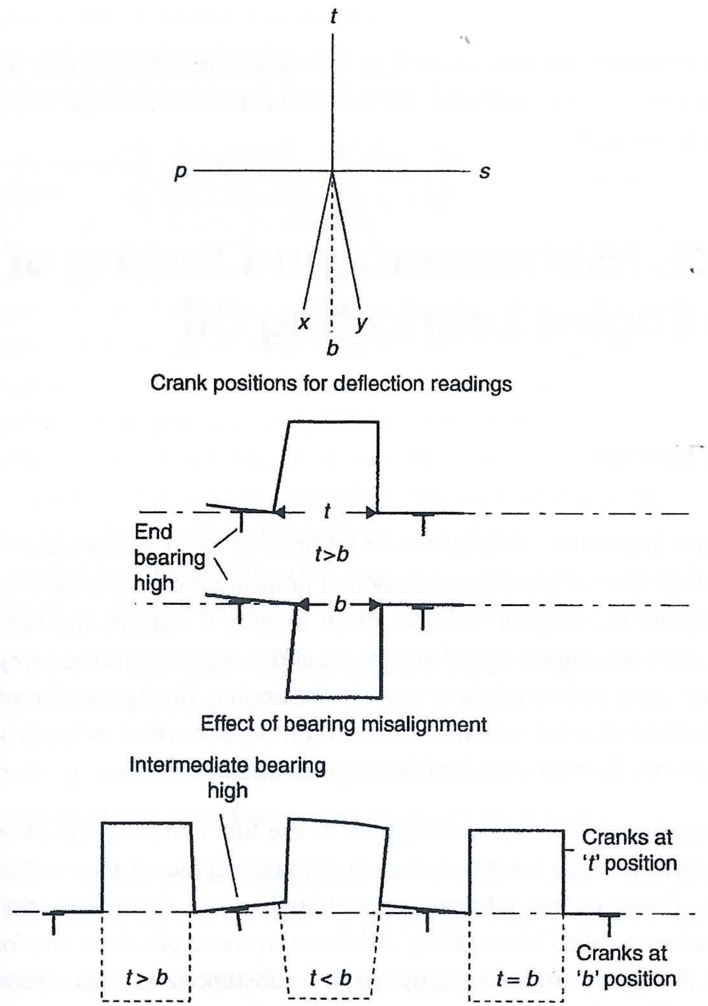
Table 2.1 *Gauge readings in mm/100*

Crank position	Cylinder number					
	1	2	3	4	5	6
x	0	0	0	0	0	0
p	5	2	6	−8	−3	1
t	10	3	12	−14	−8	4
s	5	3	6	−8	−6	3
y	−2	2	−2	0	0	−2
b=(*x*+*y*)/2	−1	1	−1	0	0	−1
Vertical misalignment (*t*–*b*)	11	2	13	−14	−8	5
Horizontal misalignment (*p*–*s*)	0	−1	0	0	3	−2

flexibility is required for the crankshaft as might have been the case for opposed piston engines with large distances between the main bearings.

The vertical misalignment figures shown in figure 2.24 give the reader information that the end main bearing adjacent to No. 1 cylinder and the main bearing between Nos 3 and 4 cylinders are high. Vertical and horizontal misalignments can be checked against the permissible values supplied by the engine builder, often in the form of a graph. If any values exceed or equal maximum permissible values then bearings will have to be adjusted or renewed where required. Indication of incorrect bearing clearances may be given when the engine is running. In the case of medium- or high-speed diesels, load reversal at the bearings generally occurs. With excessive bearing clearances loud knocking takes place and then white metal usually gets hammered out.

If bearing lubrication for a unit is from the same source as piston cooling, then a decrease in the amount of cooling oil return may be observed in the sight glass, together with an increase in its temperature. If bearing clearances are too small, overheating and possible seizure may take place. Increased oil mist and vapour at a particular unit may be observed – together with an increase in bearing temperature which could then lead to a crankcase explosion. Regular checks must be made to ascertain the oxidation rate of the oil. If this increases high temperatures are encountered and as the oil oxidises (burns) its colour blackens.



▲ **Figure 2.24** *Crank positions for deflection*

where the technology is already starting to take hold and one area is with the recording of crankshaft deflections. The job traditionally was very tedious, messy and awkward. Engineers invariably had a 'crankcase' boiler suit that they would wear for jobs such as this and then discard upon completion due to the oil that will have dropped onto them inside the crankcase.

The Bluetooth-enabled measuring device can be placed in position between the marks on the crank webs while they are close to the open crankcase door. The crankshaft can device. The engineer can then download the information to a laptop also using a wireless connection. Dimensions are still recorded with the crankshaft in the positions indicated in figure 2.24.

Choice, Maintenance and Testing of Main Engine Lubricating Oil

Choice of base oil

There is more about the composition of lubricating oil in Volume 8 of the Reed's series. However, it is sufficient here to say that lubricating oil has several roles to play in protecting the moving components of an engine. It reduces the heat stress by cooling the internal components of the engine and it also protects the components by keeping them apart as they rotate or move in the engine. Finally, the oil must keep the components clean and free from the effects of combustion which includes neutralising any acids that may form as a result of the engine's operation.

The 'trunk type' engine is more susceptible to the fuel and products of combustion finding their way into the crankcase, and contaminating the oil, than is the crosshead type engine. Diesel engine lubricating oil should have a detergent additive; these oils are sometimes called 'Heavy Duty'. Additives in these oils deter the formation of deposits on the metal components, by keeping substances, such as carbon particles, in suspension. They also counteract the corrosive effect of sulphur compounds, some of the fuels used may be low in sulphur content, in this case the alkaline additive in the lubricating oil could be less.

The lubrication of large two-stroke low-speed engines is twofold. The crankcase oil will use a straightforward mineral oil, generally with an anti-oxidant and corrosion inhibitor added because the working cylinder is separate from the crankcase and there is less in the way of contaminants from the product of combustion.

Due to the crankcase being separate from the underside of the cylinder, no oil is thrown onto the cylinder walls and therefore this type of engine has to have cylinder

The sulphur content of the fuel has an effect on its lubricity and on its readiness to produce acids when combined with water. Therefore, when the type of fuel is changed over the grade of cylinder oil needs to be changed as well.

Maintenance

When the engine is new correct pre-commissioning should deliver a clean system free from sand, metal, dust, water and other foreign matter. To clear the system of contaminants all parts must be vibrated by hammering or some other such method to loosen rust flakes, scale and weld spatter (if this is not done then these things will work loose when the engine is running and cause damage). A good flushing oil should then be used and a clear discharge should be obtained from the pipes before they are connected up; filters must also be opened up and cleaned during this stage. Finally, the flushing operation should be frequently repeated with a new charge of oil of the type to be used in the engine. When the engine is running, continuous filtration and centrifugal purification is essential.

Oxidation of the oil is one of the major causes of its deterioration, it is caused by high temperatures. This may be due to:

1. Small bearing clearances (hence insufficient cooling).
2. Not continuing to circulate the oil upon stopping the engine.

In the case of oil-cooled piston types, piston temperatures could rise as the residual heat soaks into the piston and the static oil within them becomes overheated.

3. Incorrect use of oil preheater for the purifier, for example, shutting off oil before the heat or running the unit part full.
4. Metal particles of iron and copper can act as catalysts that assist in accelerating oxidation action. Rust and varnish products can behave in a similar fashion.

When warm oil is standing in a tank, water that may be in it can evaporate and condense out upon the upper cooler surfaces of the tank not covered by oil. Rusting could take place and vibration may cause this rust to fall into the oil. Tanks should be given some protective type of coating to avoid rusting.

Oil from the scavenge space and stuffing box drains should not be put into the main oil system and the piston rod gland (commonly known as the stuffing box) and any

Regular examination and testing of the main circulating oil is important. Samples should be taken from a pipeline in which the oil is flowing and not from a tank or container in which the oil is stationary where it could possibly have stagnated and accumulated contaminants that are not from the engine. If this happens a representative sample of the oil lubrication of the engine will not have been taken and if an adverse analysis is subsequently returned then the wrong corrective action could be taken.

Smelling the oil sample may give an indication of fuel oil contamination, or if an acrid smell is present this could be a sign of heavy oxidation. Dark colour gives an indication of oil deterioration possibly due to oxidation and a black colour denotes the presence of carbon.

Dipping fingers into the oil and rubbing the tips together might detect reduction in oiliness – generally due to fuel contamination – and the presence of abrasive particles. The latter may occur if a filter has been incorrectly assembled, damaged or automatically by-passed. Water vapour can condense on the surfaces of sight glasses, thus giving an indication of water contamination. But various tests are available to detect water in oil, for example, immersing a piece of glass in the oil, water finding paper or paste – copper sulphate crystals change colour from white to blue in the presence of water – plunging a piece of heated metal such as a soldering iron into the oil causes spluttering if water is present.

A check on the amount of sludge being removed from the oil by the purifier is important; an increase would give an indication of oil deterioration. Lacquer formation on bearings and excessive carbon formation in oil-cooled pistons are other indications of oil deterioration.

Oil samples for analysis ashore should be taken about every 1,000–2,000 h (or more often if defects are suspected) and it would be recommended that the oil be changed if one or more of the following limiting values are reached:

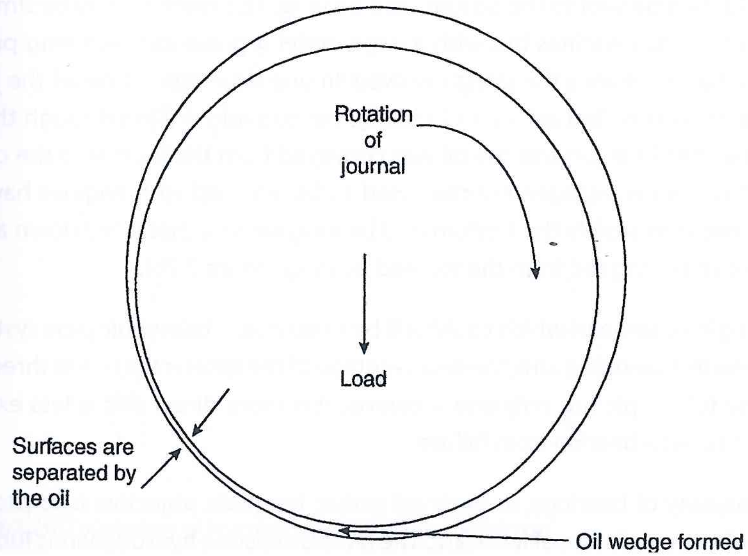
1. 5% change in the viscosity from new. Viscosity increases with oxidation and by contamination with heavy fuel, diesel oil can reduce viscosity.
2. 0.5% contamination of the oil.
3. 0.5% emulsification of the oil, this is also an indication of water content. Fresh water is generally permissible up to 0.2%, but sea water is dangerous.
4. 1.0% Conradson carbon value. This is a measure of the carbon residue left after

sea water give the inorganic and oxidation produces the weak organic acids. The overall number gives an indication of the overall quality of the oil.

Determining the quality of lubricating oil in an engine is the basis of the modern CBM schemes, which are discussed in more detail in chapter 12 of Volume 8. However, regular oil sample analysis will produce a string of results from which trends can be seen and the internal condition of an engine can be determined. This internal condition can then be used as a basis for determining the maintenance required.

Lubrication Systems

Diesel engine bearings are kept in good condition for the working life of the engine by effective lubrication. The designs of the lubrication system can be the make or break of the success of the engine. The objective for good bearing lubrication is to create the correct environment for 'hydrodynamic' lubrication to occur. This is where a film of oil is set up to lift the journal away from the bearing surface. In this case the two surfaces never touch and the only friction is that which is within the structure of the oil (figure 2.25).



Hydrodynamic lubrication is a continual process all the time a shaft journal is rotating in a bearing. The oil forms a wedge between the two components and the journal travels in its circular motion on the wedge of oil. Relating to a crankshaft main bearing, as the load on the shaft grows due to the forces of combustion the oil wedge is subject to more stress and becomes thinner. Eventually the oil film would break down and the journal would come into contact with the bearing surface. Similarly if the viscosity of the oil was low – possibly due to contamination or by using the wrong type of oil – then again the oil film will break down.

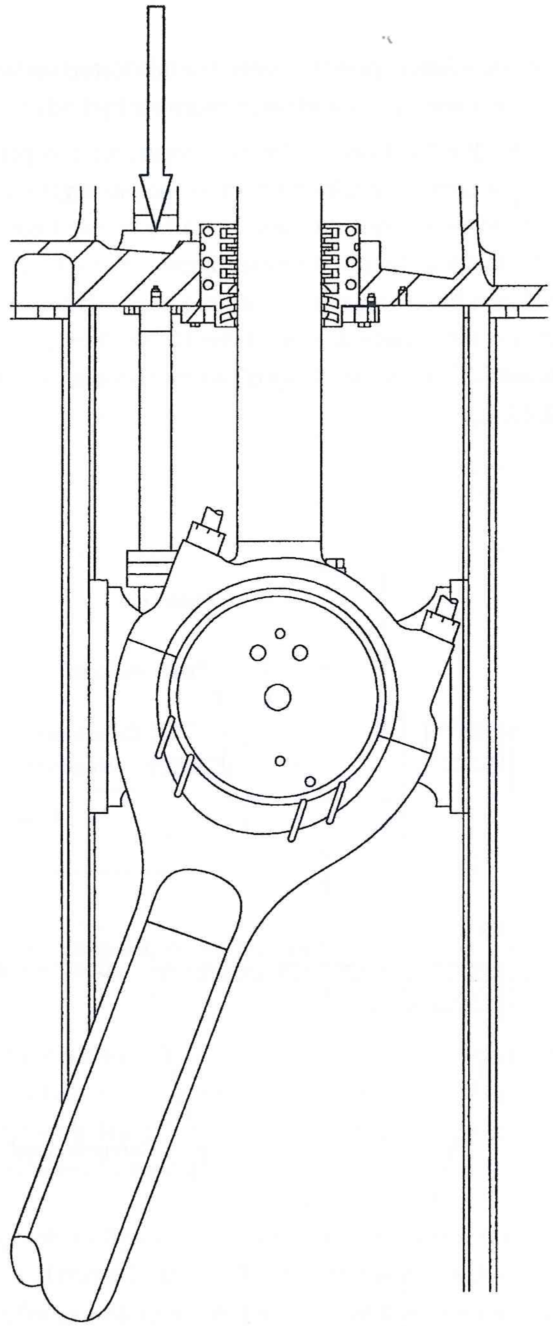
The crosshead bearing on the other hand is a different proposition. This bearing carries out an oscillating motion; therefore as soon as it starts moving in one direction the bearing stops and reverses direction. This means that it does not have sufficient time to set up efficient hydrodynamic lubrication and as a consequence the effective lubrication of this bearing has been difficult. The 'two–pillar' design, which until recently has been a feature of these bearings, does not help due to the reduced surface area.

Engine lubrication systems for the bearings and guides, etc. should be simple and effective. Considering the lubrication of a bottom end bearing, various alternative routes, are available, to channel the oil to the appropriate places and the objective would be to chöose a route that will be the most reliable, least expensive and least complicated.

Oil could be supplied to the main bearing and by means of holes drilled in the crankshaft the oil could then be sent to the bottom end bearing. This method may be simple and satisfactory for small engines but with a large diesel it presents machining problems which would also enhance the stress involved. In one large type of diesel the journals and crankpins were drilled axially and radially, but to avoid drilling through the crank web and the *shrinkage surfaces* the oil was conveyed from the journal to the crankpin by pipes. A common arrangement that used to be adopted with engines having oil-cooled pistons is to supply the bottom end bearing with oil that is led down a central hole in the connecting rod from the top end bearing (figure 2.26).

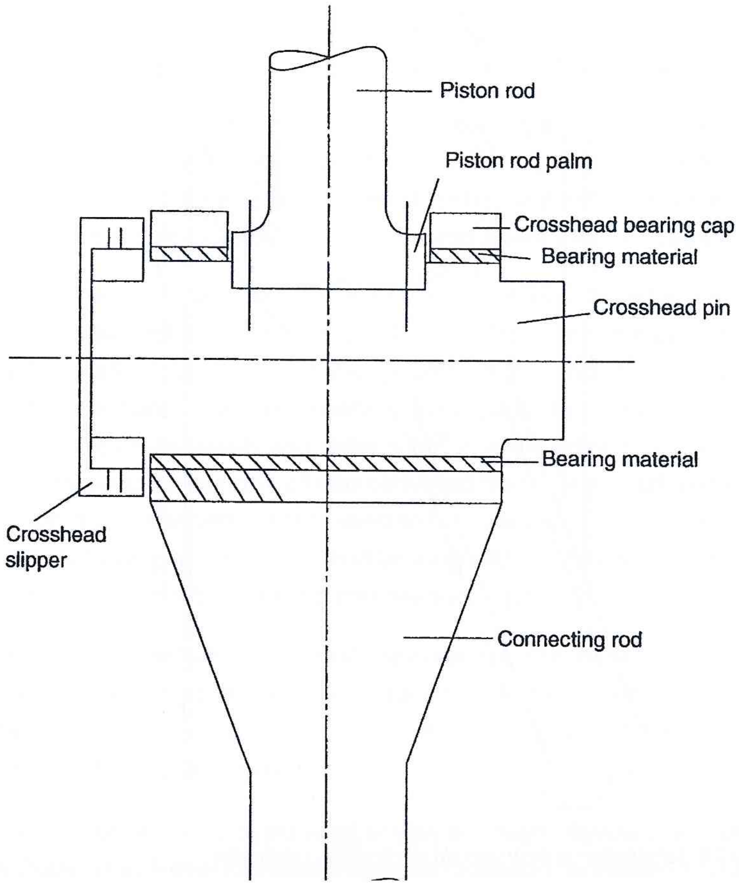
On older engines, some of which could still be in service, a telescopic pipe system was used along with a swinging arm, the disadvantage of the latter is that it has three glands whereas the telescopic has only one. However, it is more direct and is less expensive especially it saved a bearing from failure.

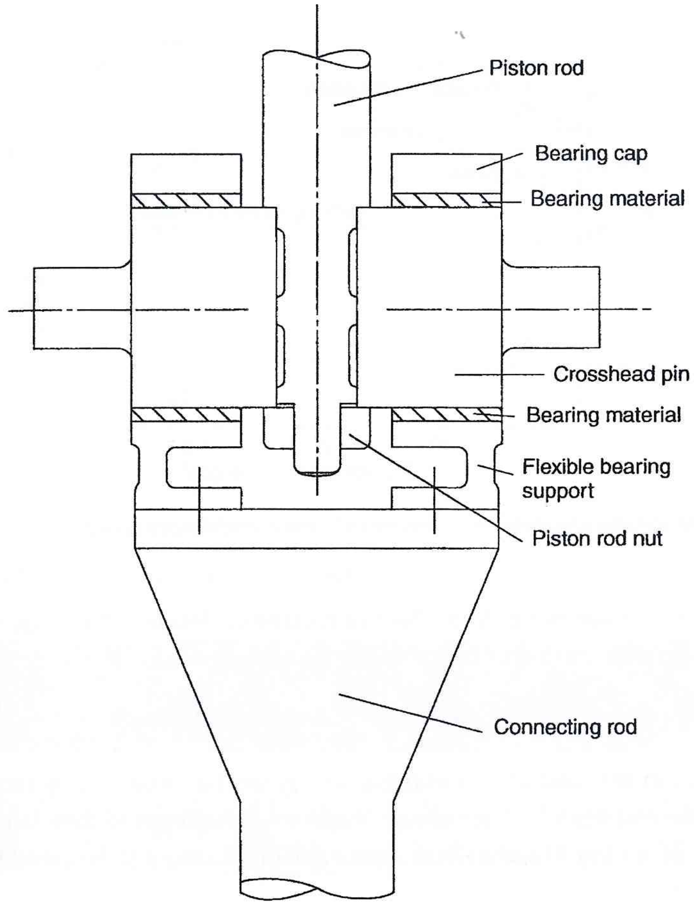
With the majority of bearings, as outlined earlier, the main objective is to provide an efficient hydrodynamic film of lubricant. The factors assisting hydrodynamic lubrication



▲ **Figure 2.26** *Lubrication of bearings via a telescopic connection*

2. *Speed:* Increasing the relative speed between the lubricated surfaces pumps oil into the clearance space more rapidly and helps promote hydrodynamic lubrication.
3. *Pressure:* Increasing the load on the bearing increases the pressure (load/area) on the oil causing a break down in the oil film. Increasing the load at the engine design stage can be offset by increasing the area of the bearing surface which could be done by making the pin diameter larger – this will also increase relative speed. Manufacturers have now redesigned the crosshead bearing on their latest engines, to increase the surface area available to take the load from the forces of combustion (figure 2.27). Previous designs had much lower surface areas as can be seen in figure 2.28.

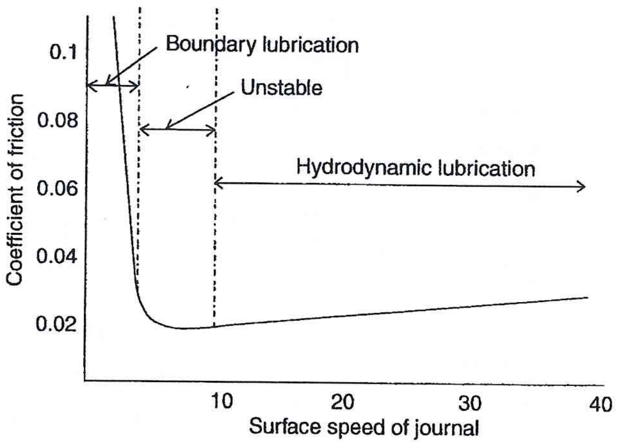




▲ **Figure 2.28** *Crosshead with flexible bearing supports*

4. *Clearance:* If bearing clearance is too great then the inertia forces lead to 'bearing knock'. This impulsive loading results in pressure above normal and break down of the hydrodynamic layer. Figure 2.29 illustrates these points graphically for a rotating journal type of bearing.

Referring to the graph in figure 2.29, hydrodynamic lubrication should exist in the main, bottom end and guide bearings. The top end and crosshead bearings will have a variable condition, for example, when the unit is at TDC the relative velocity between crosshead guide and the bearing surface is zero and the pressure from combustion is building. The swing of the connecting rod is however building the relative speed of

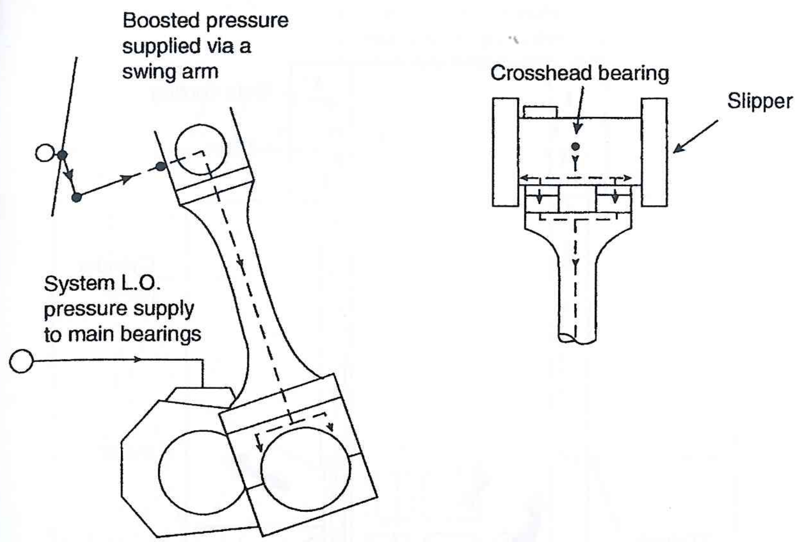


▲ **Figure 2.29** *Relationship between coefficient of friction and surface speed*

complications involved not only in effective crosshead design in the large two-stroke engine but also with 'small end' design in the four-stroke engine.

Methods of improving top end bearing lubrication are as follows:

1. Reduction in the load on top end exerted by inertia forces – only required with four-stroke engines which are usually medium- or high-speed diesels, although at the time of writing Akasaka Diesels were still producing a slow-speed four-stroke engine
2. Use as large a surface area as possible, that is, the complete underside of the crosshead pin (figure 2.27).
3. Avoid large axial variation of bearing pressure by more flexible seating and design (figure 2.28).
4. Increase oil supply pressure – on the modern engines this is achieved by using a pump external to the engine to supply oil pressure directly to the top end bearing which tends to keep the crosshead pin 'floating' at all times as the crosshead bearing rocks to and fro.
5. An increase in oil supply pressure to the crosshead bearing can also be accomplished by the oil inlet to the engine being at the position of the crosshead enabling the full pressure of the lubrication oil system to be placed on the crosshead bearing.



▲ **Figure 2.30** *Lubrication system for main bearings*

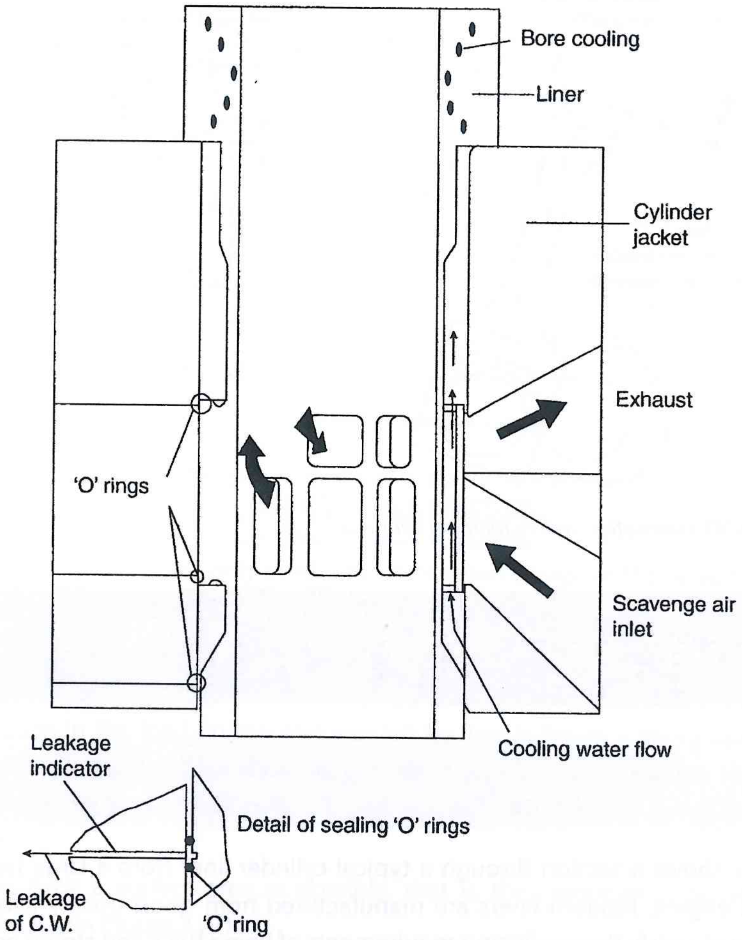
Cylinders and Pistons

Cylinders

Figure 2.31 shows a section through a typical cylinder liner from a large two-stroke low-speed engine. Modern liners are manufactured from good quality alloyed cast iron and must satisfy the conflicting requirements of being thick and strong enough to withstand the high pressures and temperatures that occur during combustion and thin enough to allow good heat transfer.

This conflict is reconciled by the use of bore cooling. Figure 2.32 illustrates that by boring the upper part of the liner at an angle to the longitudinal axis the bore at mid-point is close to the surface of the liner. The close proximity of the liner surface to the cooling water results in effective heat transfer. By using this technique of bore cooling good heat transfer is accompanied by high overall strength.

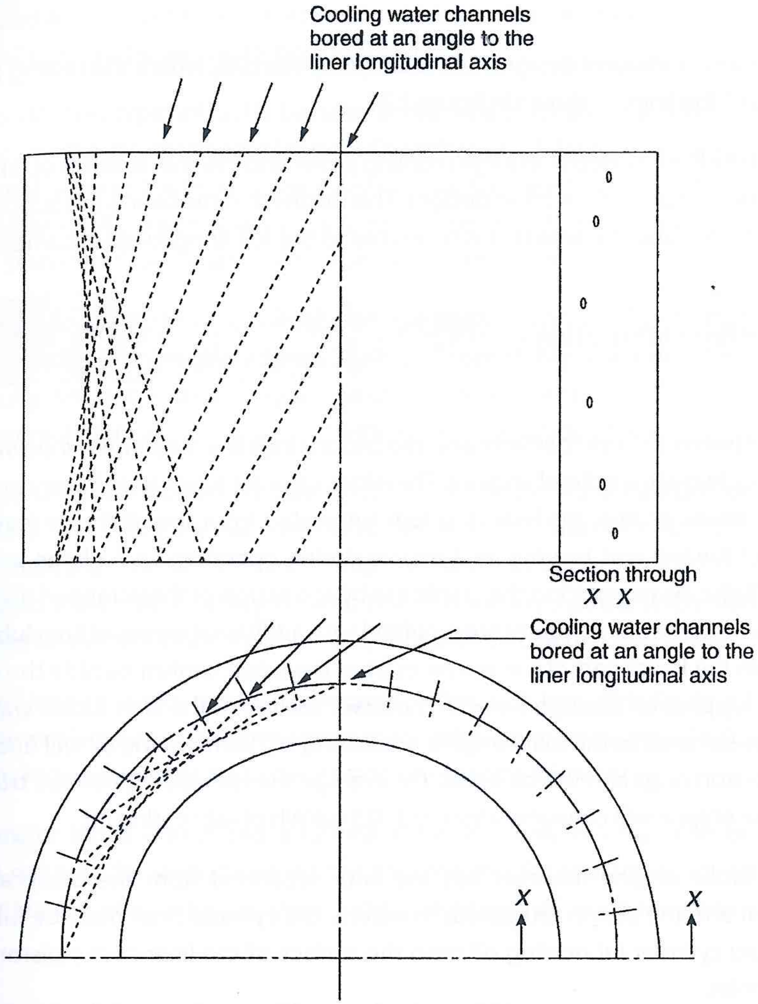
Maintaining the correct surface temperatures in the vicinity of the combustion space by good heat transfer does however cause the risk of low temperature corrosion or



▲ **Figure 2.31** *Two-stroke cylinder liner*

cylinder cooling system to maintain optimum cylinder liner temperature (figure 7.14 in Chapter 7).

Longitudinal expansion of the liner takes place through the lower cooling jacket. The sealing of the cooling water is accomplished by silicone rubber 'O' rings installed in grooves machined in the liner, which slide over the jacket as the liner expands and contracts (figure 2.31). The 'O' rings are in groups of two, the space between them being open to the atmosphere via an inspection hole. Leakage of water past a 'O' ...



▲ **Figure 2.32** *Diagrammatic view of cylinder liner bore cooling*

practice of keeping the engine cooling water at the normal operating temperature when the engine is shut down will mean that the liner expansion/contraction is kept as small as possible.

Modern materials have recently enabled significant advances in cylinder liner design with the liner walls becoming much thinner but still giving the same strength as before. The cylinder liners in MAN Diesel & Turbo two-stroke engines are made from alloyed cast iron and are placed in the engine mono-block by sitting on a flange which is designed from the more traditional designs, still retained by Wärtsilä, where the resting flange is at the top of the liner, as shown in figure 2.31.

The top of the liner is fitted with a light cooling jacket and the liner extends up above the cylinder frame more than in other designs. This means that the liner is not supported by its neck which is also the area that is subjected to the full firing loads of combustion.

Cylinder liner lubrication

Friction between the cylinder liner and the piston rings is a major loss of power in the drive line of.two-stroke diesel engines. Therefore, cylinder lubrication is very important on a four-stroke engine; the liner is 'splash' lubricated from the oil that is leaving the surfaces of the big end bearing as it rotates during operation and a large volume of the oil will also be returned to the crankcase by the action of the 'scrapper' ring on the piston. Some four-stroke engines are provided with additional means of liner lubrication either from the lower part of the piston or from the a mechanism outside the cylinder liner that supplies oil through holes in the lower section of the liner. Either way the oil used is the same oil as the main engine lubricating oil. Some of the oil will find its way past the piston rings and will be burnt. On average a large well-maintained trunk type four-stroke engine will consume about 0.3–0.5 g/kWh of lubricating oil.

The two-stroke engine however has the liner separated from the crankcase by a diaphragm and the piston rod gland. Therefore, the cylinder liner must be lubricated by injecting cylinder lubricating oil onto the surface of the liner as the piston moves past the holes.

This system is a 'total loss' lubricating system meaning that when the oil is injected to lubricate the cylinder and having completed its task the oil is either burnt or it comes out in the scavenge space. This system of having the two lubricating systems separate means that a different cylinder oil can be used to the main lubricating oil for the engine.

As the lubricating oil is 'lost' it also means that this operation is costly and therefore as technology has improved so has the research into lowering consumption by studying the path of the cylinder oil through the engine. The problem has recently been compounded by the notification of the introduction of the use of LSF (below

sulphur) and another with the LSF. At least one major oil company has now produced a cylinder oil that can be used with both LSF and HSF.

The reason for two types of fuel is because at the time of writing the countries that are signatories to the International Maritime Organisation (IMO) have agreed that there will be areas around the world, available to ships, that are to be classed as Emission Control Areas (ECA). When a vessel sails into those areas then the emissions (mostly NOx and SOx) from flue gasses must be below certain levels.

Currently the answer to this is to switch from common heavy fuel oil onto the LSF as the vessel approaches the designated area. The experience of ship operators suggested that the difficulty was not so much a question of switching fuels but more about ensuring that the correct lubricant is being used with a particular grade of fuel.

Engine manufacturer guidelines state that lubricants which are suitable for use with high sulphur fuel (HSF) are not suitable for use with LSF and, technically, a different base number (BN) lubricant should be used. A low BN, typically 50 or 40, corresponds with LSF and a high BN, typically 70, with HSF. This means that the ship's staff will also be required to switch lubricants when they switch from one fuel to another.

The use of lower basicity cylinder lubricants within ECA runs directly counter to the lubrication requirements for slow steaming or other conditions outside ECAs, which conversely requires owners and operators to run specific lubricants.

One oil major Mobil introduced a lubricant that not only has the high detergency (basicity) required for slow steaming but also the low BN characteristics needed for a lubricant being used with LSF. The principal objectives of cylinder lubrication are as follows:

1. To separate sliding surfaces with an unbroken oil film.
2. To form an effective seal between piston rings and cylinder liner surface to prevent blow past of gases.
3. To neutralise corrosive combustion products and thus protect cylinder liner, piston and rings from corrosive attack.
4. To soften deposits and thus prevent wear due to abrasion.
5. To remove deposits to prevent seizure of piston rings and keep engine clean.
6. To cool hot surfaces without burning.

In practice some oil burning will take place, if excessive this would be indicated of computer-controlled systems such as the MAN Diesel & Turbo 'Alpha Adaptive Cylinder-oil Control' (Alpha ACC) or the Wärtsilä RTA Pulse Lubrication System (PLS). The average oil consumption for this type of system woul be 0.7–0.9 g/kWh.

When the engine is new, cylinder lubrication rate should normally be greater than when the engine becomes run in. Reasons for this initial increased lubrication are as follows:

1. Liner surface unevenness will cause localized high temperatures which in turn will cause increased oxidation of the oil and reduce its lubrication properties.
2. Sealing of the rough surfaces is more difficult.
3. Worn metal needs to be washed away.

The actual amount of lubricating oil to be delivered into a cylinder per unit time depends upon stroke, bore and speed of engine, engine load, cylinder temperature, type of engine, position of cylinder lubricators and type of fuel being burnt.

Position of the cylinder lubricators for injection of oil has always been a topic of discussion, the following points are of importance:

1. They must not be situated too near the ports, oil can be scraped over edge of ports and blown away.
2. They should not be situated too near the high temperature zone or the oil will burn easily.
3. There must be sufficient points to ensure as even and as complete a coverage as possible.

The modern systems will deliver the oil to the point of use within 8–10 ms and therefore timed injection is possible. The objective is to inject the cylinder lubricating oil into the piston ring pack when the piston rings pass the quill level. Some systems deliver a measured amount of oil with every revolution while others deliver a larger dose every fifth or sixth revolution.

Once the lubricator pump has delivered the cylinder oil to the quills in the cylinder liner it gets picked up by the piston ring pack as they pass the openings in the liner wall. The key to success is for an even distribution of the oil on the cylinder liner's running surface and to keep the oil on the cylinder wall replenished to provide sufficient additives to neutralise the acid that will be forming and to keep the liner clean.

around the liner 900 mm from the top. The lubricator delivers 310 mm^3 to each time it is activated which depending upon the speed and load could be every two to five revolutions of the engine. Therefore, 8.3 m^2 is lubricated by 8×310 mm^3 which then has to be evenly distributed across, up and down the liner.

Vertical distribution of the cylinder oil is mainly performed by the piston rings during their travel up and down the cylinder. The cylinder oil is injected into the piston ring pack during the piston's upward stroke; factors such as oil viscosity, feed rate and the volume of oil per injection are calculated by the control software. The correct oil viscosity is important to encourage the spreadability of the cylinder oil. The applied feed rate and volume of oil injected for each operation of the lubricator are key factors in the critical balance between under and over lubrication.

There was some experimentation with the new PSL system before the 'zig–zag' groove principle was adopted for all new engines and retro-fitted to a high number of large bore engine cylinder liners. This system measures the temperature in two diametrically opposite positions near the running surface in the upper part of each cylinder liner. It then filters and interprets the development of the temperatures, and in case the temperature level escalates, a 'high friction alarm' is generated.

Under-lubrication could lead to corrosion, accumulated contamination from unburned fuel and combustion residues and in the worst case metal-to-metal contact, known as 'scuffing'. Over-lubrication can lead to a number of problems, including the

- loss of unused oil in the scavenge ports
- piston rings being prevented from moving (rotating) in their grooves by 'hydraulic lock'
- 'chemical bore polish'
- 'mechanical bore polish'.

Cylinder liner wear

Cylinder liner wear can be divided into:

- Abrasive wear
- Corrosive wear.

Abrasive wear

air filtration system is very important for the continued good health of the engine's internal components.

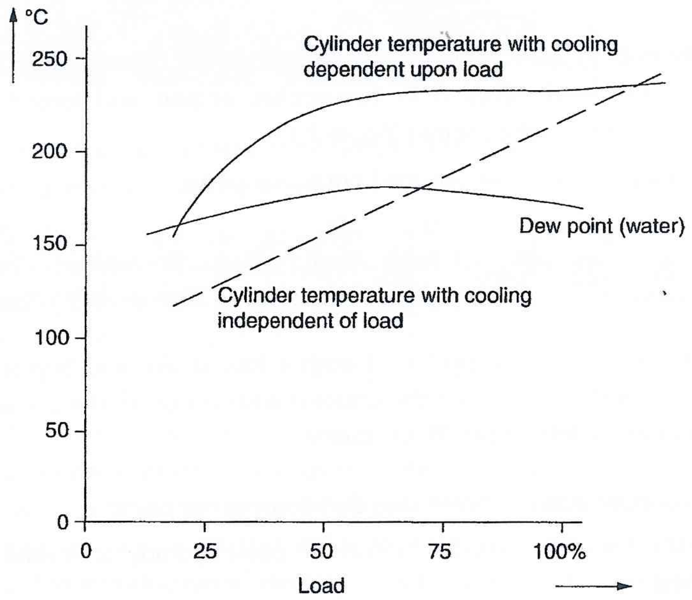
The quality of bunker fuel will be discussed in more detail in Chapter 3 of this volume. However, it is appropriate to say at this point that because marine engines burn residual fuel oil it is very easy to also have within the fuel, components that are left over from the refining process. Catalytic cracking of oil during the refining process uses very hard particles of aluminium and silicon that come about due to the catalytic cracking process in the refinery. They are in a form of complex alumino-silicates and can vary in size and hardness. Bunker specification should mean that these are not in the final product but all too often they get through the quality assurance processes. It is important to have samples taken at the vessels' bunker station while bunkering is taking place. If there is any damage to the engine that could be because of 'Cat' fines then evidence will be required and without the samples a successful outcome might be difficult.

Corrosive wear

Overall liner wear is reducing as understanding about the lubrication process improves. Corrosive wear is still a more common cause of cylinder liner wear and comes from burning heavy fuel containing significant amounts of sulphur. As the fuel burns the sulphur combines with oxygen to produce oxides of sulphur which further form sulfuric acid on contact with water. To minimise the formation of acids it is important that cylinder liner temperatures are maintained above the dew-point (figure 2.33).

Good operational practice is the key to keeping cylinder liner wear as low as possible. It is very important that ship's engineers understand how to operate the engine correctly. This includes:

- Correct quantity and grade of cylinder lubrication.
- Correctly fitted piston rings.
- Correct warming through prior to starting.
- Well-maintained and timed fuel injectors.
- Well-managed fuel storage and purification plant.
- Correct cooling water and lubricating oil temperatures.
- Correct scavenge air temperatures.
- Engine load changes carried out gradually.



▲ **Figure 2.33** *Temperature of cylinder liner surface throughout. Engine load range*

resulted in liners and piston rings operating under very severe conditions. Despite these adverse operating conditions cylinder liner wear rates have been reduced with large two-stroke manufacturers claiming 0.03 mm/1,000 h and medium-speed four-stroke engine manufacturers claiming wear rates of 0.02 mm/1,000 h when operating on heavy fuel and with the recent trial of a stepped piston and liner arrangement MAN Diesel have reported a wear rate as low as 0.0045 mm/1,000 h. These advancements are helping to extend the time between overhaul (TBO) and manufacturers are working towards running main engines from dry dock to dry dock without the need of a major overhaul where the piston has to be removed.

These wear rates have been achieved as a result of a number of factors, such as:

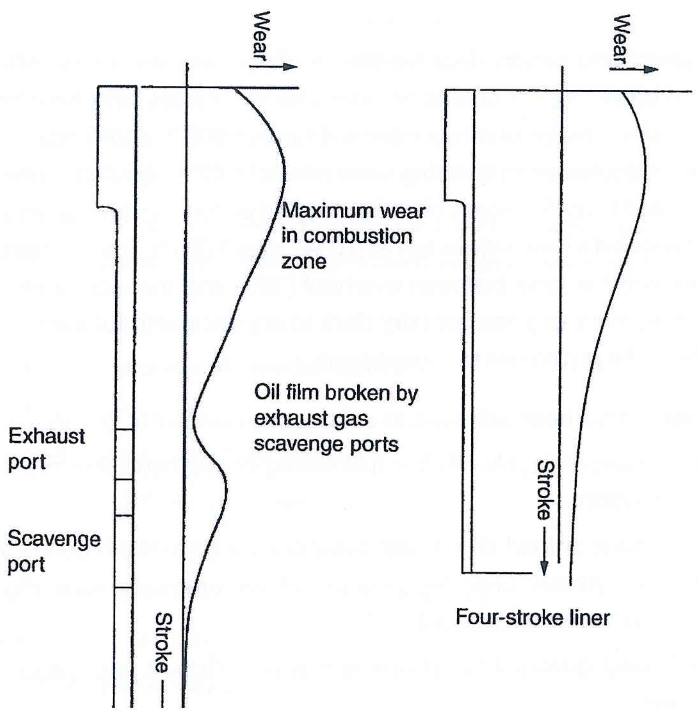
- The development of highly alkaline lubricating oils to neutralise the acids formed during combustion.
- The development of load-dependent temperature control of cooling water which maintains the cylinder liner temperature at an optimum level (figure 7.14 in Chapter 7) [cooling water section].
- The use of good quality alloyed cast iron with sufficient hard phase content for cylinder liners.

- Improvements in lubricating oil distribution across cylinder liner surface. This includes multi-level injection in two-strokes engine and forced piston skirt lubrication in four-stroke engines (figure 2.37).
- Improved separation of condensate from scavenge air.

Cylinder liner wear profile

Figure 2.34 shows the wear profile of both a four-stroke and two-stroke engine cylinder liner. It can be seen that the greatest wear occurs in the upper part of the liner adjacent to the firing zone. This is due to:

- The high temperatures and pressures that occur at this point.
- Because the piston reverses direction at this point hydrodynamic lubrication is not established.
- Acids formed during combustion attack the liner material.



Slow steaming

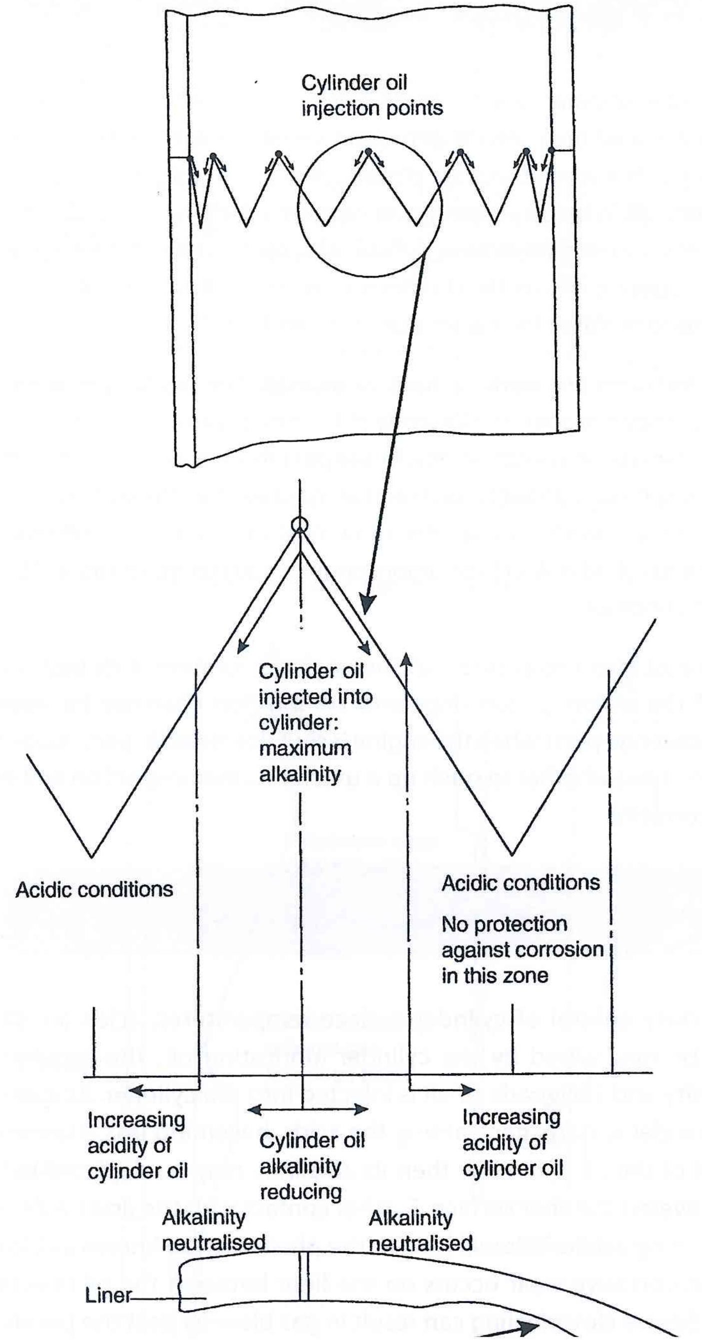
During different economic climates ships will need to operate in different ways, for example, a high-speed liner service between two ports might be the correct strategy during a rising or buoyant economic climate. However it might not be correct if the trading markets fall. When the changes in economic fortune come about during the lifetime of a vessel then owners have difficult choices to make. If the engine is flexible and is able to operate efficiently at different speeds and power outputs then it will prove much more useful to the owner due to its flexibility.

Engine manufacturers are working hard to increase the working envelope of their engines so that they will operate efficiently at full design power/speed output and also at reduced power/speed combinations. In the past this has proved to be very difficult to achieve as engine components, such as fuel injectors, had to be changed when slow steaming. With the introduction of electronic control and a better understanding of processes such as cylinder liner lubrication, engines can run much more efficiently and under varying conditions.

However, the best check to ensure that the engine is running at its best is by regular inspection of the piston, piston rings and combustion chamber by viewing them through the scavenge ports when the engine is shut down and in port. Judgements can then be made about whether to open up a unit for further inspection or if everything is operating correctly.

Cloverleafing

Despite the close control of cylinder surface temperatures, acids are still formed which must be neutralised by the cylinder lubricating oil. This requires that the correct quantity and TBN grade of oil is injected into the cylinder. As soon as the oil enters the cylinder it starts neutralising the acids, becoming less alkaline as it does so. If the TBN of the oil is too low then its alkalinity may be depleted before it has completely covered the liner surface. Further contact with the acids may lead to the oil itself becoming acidic. This will lead to the phenomenon known as 'cloverleafing' in which high corrosive wear occurs on the liner between the oil injection points (figure 2.35). Severe cloverleafing can result in gas blow-by past the piston rings and



Micro-seizure

This is due to irregularities in the liner and piston rings coming into contact during operation as a result of a breakdown of lubrication due to an insufficient quantity of lubricating oil, insufficient viscosity or excessive loading. This results in instantaneous seizure and tearing. In appearance micro-seizure resembles abrasive wear since the characteristic marks run axially on the liner. Micro-seizure may not always be destructive, indeed it often occurs during a running-in period. It becomes destructive if it is persistent and as a result of inadequate lubrication.

Pistons and Rings

Pistons

Pistons must be strong enough to withstand the very high firing pressures that are common today, be able to dissipate sufficient heat to maintain the correct piston crown temperatures and withstand the stresses imposed by friction. Pistons are manufactured from cast steel, forged steel and cast iron although all of these materials have limitations. Cast iron is weak in tension especially at elevated temperatures. It does, however, have high compressive strength which enables it to resist the hammering which occurs at the ring grooves. As a result of of its graphite content, cast iron performs well when exposed to rubbing. This makes it a suitable material for piston skirts. Cast steel resists heat stresses better than cast iron but is difficult to ensure that the molten material flows to the extremities of intricate moulds. Cast steel also requires extensive heat treatment to relieve casting stresses. Forged steel is a suitable material because the directional grain flow exhibited as a result of forging produces a strong tough component. Forged steel is prone to high wear at the ring grooves and also requires a greater degree of machining which tends to increase the production cost.

Modern pistons are composite components, made from materials that exhibit suitable properties for the different parts of the structure.