REED'S MOTOR ENGINEERING KNOWLEDGE FOR MARINE ENGINEERS

Donated by

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REED'S MOTOR ENGINEERING KNOWLEDGE FOR MARINE ENGINEERS

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PREFACE

The object of this book is to prepare students for the Certificates of Competency of the Department of Transport in the subject of Motor Engineering Knowledge.

The text is intended to cover the ground work required for both examinations. The syllabus and principles involved are virtually the same for both examinations but questions set in the First Class require a more detailed answer.

The book is not to be considered as a close detail reference work but rather as a specific examination guide, in particular **all the sketches are intended as direct application to the examination requirements.**

The best method of study is to read carefully through each chapter, practising sketchwork, and when the principles have been mastered to attempt the few examples at the end of the chapter. Finally, the miscellaneous questions at the end of the book should be worked through. The best preparation for any examination is to work on the examples, this is difficult in the subject of Engineering 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 information first and then attempts each question in the book in turn, basing his answer on either a good descriptive sketch and writing or a description covering about 1½ pages of A4 paper in ½ hour.

ACKNOWLEDGEMENTS TO THIRD EDITION

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Anthony S. Prince, 1994.

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CHAPTER 1

BASIC PRINCIPLES

DEFINITIONS AND FORMULAE

Isothermal Operation (PV = constant)
An ideal reversible process at constant temperature. Follows Boyle's law, requiring heat addition during expansion and heat extraction during compression. Impractical due to requirement of very slow piston speeds.

Adiabatic Operation (PV^{γ} = constant)
An ideal reversible process with no heat addition or extraction. Work done is equivalent to the change of internal energy. Requires impractically high piston speeds.

Polytropic Operation (PV^{n} = constant)
A more nearly practical process. The value of index n usually lies between unity and gamma.

Volumetric Efficiency
A comparison between the mass of air induced per cycle and the mass of air contained in the stroke volume at standard conditions. Usually used to describe 4-stroke engines and air compressors. The general value is about 90 per cent.

Scavenge Efficiency
Similar to volumetric efficiency but used to describe 2-stroke engines where some gas may be included with the air at the start of compression. Both efficiency values are reduced by high revolutions, high ambient air temperature.

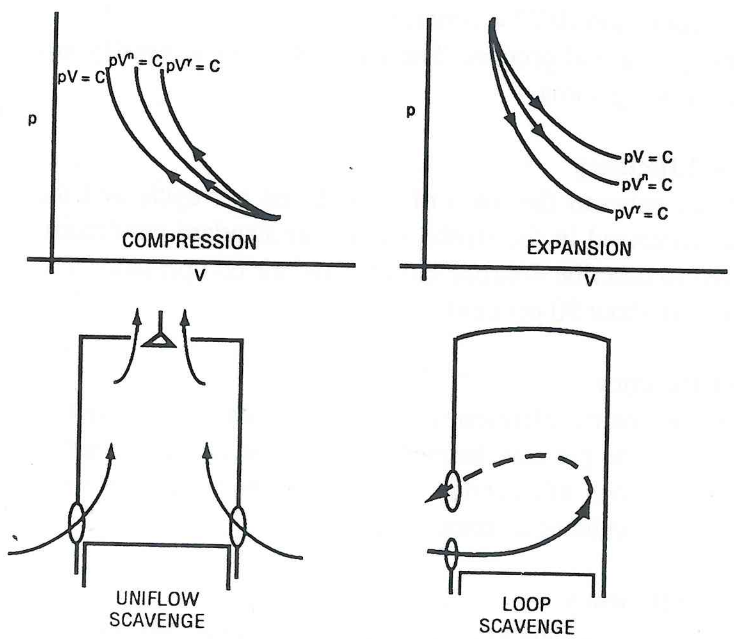
Mechanical Efficiency
A measure of the mechanical perfection of an engine. Numerically expressed as the ratio between the indicated power and the brake power.

Uniflow Scavenge
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 cylinder length, *e.g.* B and W Sulzer RTA (see Fig. 1).

Loop Scavenge
Exhaust and scavenge air entry at one end of cylinder (bottom), *e.g.* Sulzer RD RND and RL. This general classification simplifies and embraces variations of the sketch (Fig.1) in cases where air and exhaust are at different sides of the cylinder with and without crossed flow loop (cross and transverse scavenge)

Brake Thermal Efficiency
The ratio between the energy developed at the brake (output shaft) of the engine and the energy supplied.

FIG 1
COMPRESSION, EXPANSION



Specific Fuel Consumption
Fuel consumption per unit energy at the cylinder or output shaft, kg/kWh (or kg/kWs), 0·19 kg/kWh would be normal on a shaft energy basis for a modern engine.

Compression Ratio
Ratio of the volume of air at the start of the compression stroke to the volume of air at the end of this stroke (inner dead centre). Usual value for a compression ignition (CI) oil engine is about 12·5 to 13·5, *i.e.* clearance volume is 8 per cent of stoke volume.

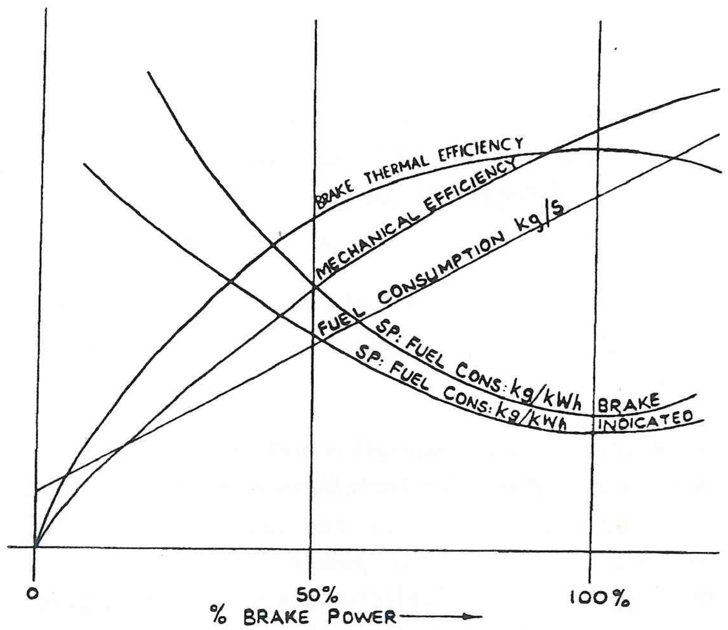
Fuel – Air Ratio
Theoretical air is about 14·5 kg/kg fuel but actual air varies from about 29-44 kg/kg fuel. The percentage excess air is about 150 (36·5 kg/kg fuel).

Performance Curves Fuel Consumption and Efficiency
With main marine engines for merchant ships the optimum designed maximum thermal efficiency (and minimum specific fuel consumption) are arranged for full power conditions. In naval practice minimum specific fuel consumption is at a given percentage of full power for economical speeds but maximum speeds are occasionally required when the specific fuel consumption is much higher. For IC engines driving electrical generators it is often best to arrange peak thermal efficiency at say 70% load maximum as the engine units are probably averaging this load in operation.

The performance curves given in Fig. 2 are useful in establishing principles. The fuel consumption (kg/s) increases steadily with load. Note that halving the load does not halve the fuel consumption as certain essentials consume fuel at no load (*e.g.* heat for cooling water warming through, etc.). Willan's law is a similar illustration in steam engine practice.

Mechanical efficiency steadily increases with load as friction losses are almost constant. Thermal efficiency (brake for example) is designed in this case on the sketch for maximum at full load. 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

FIG 2
PERFORMANCE CURVES



fairly constant friction loss. In designing engines for different types of duty the specific consumption minima may be at a different load point. As quoted earlier this could be about 70% for engines driving electrical generators.

Heat Balance

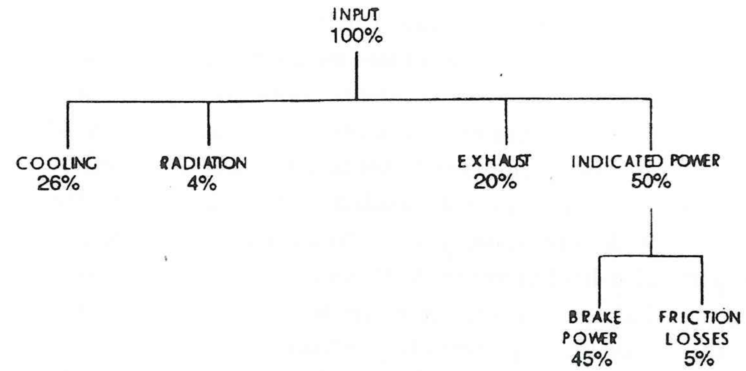
A simple heat balance is shown in Fig. 3.

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 diesel engines is superior to any other form of engine in use at present.

1. The use of a waste heat (exhaust gas) boiler gives a plant efficiency gain as this heat would otherwise be lost up the funnel.

2. Exhaust gas driven turbo-blowers contribute to high

FIG 3
SIMPLE HEAT BALANCE



mechanical efficiency. As the air supply to the engine is not supplied with power directly from the engine, *i.e.* chain driven blowers or direct drive scavenge pumps, then more of the generated power is available for effective brake power.

Consideration of the above shows two basic flaws in the simplification of a heat balance as given in Fig. 3.

(a) The difference between indicated power and brake power is not only the power absorbed in friction. Indicated power is necessarily lost in essential drives for the engine such as camshafts, pumps, etc. which means a reduced potential for brake power.

(b) Friction results in heat generation which is dissipated in fluid cooling media, *i.e.* oil and water, and hence the cooling analysis in a heat balance should include the frictional heat effect as an assessment.

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 practice of utilising a recovery system for heat normally lost in coolant systems.

Load Diagram.
Fig. 4 shows a typical load diagram for a slow-speed 2-stroke engine. It is a graph of brake power and shaft speed. Line 1 represents the power developed by the engine on the test bed and runs through the MCR [maximum continuous rating] point. Lines parallel to 2 represent constant values of p_{mep}. Line 3 shows the maximum shaft speed which should not be exceeded. Line 4 is important since it represents the maximum continuous power and mep, at a given speed, commensurate with an adequate supply of charge air for combustion. Line 5 represents the power absorbed by the propeller when the ship is fully loaded with 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 4 and 5, while a vessel in ballast will operate in the region to the right of 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.

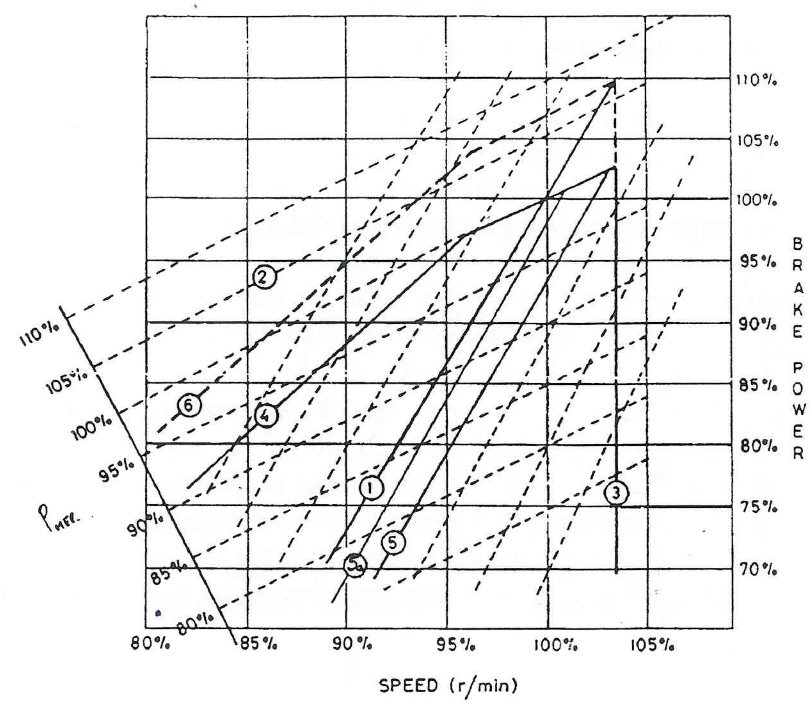
IDEAL CYCLES

These cycles form the basis for reference of the actual performance of IC engines. In the cycles considered in detail all curves are frictionless adiabatic, *i.e.* isentropic. The usual assumptions are made such as constant specific heats, mass of charge 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 and gas engines and, (b) compression ignition (CI) such as diesel and oil engines. Older forms of reference used terms such as light and heavy oil engines but this is not very explicit or satisfactory. Four main air standard cycles are first considered followed by a brief consideration of other such cycles less often considered. The cycles have been sketched using the usual method of P-V diagrams.

Otto (Constant Volume) Cycle
This cycle forms the basis of all SI and high speed CI engines.

The four non-flow operations combined into a cycle are shown in Fig. 5.

FIG 4
ENGINE LOAD DIAGRAM



$$\text{Air Standard Efficiency} = \text{Work Done/Heat Supplied}$$
$$= \frac{(\text{Heat Supplied - Heat Rejected})}{\text{Heat Supplied}}$$

referring to Fig. 5

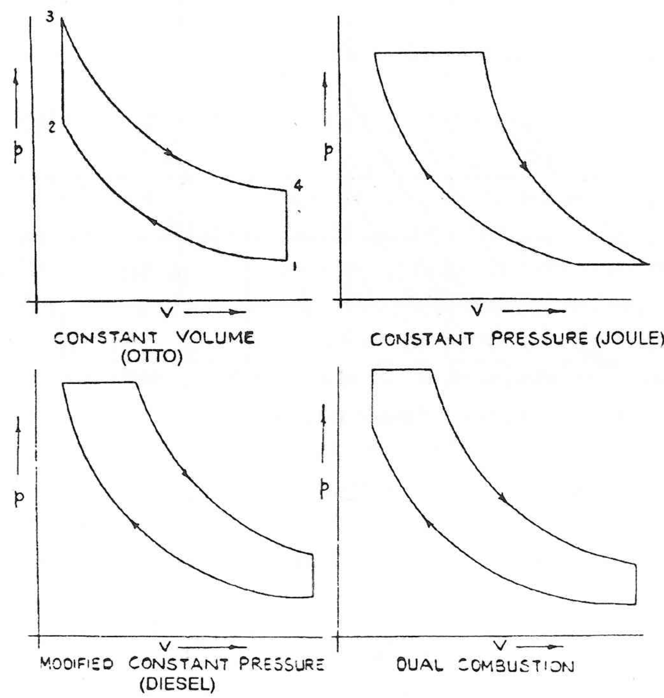
$$\text{Air Standard Efficiency} = 1 - \text{Heat Rejected / Heat Supplied}$$
$$= 1 - MC\,(T_4 - T_1)\,/MC\,(T_3 - T_2).$$
$$= 1 - 1/(r^{\gamma-1})$$

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

Note

Efficiency of the cycle increases with increase of compression ratio. This is true of the other four cycles.

FIG 5
THEORETICAL (IDEAL) CYCLES



Diesel (Modified Constant Pressure) Cycle

This cycle is more applicable to older CI engines utilising long periods of constant pressure fuel injection period in conjunction with blast injection. Modern engines do not in fact aim at this cycle which in its pure form envisages very high compression ratios. The term semi-diesel was used for hot bulb engines using a compression ratio between that of the Otto and the Diesel ideal cycles. Early Doxford engines utilised a form of this principle with low compression pressures and 'hot spot' pistons. The Diesel cycle is also sketched in Fig. 5 and it may 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, *i.e.* the constant pressure period is very short. This produces very heavy firing loads but gives the necessary good combustion.

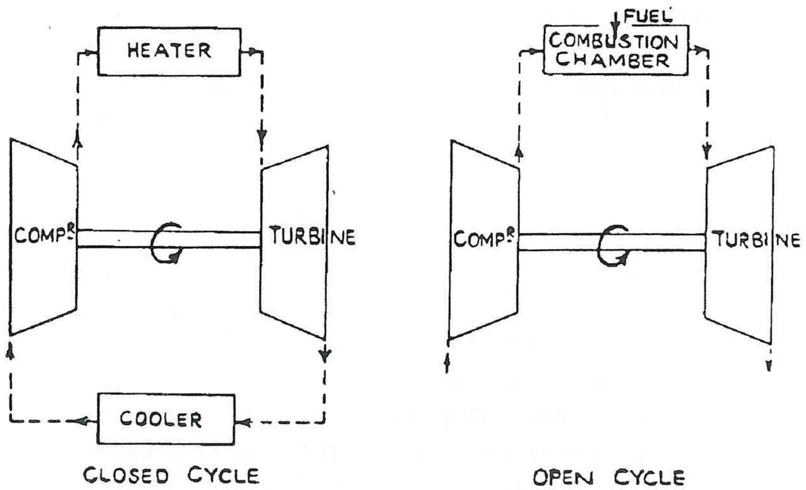
Joule (Constant Pressure) Cycle

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 diagram is shown in Fig 5. and again as a circuit cycle diagram on Fig. 6. in which intercoolers, heat exchangers and reheaters have been omitted for simplicity.

Other Cycles

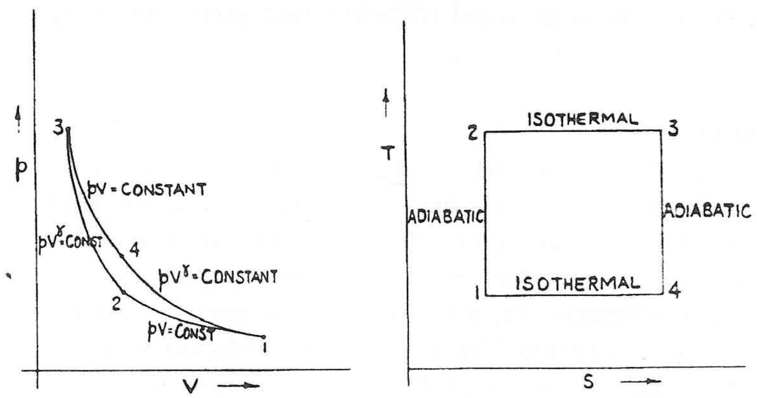
The efficiency of a thermodynamic cycle is a maximum when the cycle is made up of reversible operations. The Carnot cycle of isothermals and adiabatics satisfies this condition and this maximum efficiency is, referring to Fig. 7 given by $(T_3 - T_1)/T_3$ where the Kelvin temperatures are maximum and minimum for the cycle. The cycle is practically not approachable as the mean effective pressure is so small and compression ratio would be excessive. All the four ideal cycles have efficiencies less than the

FIG 6
GAS TURBINE CIRCUIT-CYCLES



Carnot. The Stirling cycle and the Ericsson cycle have equal efficiency to the Carnot. Further research work is being carried out

FIG 7
THEORETICAL (IDEAL) CYCLES



on Stirling cycle engines in an effort to utilise the high thermal efficiency potential. The Carnot cycle is sketched on both P-V and T-S axes Fig 7.

ACTUAL CYCLES AND INDICATOR DIAGRAMS

There is an analogy between the real IC engine cycle and the equivalent air standard cycle in that the P-V diagrams are similar. The differences between these cycles are now considered and for illustration purposes the sketches given are of the Otto cycle. The principles are however generally the same for most IC engine cycles.

(a) The actual compression curve (shown full line on Fig. 8.) gives a lower terminal pressure and temperature than the ideal adiabatic compression curve (shown dotted). This is caused by heat transfer taking place, variable specific heats, a reduction in γ due to gas-air mixing, etc. Resulting compression is not adiabatic and the difference in vertical height is shown as x.

(b) 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 chain dotted) and the curve with initial point $x + y$ lower (shown dotted).

(c) 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 Fig. 8.

In general the assumptions made at the beginning of the section on ideal cycles are worth repeating, *i.e.* 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 Fig. 8.

FIG 8
ACTUAL CYCLES (OTTO BASIS)

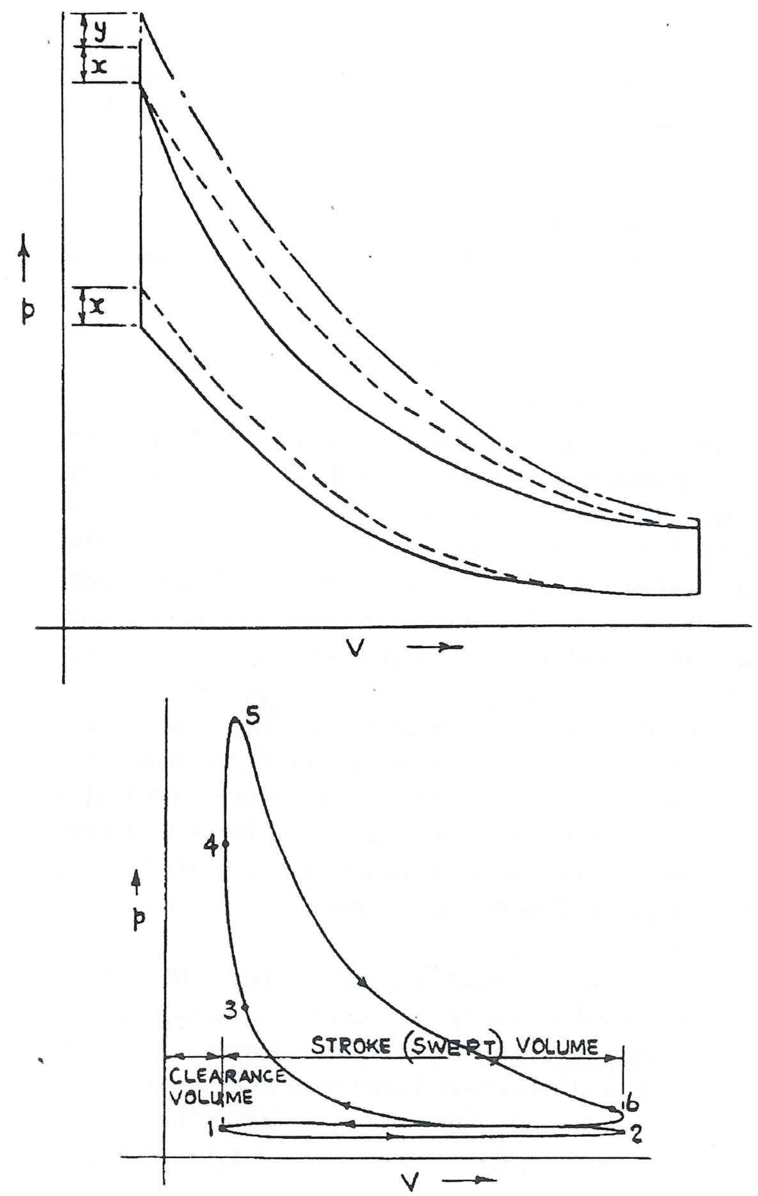
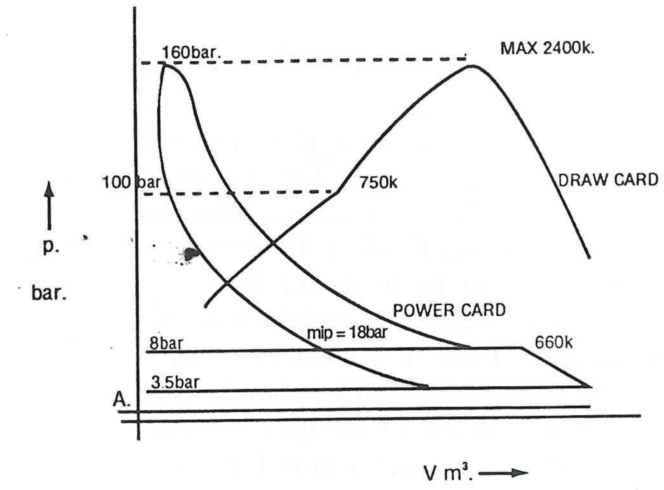
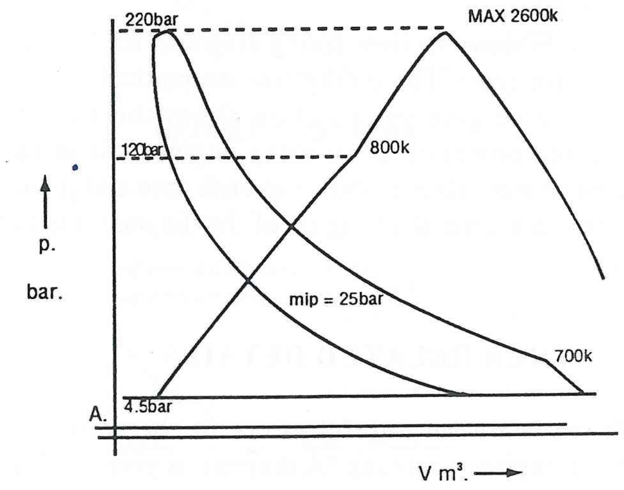


FIG 9

TYPICAL INDICATOR (POWER & DRAW) DIAGRAMS



2 Stroke Cycle (CI)



4 Stroke Cycle (CI)

Typical Indicator Diagrams
The power and draw cards are given on Fig. 9. and should be closely studied. Diagrams given are for compression ignition engines of the 2- and 4-stroke types.

Pressures and temperatures are shown on the sketches where appropriate. The draw card is an extended scale picture of the combustion process. In early marine practice the indicator card was drawn by hand–hence the name. In modern practice an 'out of phase' (90 degrees) cam would be provided adjacent to the general indicator cam. Incorrect combustion details show readily on the draw card. There is no real marked difference between the diagrams for 2-stroke or 4-stroke. In general the compression point on the draw card is more difficult to detect on the 2-stroke as the line is fairly continuous. There is no induction – exhaust loop for the 4-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 given also in Fig. 10; with the fuel shut off expansion and compression should appear as one line. Errors would be due to a time lag in the drive or a faulty indicator cam setting or relative phase difference between camshaft and crankshaft. Normally such diagrams would only be necessary on initial engine trials unless loss of compression or cam shift on the engine was suspected.

Fig. 11. is given to show the light spring diagrams for CI engines of the 2- and 4-stroke types. These diagrams are particularly useful in modern practice to give information about the exhaust – scavenge (induction) processes as so many engines utilise turbo-charge. The turbo-charge effect is shown in each case and it will be observed that there is a general lifting up of the diagram due to the higher pressures.

OTHER RELATED DETAILS

Fuel valve lift cards are very useful to obtain characteristics of injectors when the engine is running. A diagram is given in Fig 12 relating to a Doxford engine.

Typical diagram faults are normally best considered in the

FIG 10
COMPRESSION DIAGRAMS

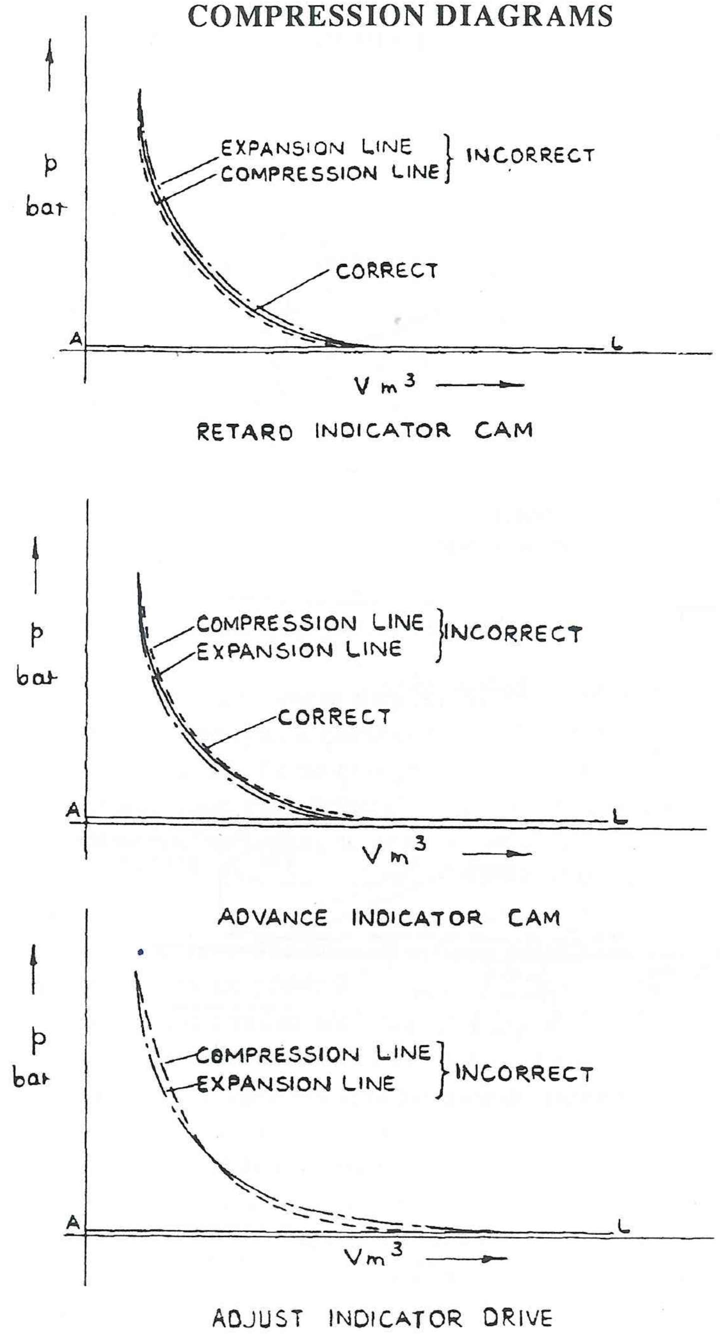
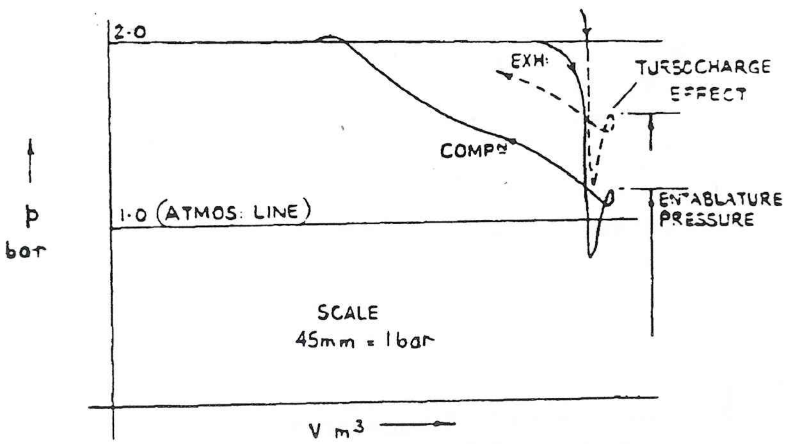
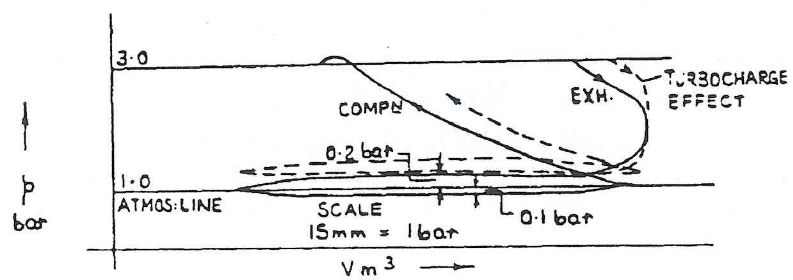


FIG 11
TYPICAL INDICATOR (LIGHT SPRING) DIAGRAMS

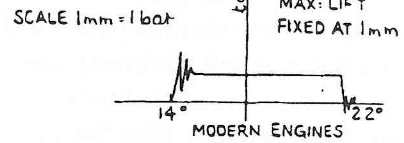


2 STROKE ENGINE (CI)

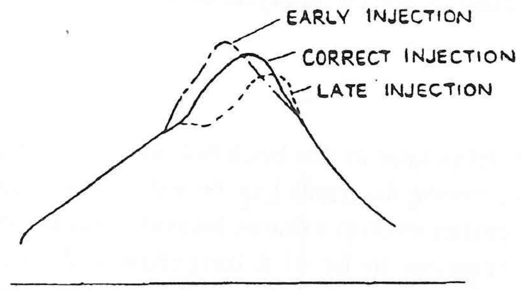


4 STROKE ENGINE (CI)

FIG 12
FUEL VALVE LIFT DIAGRAMS



RELATED DETAILS



TYPICAL FAULTS SHOWN ON DRAW CARD

particular area of study where they are likely to occur. However as an introduction, two typical combustion faults are illustrated on the draw card of Fig. 12. Turbo-charge effects are also shown in Fig. 11. and compression card defects in Fig. 10. It should 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.

Compression ratio has been discussed previously and with SI engines the limits are pre-ignition and detonation. Pinking 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.

Accuracy of indicator diagram calculations is perhaps worthy of specific comment. The area of the power card is quite small and planimeter errors are therefore significant. Multiplication by high spring factors makes errors in evaluation of m.i.p. 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 rather inaccurate nature of the final results is appreciated then the real value of the diagrams can be established. From the power card viewpoint comparison is probably the vital factor and indicator diagrams allow this. However modern practice 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– combustion fault diagnosis and the light spring diagram for the analysis of scavenge – exhaust considerations.

Turbo-charging

This is considered in detail later in this book but one or two specific comments relating to timing diagrams can be made now. Exhaust requires to be much earlier to drop exhaust pressure quickly before air entry and also requires to be of a longer period to allow discharge of the greater gas mass. Air period is usually slightly greater. This could mean for example in the 2-stroke cycle exhaust from 76 degrees before bottom dead centre to 56 degrees after (unsymmetrical by 20 degrees) and scavenge 40 degrees before and after. For the 4-stroke cycle air open as much as 75 degrees before top centre for 290 degrees and exhaust open 45 degrees before bottom centre for 280 degrees, i.e. considerable overlap.

Actual Timing Diagrams

Fig. 13. 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 scavenged engine, however, the piston controls the flow of gas into the exhaust with the result that the opening and closing of these

FIG 13 a
CRANK TIMING DIAGRAM FOR 2-STROKE LOOP SCAVENGED TURBO-CHARGED ENGINE.

EXHAUST & SCAVENGE SYMMETRICAL ABOUT BDC.

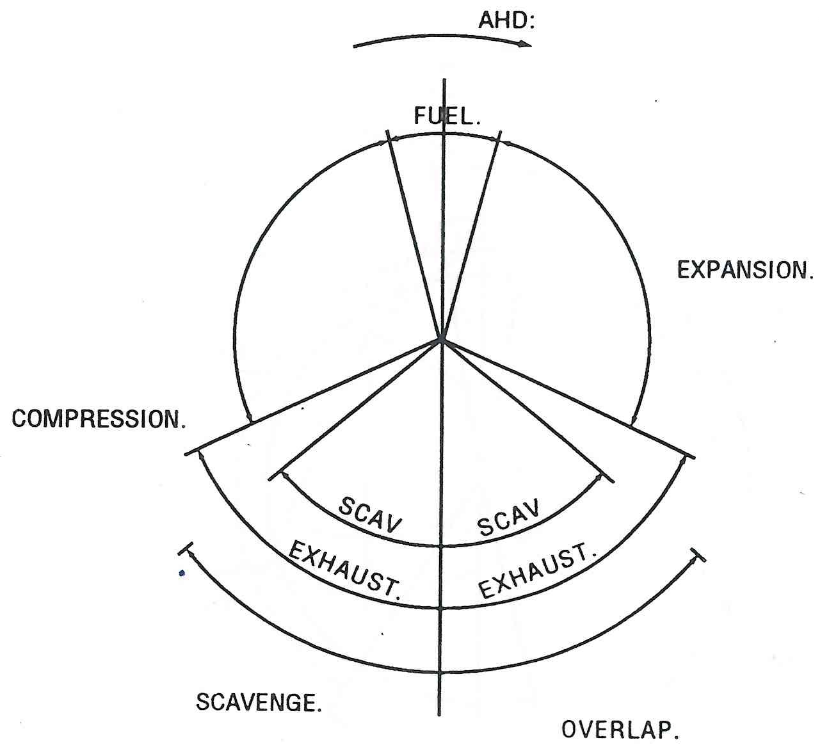


FIG 13 b
4-STROKE NATURALLY ASPIRATED ENGINE.

NOTE THE DIFFERENCE OF OVERLAP BETWEEN TURBO-CHARGED & NATURALLY ASPIRATED 4 STROKE ENGINE.

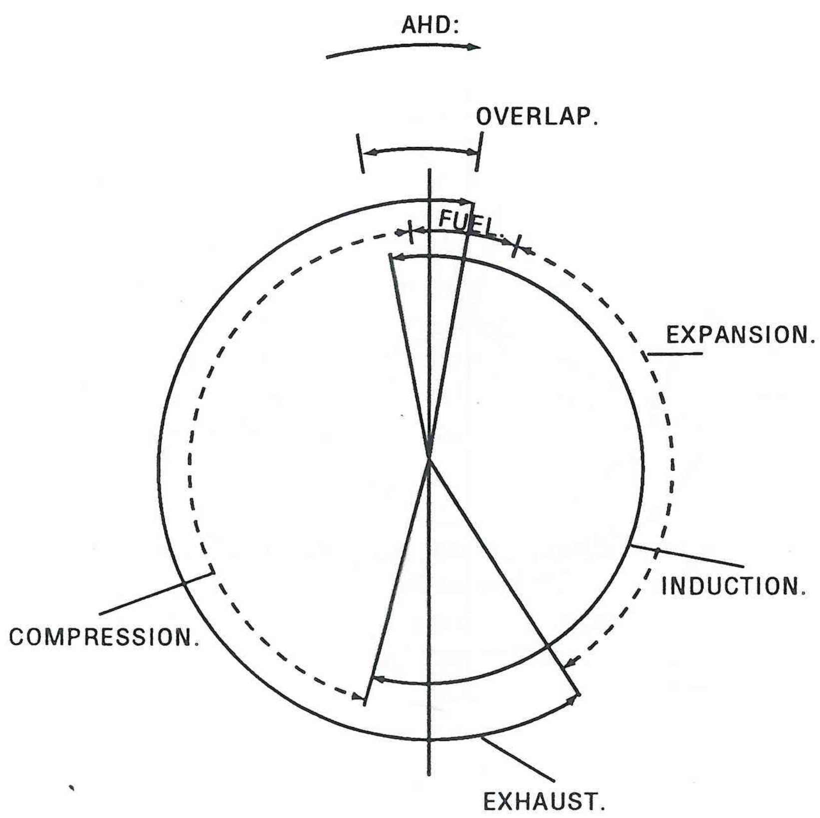


FIG 13 c
4 STROKE TURBOCHARGED ENGINE

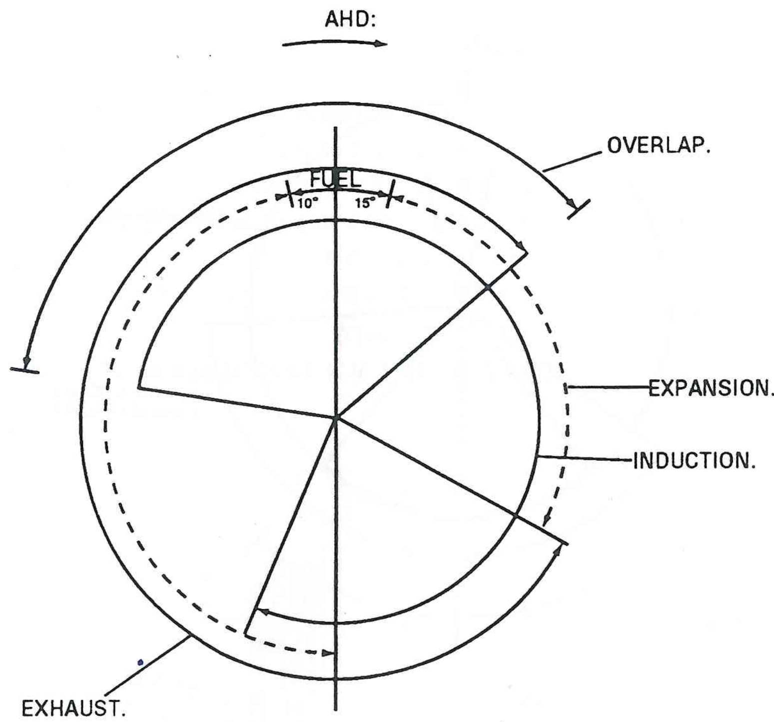
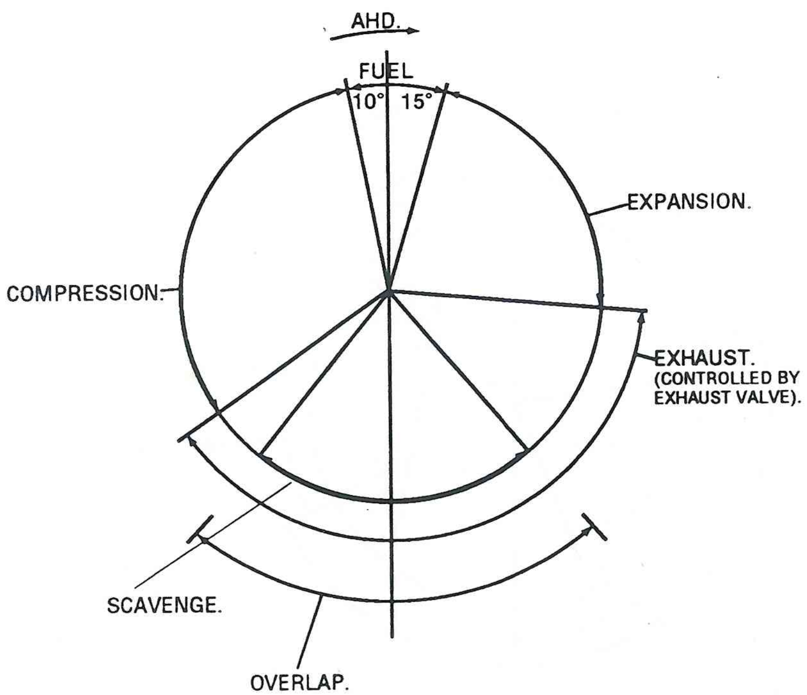


FIG 13 d
CRANK TIMING DIAGRAM FOR 2 STROKE TURBO-CHARGED ENGINE. (UNI-FLOW SCAVENGE. EXHAUST CONTROLLED BY EXHAUST V/V IN CYLINDER COVER).



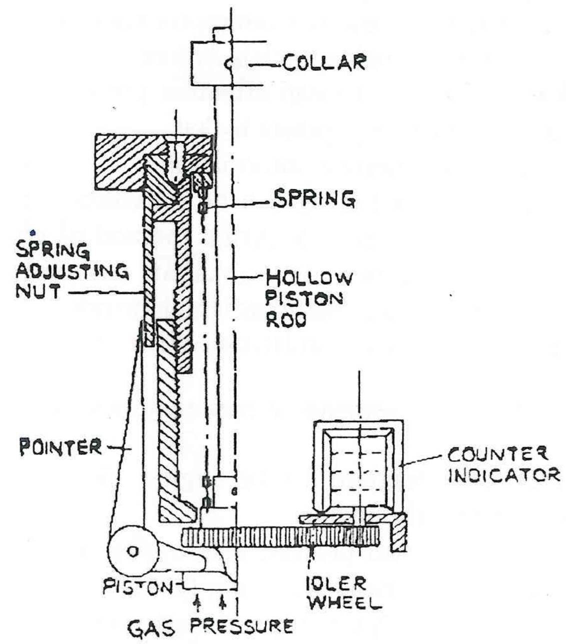
ports are symmetrical about bottom centre. To minimise the losses of charge air to exhaust the choice of exhaust opening position is dictated by the most effective point of exhaust port closure.

Comparison of the crank timing diagrams of the naturally aspirated and turbo-charge 4-stroke diesel engine show 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 compression commencing.

Types of Indicating Equipment

Conventional indicator gear is fairly well known from practice and manufacturers descriptive literature is readily available for precise details. For high speed engines an indicator of the 'Farnboro' type is often used. Maximum and compression pressures can be taken readily using a peak pressure indicator as sketched in Fig. 14.

FIG 14
PRESSURE INDICATOR



Counter and adjuster nut are first adjusted so marks on the body coincide at a given pressure on the counter with idler wheel removed. Idler wheel is now replaced. When connected to indicator cock of the engine the adjusting nut is rotated until vibrations of the pointer are damped out. Spring force and gas pressure are now in equilibrium and pressure can be read off directly on the indicating counter (Driven by toothed wheels).

Electronic Indicating

The limitations of mechanical indicating equipment have become increasingly apparent in recent years as engine powers have risen. With outputs reaching 5500 hp/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 have only limited success. The inaccuracies stem from friction and inertia of mechanical indicator gear and errors in measuring the height of the power card.

Modern practice utilises electronic equipment to monitor and analyse the cylinder peak pressures and piston position and display onto a VDU [video display unit]. The cylinder pressure is measured by a transducer 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 mean effective pressure. Fig. 15. The advantages of this type of equipment is that:

1. It supplies dynamic operational information.

This means that injection timing can be measured while the engine is running. This is a more accurate method of checking injection timing since it allows for crankshaft twist while the engine is under load, unlike static methods which do not.

2. Can compare operating conditions with optimum performance.

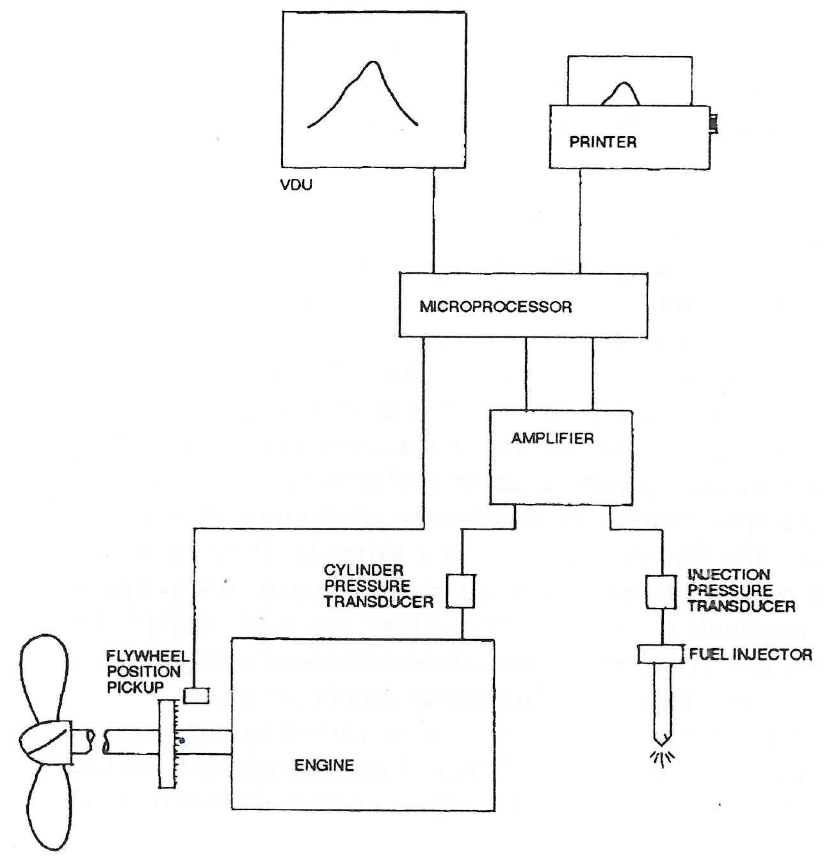
This should lead to improvements in fuel economy and thermal efficiency.

3. Can produce a load diagram for the engine, clearly defining the safe operating zone for the engine.

4. Can produce trace of fuel pressure rise in fuel high pressure lines. Valuable information when diagnosing fuel faults.

Operational experience with this type of equipment has pointed

FIG 15
ELECTRONIC INDICATOR EQUIPMENT



to unreliability of the pressure transducers when connected continuously to the engine. To overcome this problem manufacturers are experimenting with alternative methods of measuring cylinder pressure. One alternative is to permanently attach a strain gauge to one cylinder head stud of each cylinder. Since the strain measured is a function of cylinder pressure this information can be fed to the microprocessor. The increased reliability of this technique will allow the equipment to be permanently installed allowing power readings to be taken at any time. This type of equipment can be used to measure many other engine parameters to aid diagnosis and accurately monitor performance such as fuel pump pressure etc.

Fatigue

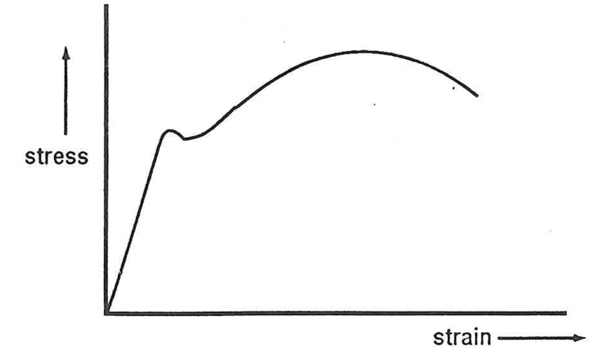
Fatigue is a phenomenon which affects materials that are subjected to cyclic or alternating stresses. Designers will ensure that the stress of a component is below the yield point of the material as measured on the familiar stress/strain graph. However if that component is subjected to cyclic stresses it may fail at a lower value due to fatigue. The most common method of displaying information on fatigue is the S~N curve Fig. 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.

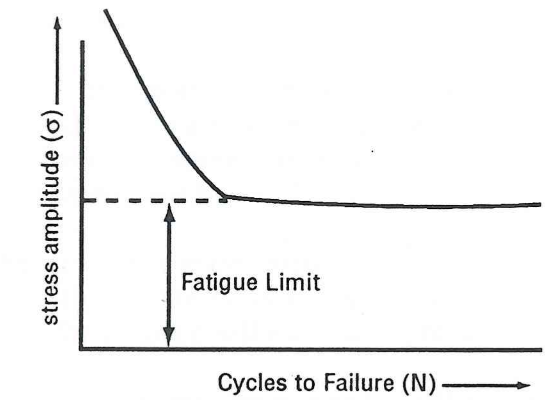
It can be seen from Fig. 16. that, in the case of ferrous materials, there is a point known as the "fatigue limit". Components stressed below this level can withstand a infinite number of stress reversals without failure. Since:

$$\text{Stress} = \frac{\text{load}}{\text{CSA}}$$

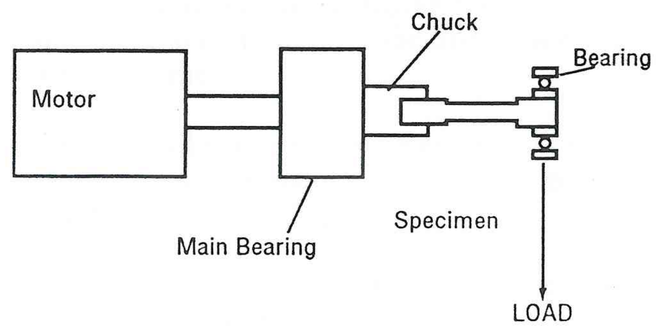
It can be seen that reducing the stress level on a component involves increasing the CSA [cross sectional area] resulting in a weight penalty. In marine practice the weight implications are, in general, secondary to reliability and long-life and so components are usually stressed below the fatigue limit. This is not the case in.

FIG 16
WOHLER MACHINE FOR ZERO MEAN STRESS FATIGUE TESTING





Typical S/N curve FOR FERROUS MATERIAL



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. The working life of 4-stroke medium speed diesel bottom end bolts are calculated in this way.

CHAPTER 2

STRUCTURE AND TRANSMISSION

The engine structure consisting of the bedplate and "A" frames or, in more modern designs the frame section must fulfil the following fundamental requirements and properties.

Strength – is necessary since considerable forces can be exerted. These may be due to out of balance effects, vibrations, gas force transmission 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, it may enable the power weight ratio to be increased. Less material would be used, bringing about a saving in cost. Both are important selling points as they would give increased cargo capacity.

Toughness – in a material is a measure of its resilience and strength, this property is required to enable the material to withstand the fatigue conditions which prevail.

Simple design – if manufacture and installation are simplified then a saving in cost will be realised.

Access – ease of access to the engine transmission system for inspection and maintenance, and in the first instance 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.

Seal – the transmission system container must seal off effectively the oil and vapours from the engine room.

Manufacture – Modern engines increasingly are manufactured in larger modular sections that allow for convenience in assembly.

BEDPLATE

This is a structure that may be made of cast iron, prefabricated steel, cast steel, or a hybrid arrangement of cast steel and prefabricated steel.

Cast iron one piece structures are generally confined to the smaller engines. That is, medium speed engines rather than the larger slow speed cross-head type of engine. This is due to the problems that arise as the size of the casting increases. These problems include poor flow of material to the extremities of the mould, poor grain size control 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 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. Castings do not require heat treatment, cast iron is easily machined, it 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, 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 closed with a light unstressed oil tray. Fig. 17.

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. Fig. 18.

FIGURE 17

SECTION THROUGH ENGINE BLOCK OF MEDIUM SPEED ENGINE WITH UNDERSLUNG CRANKSHAFT.

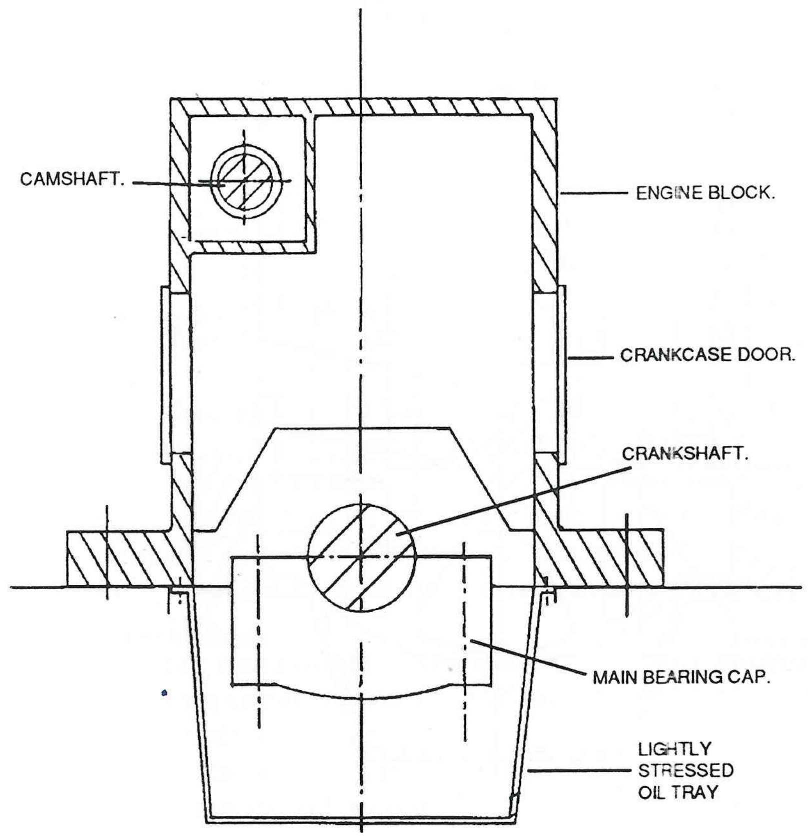
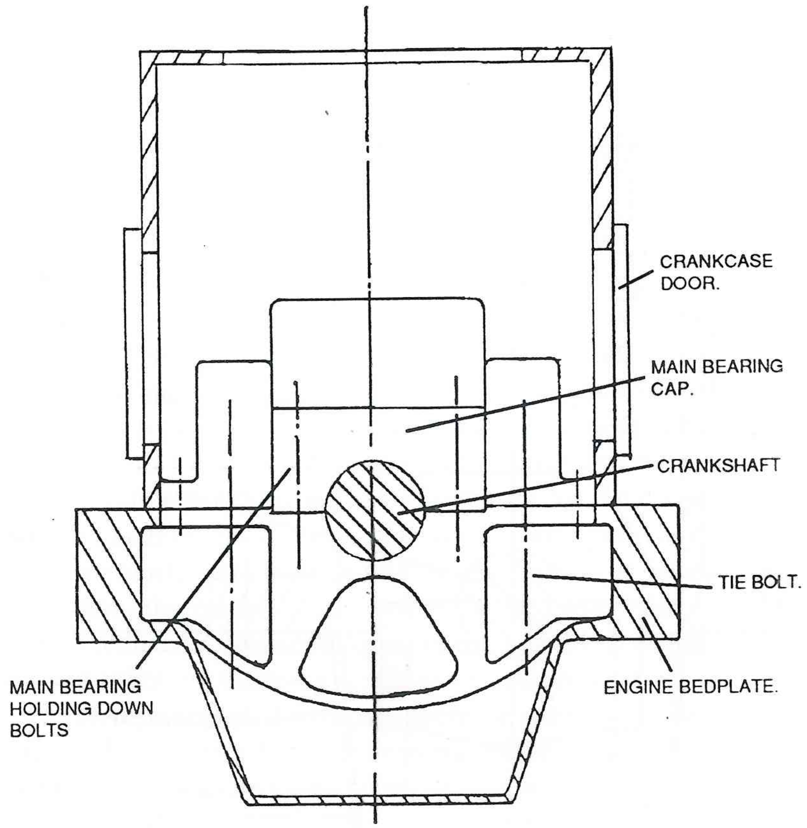


FIGURE 18
MEDIUM SPEED ENGINE BEDPLATE WITH EMBEDDED CRANKSHAFT



When welding techniques and methods of inspection improved and larger furnaces became available for annealing, the switch to prefabricated steel structure with its saving in weight and cost was made. 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.

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 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 Fig. 19. To increase the torsional, longitudinal and lateral rigidity of the structure suitable webbing is incorporated into the fabrication.

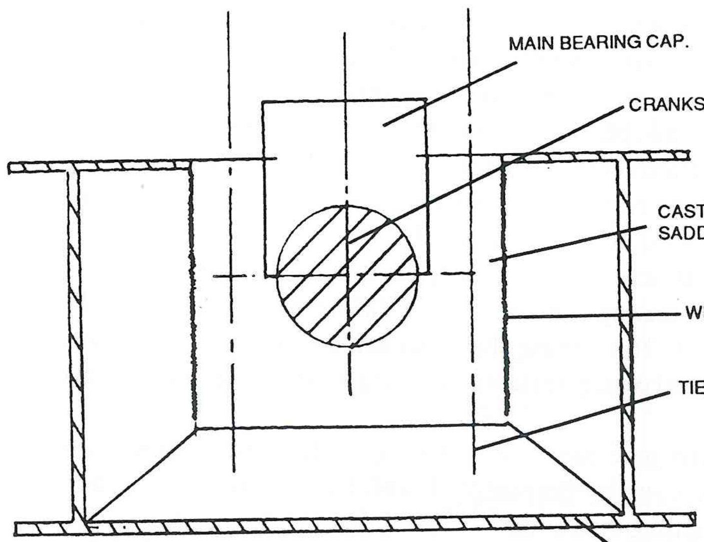
It is modern practice to cut the steel plate using automatic contour flame cutting equipment. Careful preparation is essential prior to the welding operation:

- Since it is necessary to prepare the edges of the cut plate it is necessary to make an allowance for this when cutting.
- Equipment is set correctly to ensure smallest heat affected zone, [HAZ].
- Welding consumables stored and used correctly to prevent hydrogen contamination of HAZ which could lead to post anneal 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. Flaws in welds would be cut out, rewelded 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 cool the structure is shot blasted and the welds again tested before the bedplate is machined.

FIGURE 19
MODERN FABRICATED SINGLE WALLED BEDPLATE WITH CAST STEEL BEARING SADDLE.



In order to minimise stresses due to bending in the bedplate, without a commensurate increase in material, tie-rods are used to transmit 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. They are pre-stressed at assembly so that the engine structure is under compression at all times. Engines utilising the opposed piston principle have the combustion loads absorbed by the running gear and do not require to be fitted with tie-rods. To minimise bending tie-rods are placed as close as possible to to the shaft centre line.

Fig. 20 shows diagrammatically the arrangement used in the Sulzer engine. By employing jack bolts, under compression, to

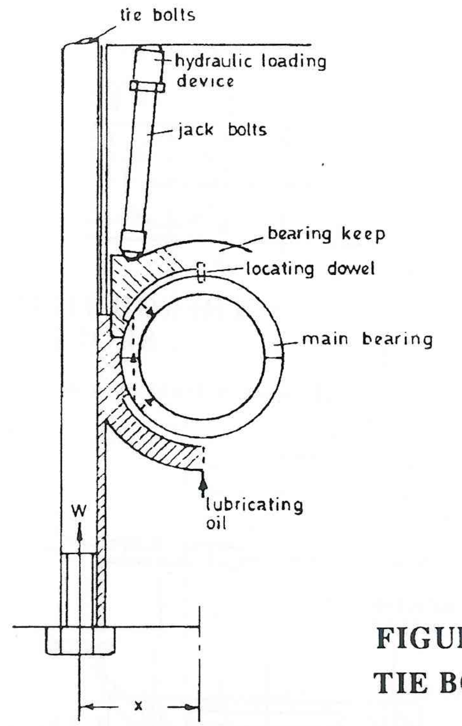


FIGURE 20
TIE BOLTS

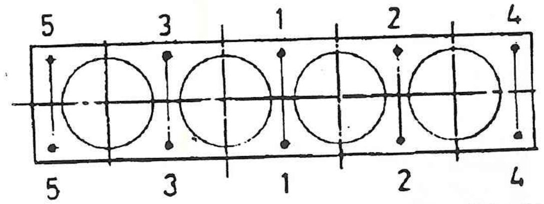
retain the bearing keeps in position the distance x is kept to a minimum. Hence the bending moment Wx where W is the load in the bolt, is also a minimum.

Because of 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.

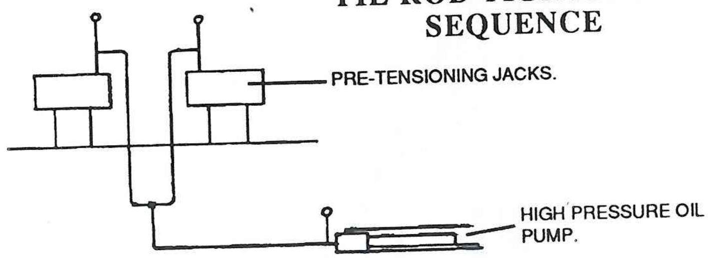
Although tie-rods are tightened, to their correct pretension during assembly, they should be checked at intervals. This is accomplished by:

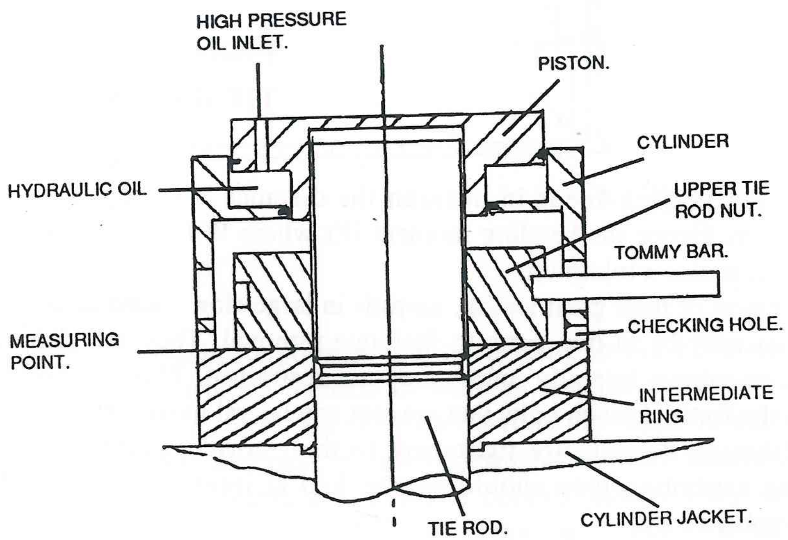
- Connecting both pre-tensioning jacks to two tie-rods lying opposite each other. Fig. 21.
- Operating the pump until the correct hydraulic pressure is reached. This pressure is maintained.
- Checking the clearance between the nut and intermediate ring with a feeler gauge. If any clearance exists then the nut is tightened onto the intermediate ring and the pressure

FIGURE 21



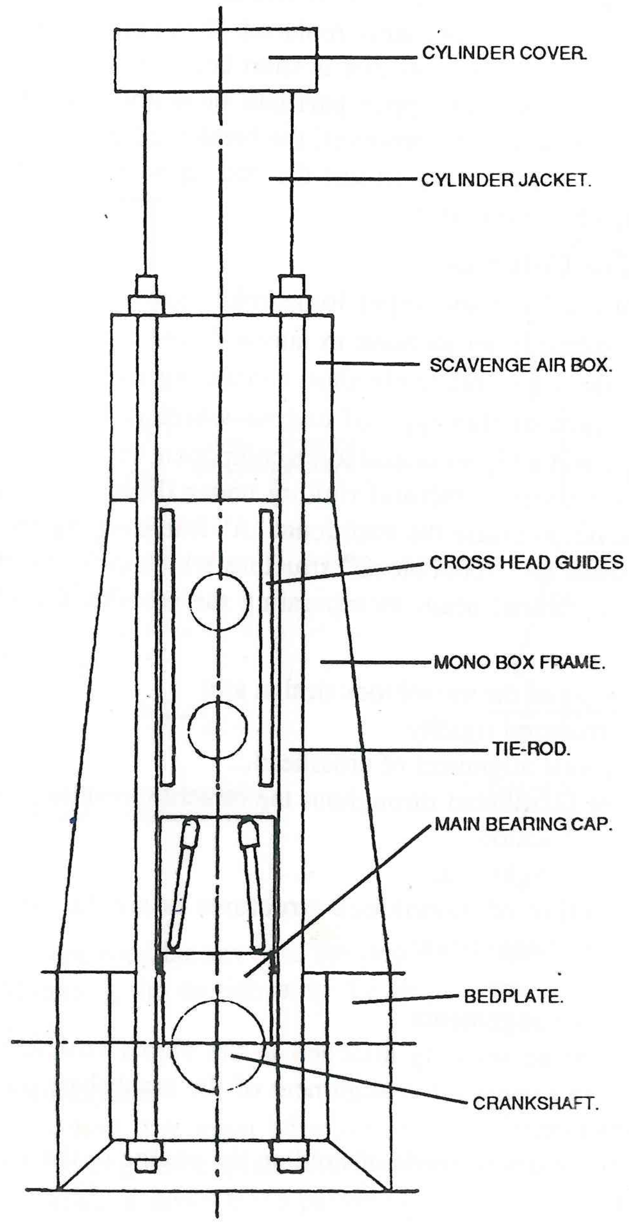
TIE ROD TIGHTENING SEQUENCE





PRE-TENSION JACK

FIGURE 22
GENERAL ARRANGEMENT OF ENGINE STRUCTURE



released. If no clearance is found the pressure can be released and the hydraulic jacks removed.

When using hydraulic tensioning equipment it is essential that it is maintained in good order and the accuracy of the pressure gauges are checked regularly.

If when inspecting the engine it is found that a tie-rod has broken then 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 it is necessary to cut the rod to be removed in sections through the crankcase.

"A" Frames or Columns

The advent of the long and super-longstroke slow-speed diesel engines has resulted in an increase in lateral forces on the guide. This is due to the use of relatively short connecting-rods to reduce the overall height of this type of engine which results in an increased angle and a higher lateral force component Fig. 23.

In order to maintain structural rigidity under these conditions designers tend not to utilise the traditional "A" frame arrangement, preferring instead the "monoblock" structure which consists of a continuous longitudinal beam incorporating the crosshead guides Fig. 24.

The advantages of the monoblock design are:

- Greater structural rigidity.
- More accurate alignment of crosshead.
- Forces are distributed throughout the structure resulting in a lighter construction.
- Improved oil tightness.

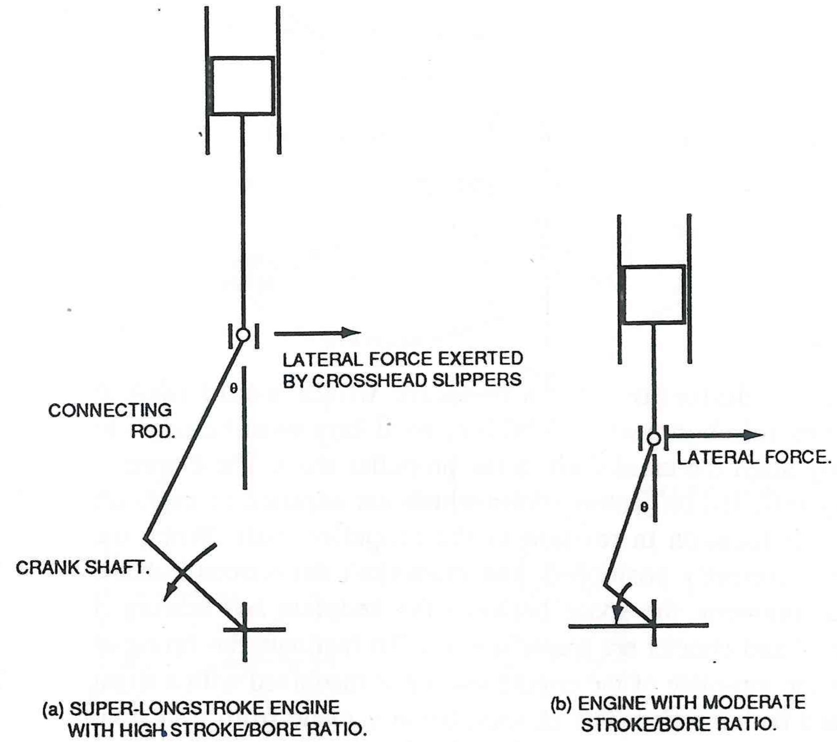
The construction of monoblock structures is similar to that described for bedplates above.

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.

FIG 23
INCREASED CONNECTING ROD ANGLE GIVING HIGHER LATERAL FORCES.



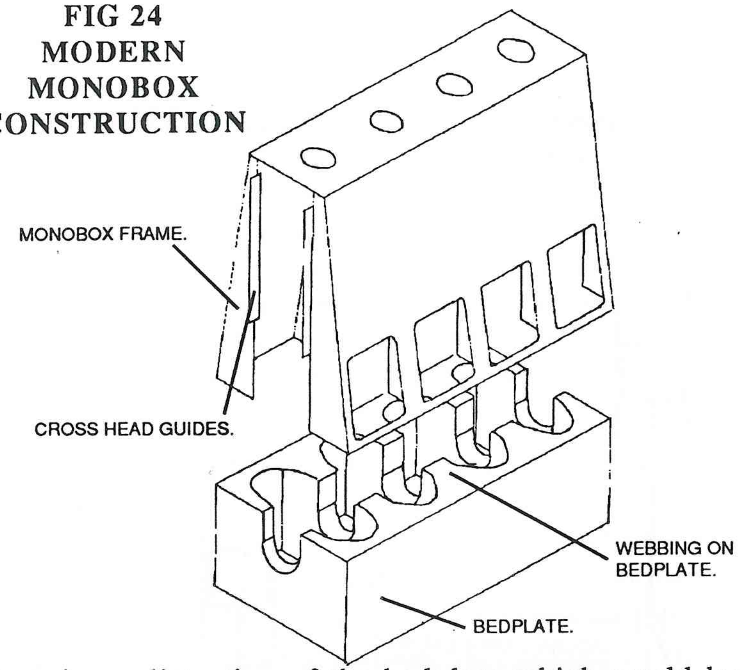
1. By rigid foundations onto the ship's structure.

2. Mounting the engine onto the ship's structure via resilient mountings.

Rigid foundations

In this method, the most common, fitted chocks are installed between the engine bedplate and the engine seating on the tank-top. The holding down bolts passing through the chocks. During installation of the engine great care must be taken to ensure that

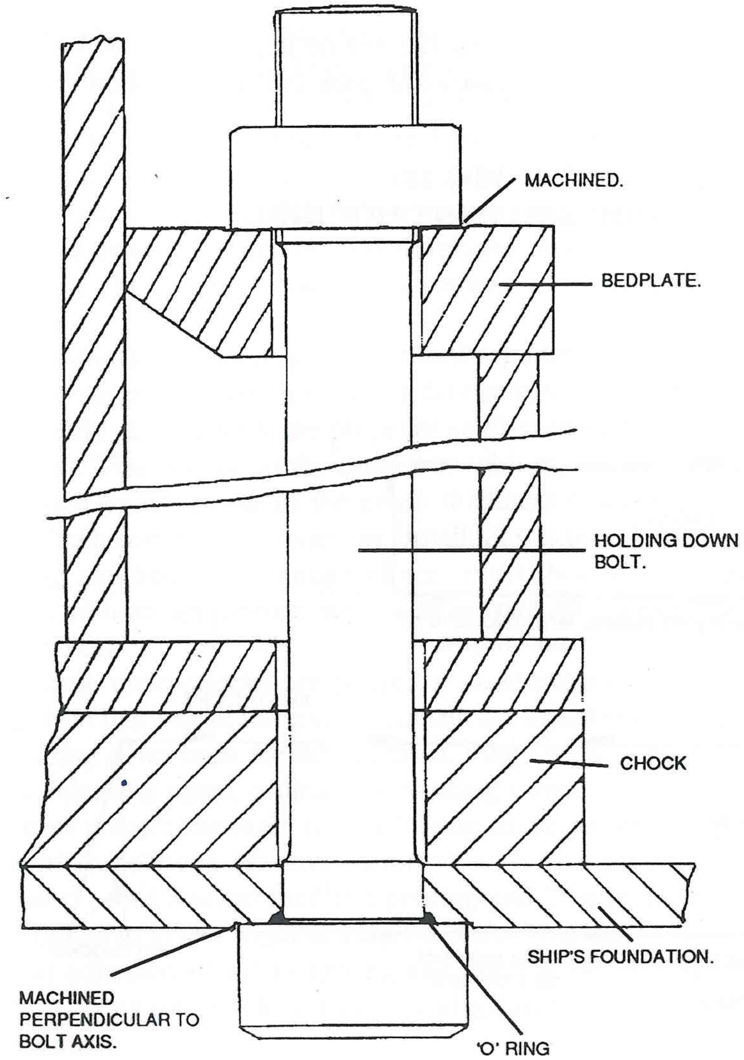
FIG 24
MODERN
MONOBOX
CONSTRUCTION



there is 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 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 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 Fig. 25. This type of bolt, because of its greater length, has greater elasticity and is therefore less prone to cracking than the superceded short unsleeved bolt. The bolts are installed through the

FIG 25
LONG SLEEVED HOLDING DOWN BOLT

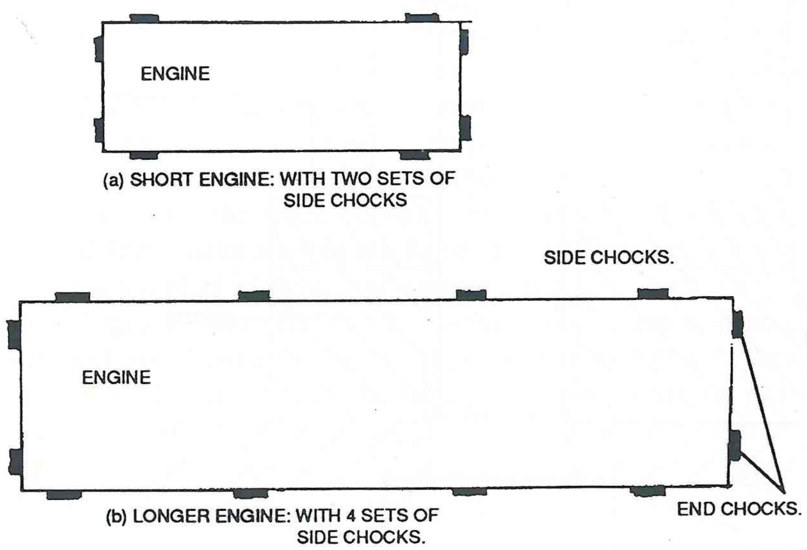


top plate and a waterproof seal is usually effected with "O" rings. Fitted bolts are installed adjacent to the engine thrust.

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 depends upon the length of the engine Fig 26.

It is extremely important that the engine is properly installed during building. The consequences of poor initial installation are

FIG 26
SIDE AND END CHOCKING



extremely serious since it may lead to fretting of chocks, the foundation and bedplate, slackening and breakage of holding down bolts and ultimately in a worsening in the alignment 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.

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 seatings and bedplate. It is claimed that the time taken to accomplish such a repair is reduced so reducing the overall costs. Although initially developed as a repair technique the use of epoxy resin chocks is becoming 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. This is done in the usual way with the exception that it is set high by about 1/1000 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.

Because 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 being sprayed with a releasing agent, ordinary bolts being coated with a silicone grease to prevent the resin from adhering to the metal. The outer sides of the chocks are now dammed 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.

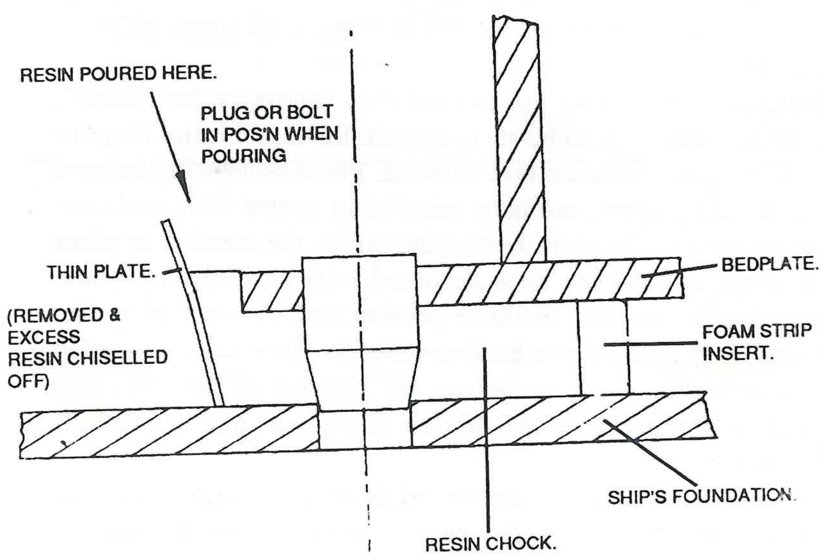
The resin and activator are mixed thoroughly with equipment that does not entrain air. The resin is poured directly into the dammed off sections. Curing will take place in about 18 hrs if the temperature of the chocking area is maintained at about 20 to

25°C. The curing time can be up to 48 hours 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.
- Lower bolt tension by a factor of 4 when compared to metal chocks.
- Elimination of misalignment due to fretting and bolt slackening. Because of the intimate fit of resin chocks and the high coefficient of friction between resin and steel the thrust forces are distributed to all chocks and bolts thus reducing the total stress on fitted bolts by about half Fig 27.

FIG 27
POURED RESIN CHOCKS

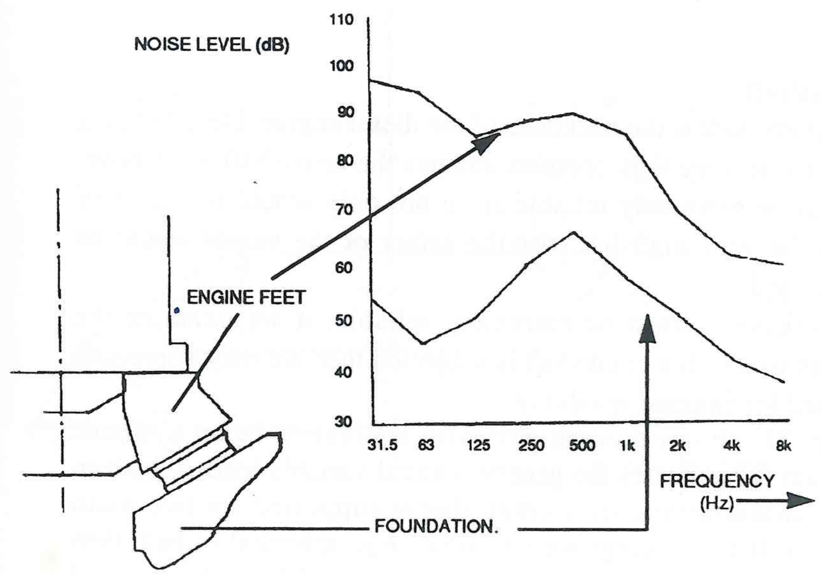


Resilient mountings

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

Diesel engines generate low frequency vibration and high frequency structure borne noise. The adoption of resilient mountings will successfully reduce both noise and vibration.
The reduction of noise and vibration of resilient and non resiliently mounted engines can be seen in Fig. 28.

FIG 28
REDUCTION IN STRUCTURE BORNE NOISE ACHIEVED BY RESILIENT MOUNTINGS.



In Fig. 29 it can be seen that 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, to a sub-frame which is resiliently mounted to the ship's structure. The engine is then coupling to the reduction gearbox through a highly elastic coupling. It is necessary to limit the amount of lateral and longitudinal movement of the engine, relative to the ship's structure. This is accomplished by stopper devices built into the holding down arrangement. happen, meet

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 locking device. This device, shown in Fig. 30, has a piston with a connection via a shut off valve between both sides. During normal running the connecting valve is open allowing the piston to displace oil between upper and lower chamber freely. During starting and stopping this valve will be closed effectively preventing relative movement between engine and ship's structure.

Crankshafts

A crankshaft is the backbone of the diesel engine. Despite being subjected to very high complex stresses the crankshaft must none-the-less be extremely reliable since not only would the costs of failure be very high but, also,the safety of the vessel would be jeopardised

Crankshafts must be extremely reliable, if we examine the stresses to which a crankshaft is subjected then we may appreciate the need for extreme reliability.

Fig. 31. shows a crank unit with equivalent beam systems. Diagram (a) indicates the general, central variable loaded, built-in beam characteristic of a crank throw supported by two main bearings. If the bearings were flexible. *e.g.* spherical or ball, then a simply supported beam equivalent would be the overall characteristic.

Examining the crank throw in greater detail, diagram (b), shows that the crank pin itself is like a built-in beam with a distributed

FIG 29
SUB-FRAME TYPE RESILIENT ENGINE MOUNTING

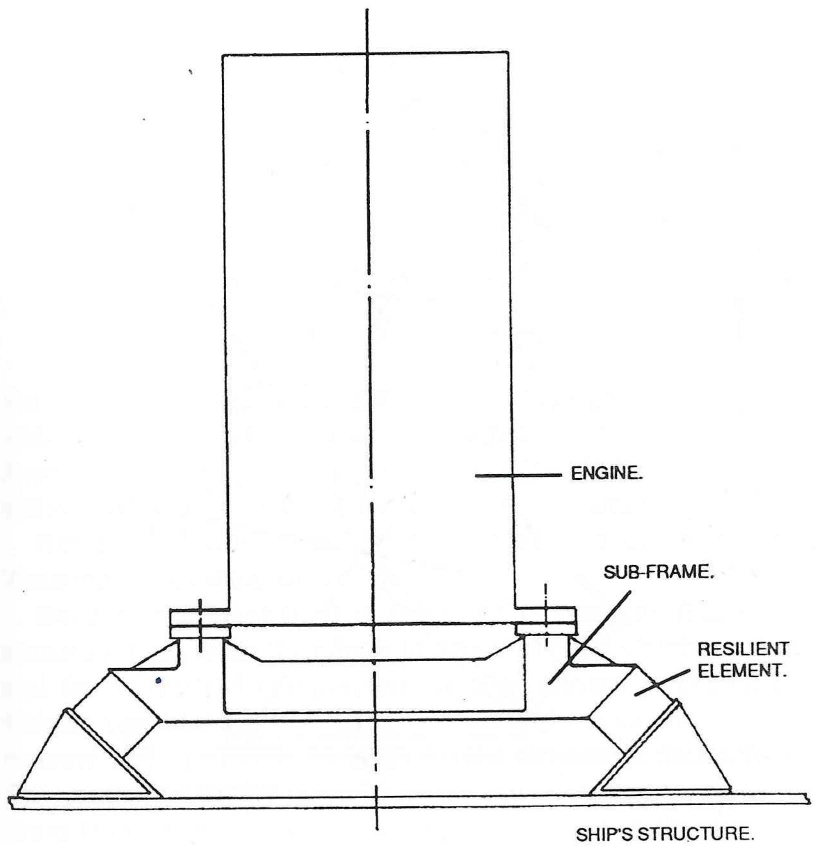


FIG 30
HYDRAULIC LOCKING DEVICE FOR ENGINE MOVEMENT LIMITATION DURING STARTING/STOPPING

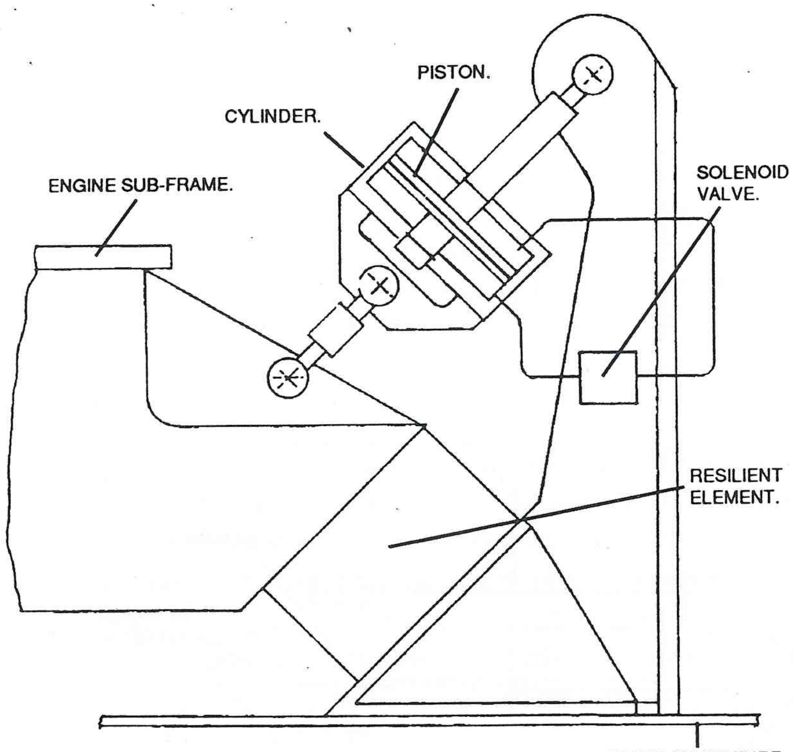
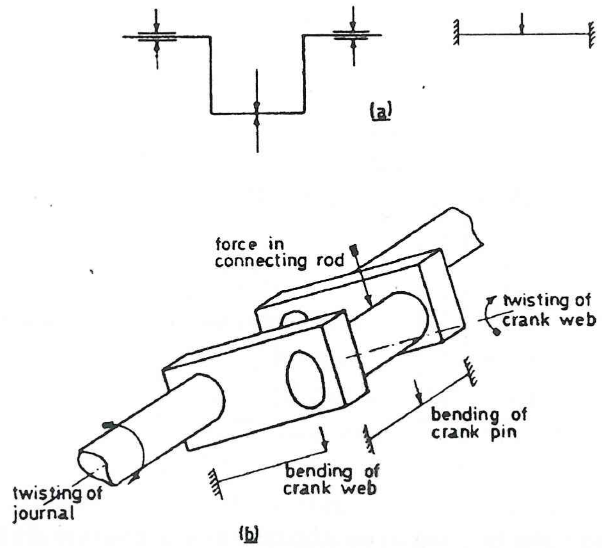


FIG 31
STRESSES IN CRANKSHAFT



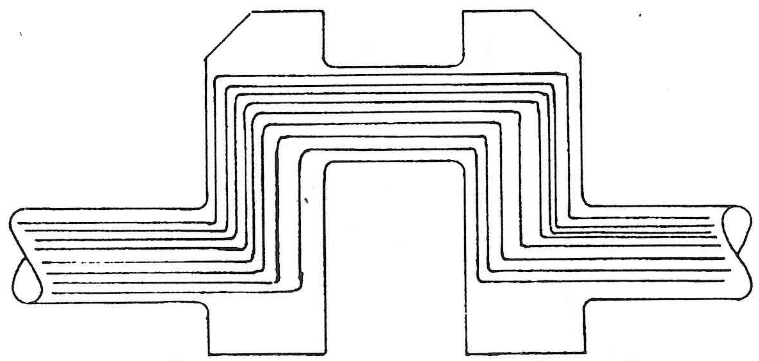
load along its length that varies with crank position. Each crank web is like a cantilever beam subjected to bending and twisting. Journals would be principally subjected to twisting, but a bending stress must also be present if we refer back to diagram (a).

Bending causes tensile, compressive and shear stresses. Twisting causes shear stress.

Because the crankshaft is subjected to complex fluctuating stresses it must resist the effects of fatigue. To this end the material and the method of manufacture must be chosen carefully. For fatigue considerations forging is preferable to casting. This is because, unlike casting, forgings exhibit directional "grain flow". The properties of the material in the direction transverse to the grain flow being significantly inferior to those in the direction longitudinal to the grain flow. Under these circumstances the drop in fatigue strength may be as must as 25 to 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. Fig. 32.

The materials chosen for forged and cast crankshafts are essentially the same. The composition of the steel will vary

FIG 32
DIRECTION OF GRAIN FLOW IN FORGED CRANKSHAFT.



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 UTS of approximately 425 to 435 MN/m². For higher output applications with harder bearing materials the carbon content is in the range of 0.35% to 0.4% which raises the UTS to approximately 700 MN/m². 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 increased by nitriding.

There are two broad categories of crankshafts:

1. One piece construction.
2. "Built up" from component parts.

1. 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 reduce the micro-defects on the surface.

Following machining the crankshaft is then tested for surface and sub-surfaced defects.

2. Built up crankshafts

There are 3 categories of built up crankshafts:

Fully built up; webs are shrunk onto journals and crankpins Fig 33a.

Semi-built up; webs and crankpin as one unit shrunk onto the journals Fig. 33b.

Welded construction; webs, journals and crankpin are welded together. Fig. 33c.

Fully and semi-built up construction

To minimise the risk of distortion of fully and semi built crankshafts, assembly is carried out vertically. Various jigs are required to ensure the correct crank angles and to provide support for the crankshaft. The webs are heated only to about 400°C. and the journals and pins inserted. Raising the temperature higher would bring the steel to the critical temperature and change the material's characteristics. When the assembly has cooled the web material adjacent to the journal will be in tension. The level of stress in this region must be well below the limit of proportionality to ensure that the material does not yield which would reduce the force of the web on the journal and lead to fretting and probable slippage. 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 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 cold rolled Fig. 34a. 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. Fig. 33.

The fully built up crankshaft has generally been superceded by the semi-built up type which display improved "grain flow" in webs and crankpin, are stiffer and can be shorter due to a reduction in the thickness of the webs.

FIG 33
3 TYPES OF BUILT-UP CRANKSHAFT

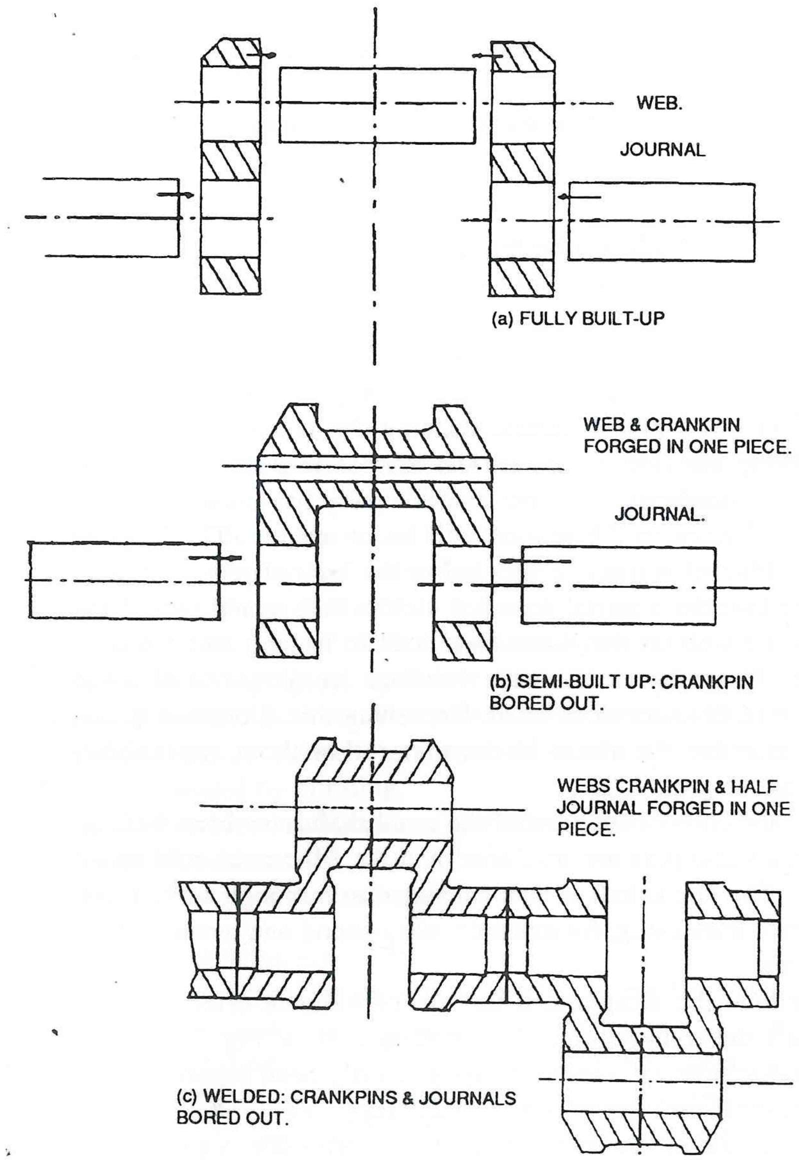
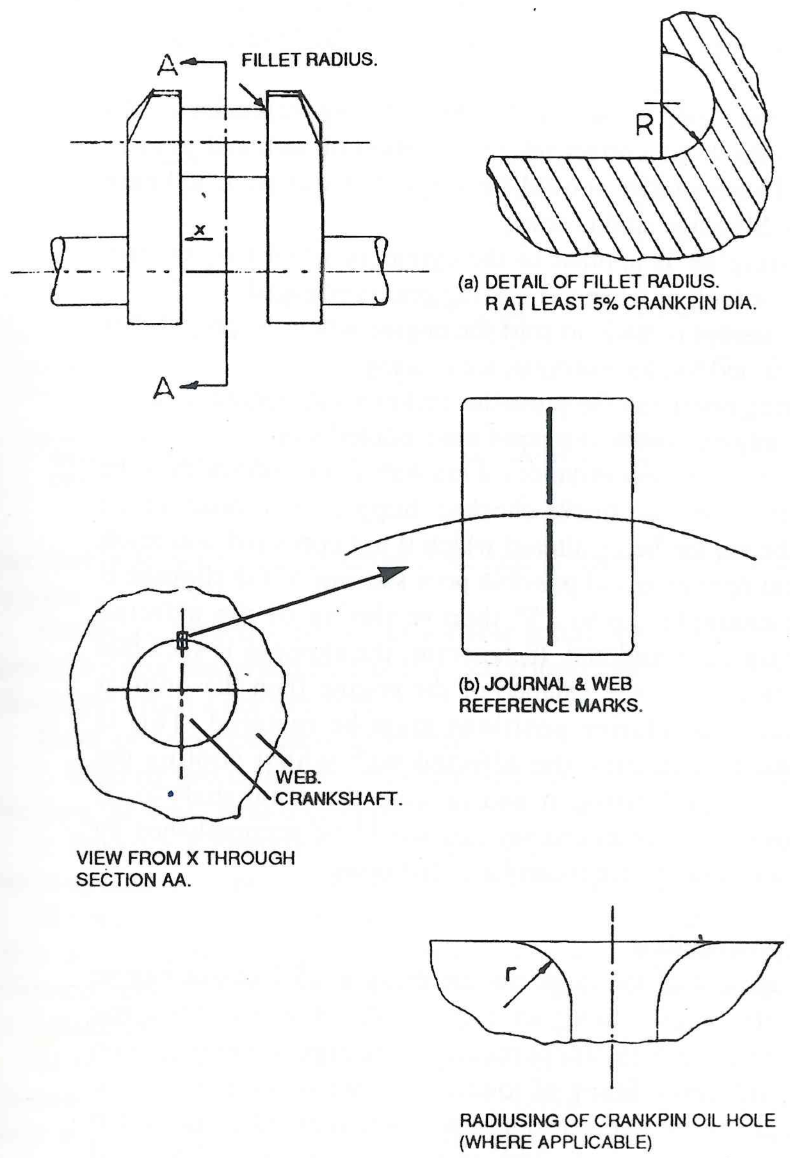


FIG 34
DETAILS OF CRANKSHAFT



The effectiveness of the grip due to shrinking depends upon:

1. The shrinkage allowance. The correct allowance will result in 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 up by shrink fitting, reference marks are made to show the correct relative position of web and journal. Fig. 34b. These marks should be inspected during crankcase inspections. Slippage could occur:

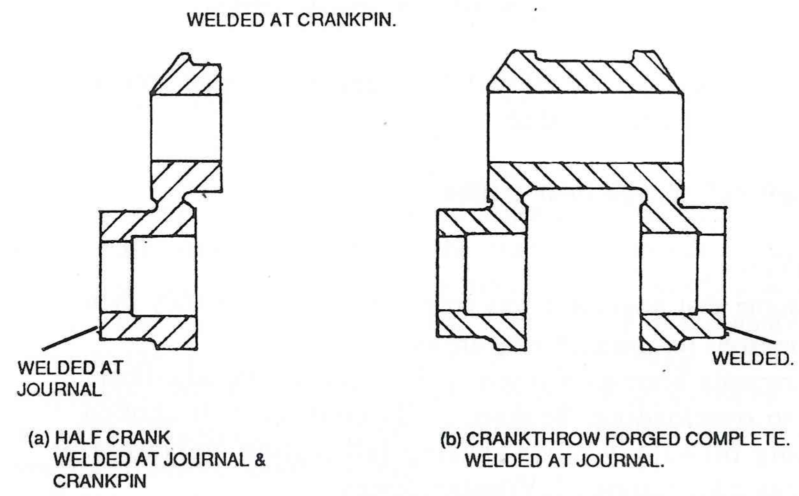
- If starting air is applied to the cylinders when they contain water or fuel, or when the turning gear is engaged.
- If an attempt is made to start the engine when the propeller is constrained by, for example, ice or a log.
- If during operation the propeller strikes a submerged object.
- If the engine 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 result in inefficient operation and possible poor starting. 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 whilst cooling the journal with liquid nitrogen and jacking the crankshaft to its original position. Needless-to-say this would be accomplished by specialist personnel under controlled conditions.

Welded Construction

The development of the large marine cross head 2-stroke engine will undoubtedly result in higher outputs without an accompanying increase in physical size. These requirements impose limitations on the traditional shrink fitting of journals and webs. To transmit the torques required the traditional shrink fitting method requires that the web is of a minimum width and radial thickness. This will

FIG 35
TWO OPTIONS OF WELDED CRANKSHAFTS



inevitably lead to a larger crankshaft and consequently a larger engine.

Welded construction is seen as a viable solution to this problem and one major manufacturer has invested considerable resources in developing such a construction.

There are two methods of assembly:

1. Welding two crankarms together then making a crankshaft by welding the crankarms together Fig. 35a.

2. Forging a crankthrow complete with half journals then welding them with others to form the crankshaft Fig. 35b.

The welding technique chosen is submerged arc narrow gap. This technique is automated and produces a relatively small heat affected zone [HAZ], 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 rewelded.

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 possible reasons for misalignment are as follows:

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

Vibration

This can be caused by: incorrect power balance, prolonged running at or near critical speeds, slipped crank webs on journals, light ship conditions leading to impulsive forces from the propeller (e.g. forcing frequency four times the revs. for a four-bladed propeller), the near presence of running machinery, excessive wear down of the propeller shaft bearing (this in bad weather conditions can lead to whipping of the shafting).

Vibration accentuated stresses, they can be increased to exceed fatigue limits and considerable damage could result. It can lead to things working loose, *e.g* coupling bolts, bearing bolts, bolts securing balance masses to crank webs and lubricating oil pipes.

Other causes

Incorrect manufacture leading to defects is fortunately a rare occurrence. In the past, failure has been caused by: slag inclusions, heat treatment and machining defects, for example badly radiused oil holes and fillets. Careless use of tools resulting in impact marks on crankpins and journals can also lead to failure. These defects all result in the the creation of stress concentrations which, because of cyclic nature of the loading of the crankshaft can raise the local level of stress in the component above the level of the fatigue limit on the S ~N graph, Fig. 16. Chapter 1, resulting in fatigue cracking and ultimate failure. This can be exacerbated if the engine is run at or close to the critical speed. The critical speed of an engine is the crankshaft speed which causes the crankshaft to vibrate at its natural frequency of torsional vibration. In other words it is the speed which induces resonance. The consequence of resonance 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 4-stroke diesel engines are subjected to fluctuating cyclic stresses and are therefore also exposed to potential fatigue failure. 4-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 top dead centre on the exhaust stroke. The forces experience by bottom end bolts in this situation is high. Reference to the S ~N graph in chapter 1 will show that to ensure 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 4-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 "waisted" design with generous radii.
- Have increased diameter at mid shank to reduce vibration.
- Be tightened accurately to the required level.

During maintenance bolts should be examined for mechanical damage which would cause a stress concentration. Damaged bolts should be replaced.

Fretting Corrosion

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 of the metal also effects the attack, in general damage to ferrous surfaces is found to decrease as hardness increases.

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, this raises bearing temperature so that lead rises to the surface and is removed. The process is repeated until failure of the bearing takes place. Scoring of crankshaft pins can then occur. Use of detergent types of lubricating oil can prevent or minimise this type of corrosion. The additives used in the oil to give it detergent properties would be alkaline, in order to neutralise the weak acids.

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.

Bearing Clearances and Shaft Misalignment

Bearing clearances can be 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 can be achieved, this can prove to be very difficult in the larger types of engine. Finally, the use of lead wire necessitating the removal of the bearing keeps.

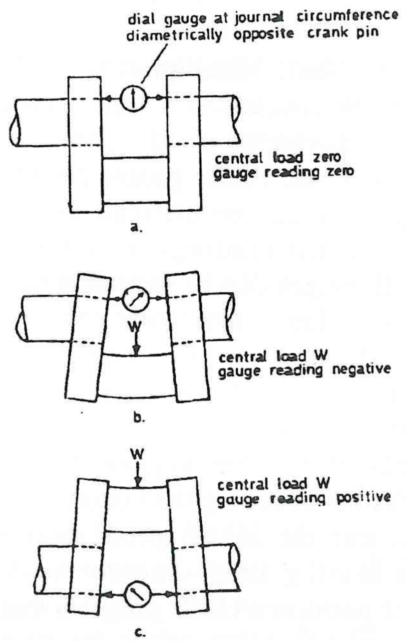
Main bearing clearances, should be zero at the bottom. If they are not, then the crankshaft is out of alignment. Some engines are provided with facilities for obtaining the bottom clearance (if any) of the main bearings with the aid of special feelers, without the need to remove the bearing keep. Another 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 high by comparison and 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 crank pin and ideally at the circumference of the main journal (see Fig. 36.) will give a horizontal deflection, when the crank is rotated through one revolution, that is directly proportional to the vertical deflection.

In Fig. 36(a). it is assumed that main bearings are in correct alignment and no central load is acting due to running gear, then

FIG 36
CHECKING CRANKSHAFT ALIGNMENT



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. 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 diametrically opposite crank pin centre when the crank is on bottom centre an average of two readings would be taken, one either side during the turning of the crank.

The following table shows some possible results from a six cylinder diesel engine:

GAUGE READINGS IN mm/100

	CYLINDER NUMBER					
CRANK POSITION	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 mis-alignment (t-b)	11	2	13	-14	-8	5
Horizontal mis-alignment (p-s)	0	-1	0	0	3	-2

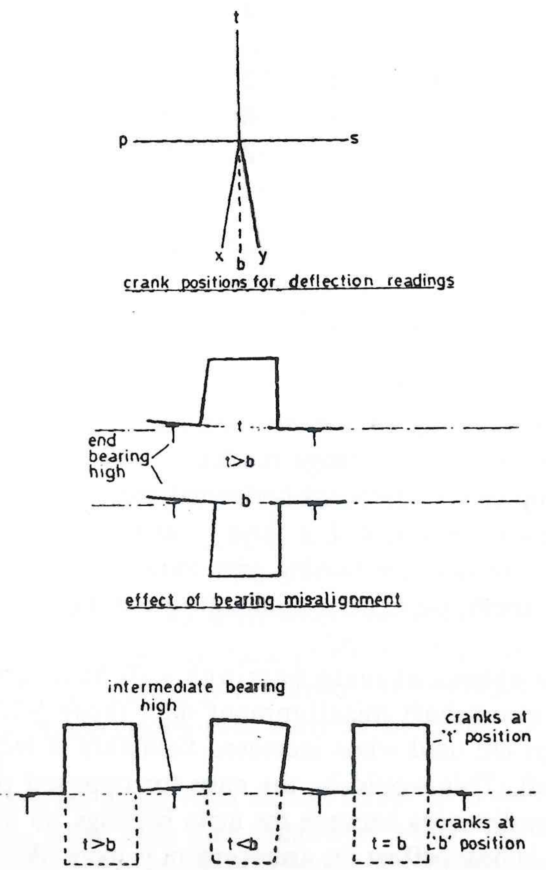
The dial gauge would be set at zero when crank is in, say, port side near bottom position and gauge readings would be taken at port horizontal, top centre, starboard horizontal and starboard side near bottom positions. Say *x, p, t, s* and *y* as per Fig. 37., but before taking each reading the turning gear should be reversed to unload the gear teeth, otherwise misleading readings may be obtained.

Engines with spherical main bearings will have greater allowances for crankshaft misalignment than those without. Spherical bearings are used when increased flexibility is required for the crankshaft. This would be the case for opposed piston engines with large distances between the main bearings, so instead of having a built-in beam effect the arrangement is more likened to a simply supported beam, with its larger central deflection for a given load.

From the vertical misalignment figures and by referring to Fig.37. the reader should be able to deduce 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 misalignment can be checked against the permissible values supplied by the engine builder, often in the form of a graph as per Fig. 38. If any values exceed or equal maximum permissible values then bearings will have to be adjusted or renewed where required. Indication of incorrect bearing

FIG. 37
CRANK POSITIONS FOR DEFLECTION

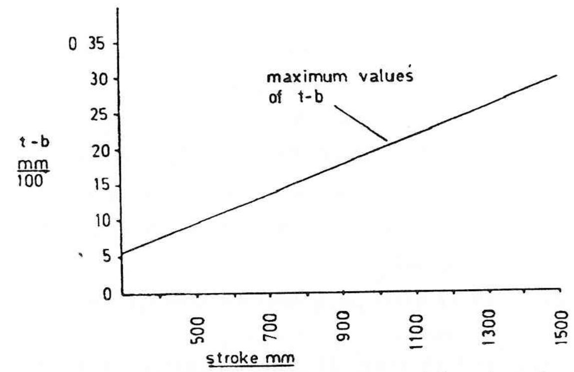


clearances may be given when the engine is running. In the case of medium or high speed diesels, load reversal at the bearings generally occurs. With excessive bearing clearances loud knocking takes place, white metal then usually gets hammered out.

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. Oil mist and vapour at a particular unit

FIG. 38
MAXIMUM VALUES



may be observed to increase – together with a hot bearing, this may lead to a crankcase explosion.

Regular checks must be made to ascertain the oxidation rate of the oil. If this is increasing then high temperatures are being encountered. n.b. as the oil oxidises (burns) its colour blackens.

CHOICE, MAINTENANCE AND TESTING OF LUBRICATING OIL FOR MAIN CIRCULATING SYSTEM

Choice

If the engine is a 'trunk type' then fuel and deleterious deposits from its combustion products may find their way into the crankcase. The oil should therefore be one which has detergent properties, these oils are sometimes called 'Heavy Duty'. Additives in these oils deter the formation of deposits 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.

Detergent oils may not be able to be water washed in a centrifuge, it is always advisable to consult the supplier.

Straight mineral oil, generally with an anti-oxidant and corrosion inhibitor added, is the type normally used in diesels whose working cylinder is separate from the crankcase.

Maintenance

When the engine is new correct pre-commissioning should give 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 clear discharges obtained from pipes before they are connected up, filters must 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 and the static oil within them become overheated.

3. Incorrect use of oil preheater for the purifier, *e.g* 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.

Drainings from scavenge spaces and stuffing boxes should not be put into the oil system and stuffing box and telescopic pipe glands must be maintained in good condition to prevent entry of water, fuel and air into the oil system.

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 some tank or container in which the oil is stationary and could possibly be stagnant.

Smelling the oil sample may give indication of fuel oil contamination or if acrid, heavy oxidation. Dark colour gives indication of oil deterioration, due mainly to oxidation.

Dipping fingers into the oil and rubbing the tips together can 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 indication of water contamination. But various tests are available to detect water in oil, *e.g.* 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 in the purifier is important, an increase would give 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 hours (or more often if suspect) 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. Water is generally permissible up to 0·2%, dangerous if sea water.

4. 1·0% Conradson carbon value. This is from cracked lubricating oil or residue from incomplete combustion of fuel oil.

5. 0·01 mg KOH/g Total Acid Number (TAN). The TAN is the total inorganic and organic acid content of the oil. Sulphuric acid from engine cylinders and chlorides from sea water give the inorganic, oxidation produces the weak organic acids. Sometimes the acids may be referred to as Strong and Weak.

LUBRICATION SYSTEMS

Lubrication systems for bearing and guides, etc. should be simple and effective. If we consider the lubrication of a bottom end bearing, various routes are available, the object would be to choose that route which will be the most reliable, least expensive and least complicated. We could supply the oil to the main bearing and by means of holes drilled in the crankshaft convey the oil to the bottom end bearing. This method may be simple and satisfactory for a small engine but with a large diesel it presents machining and stress problems.

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 crank pin by pipes.

A common arrangement, mainly adopted with engines having oil cooled pistons, is to supply the bottom end bearing with oil down a central hole in the connecting rod from the top end bearing. Fig. 39. shows an arrangement wherein a telescopic pipe-system is used and Fig.40. 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 could be less expensive.

With any of the bearings (excepting ball or roller) the main object is to provide as far as possible a good hydrodynamic film of lubricant (*i.e.* a continuous unbroken film of oil separating the working surfaces). Those factors assisting hydrodynamic lubrication are:

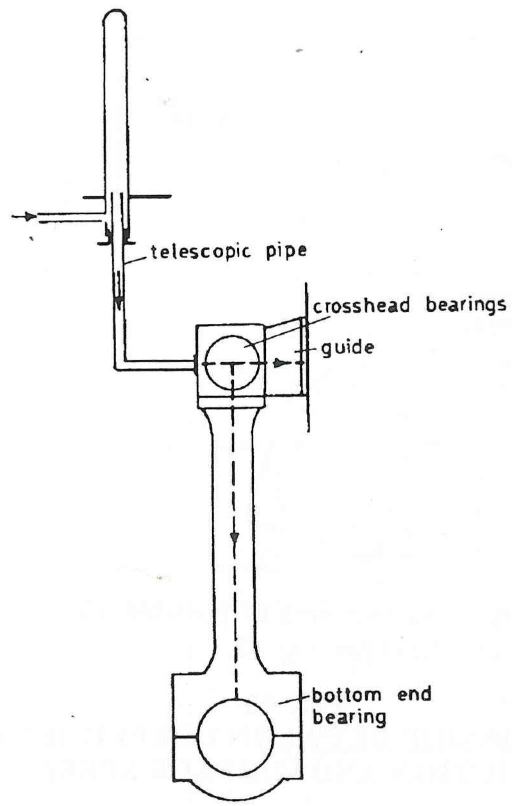
1. *Viscosity*. If the oil viscosity is increased there is less likelihood of oil film break down. However, too high a viscosity increases viscous drag and power loss.

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 bearing load and hence pressure (load/area) breaks down the oil film. In design, if the load is increased area can be increased by making the pin diameter larger – this will also increase relative speed.

FIG 39

LUBRICATION OF BEARINGS

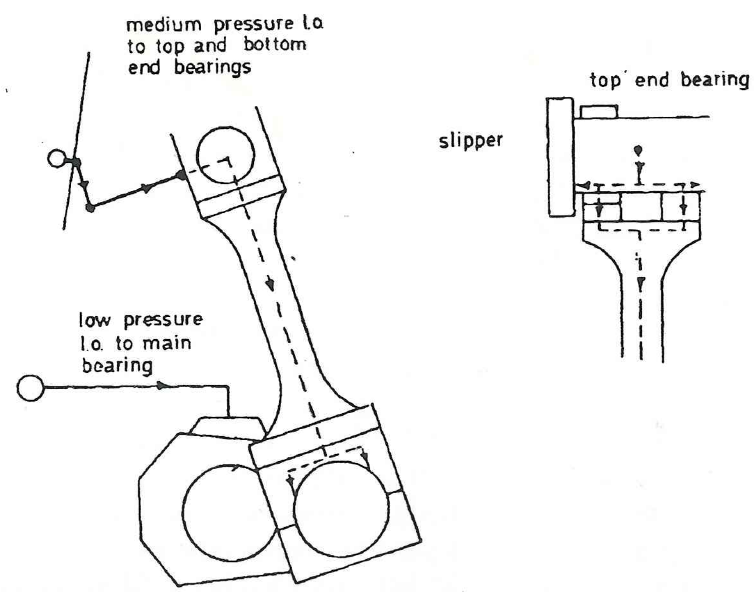


4. *Clearance*. If bearing clearance is too great inertia forces lead to 'bearing knock.' This impulsive loading results in pressure above normal and breakdown of the hydrodynamic layer. Fig. 41. illustrates the foregoing points graphically for a journal type of bearing.

Hydrodynamic lubrication should exist in main, bottom end and guide bearings. The top end bearing will have a variable condition, *e.g.* when at T.D.C. relative velocity between crosshead pin and bearing surface is zero and bearing pressure near or at maximum. Methods of improving top end bearing lubrication are:

1. Reversal of load on top end by inertia forces – only possible with medium or high speed diesels.

FIG 40
LUBRICATION SYSTEM FOR MAIN BEARINGS



2. Use as large a surface area as possible, *i.e.* the complete underside of the crosshead pin. Fig. 42.

FIG 41
RELATIONSHIP BETWEEN COEFFICIENT OF FRICTION AND SURFACE SPEED

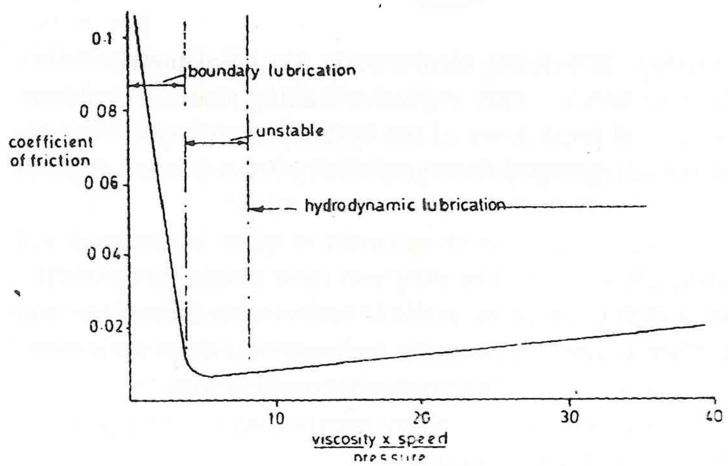
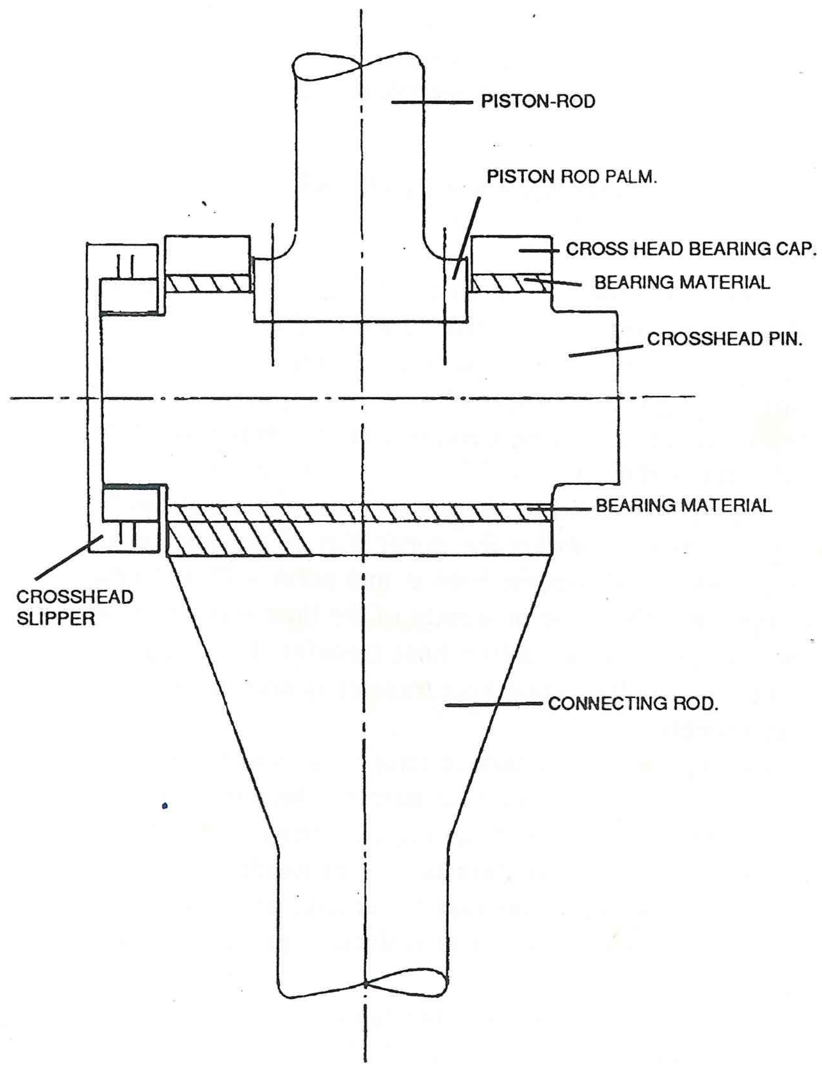


FIG 42
ONE PIECE LOWER BEARING CROSSHEAD DESIGN. SKETCH SHOWS LHS SLIPPER.



3. Avoid large axial variation of bearing pressure by more flexible seating and design. Fig. 43.

4. Increase oil supply pressure. Fig. 44. shows a method of increasing oil supply pressure to the top end bearing which tends to keep the crosshead pin 'floating' at all times. As the connecting rod oscillates, lubricating cross head oil is pumped at high pressure from the two pumps (only one is shown).

Increase in oil supply pressure can also be accomplished by installing lubricating oil booster pumps taking suction from L. O. system.

CYLINDERS AND PISTONS

Cylinders

Fig. 45. shows in section a cylinder liner from a large 2-stroke engine. The liner is manufactured from good quality lamellar 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. It can be seen in Fig. 46. 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 the technique of bore cooling good heat transfer is accompanied by high overall strength.

By maintaining the correct surface temperatures in the vicinity of the combustion space by good heat transfer, there is the risk of low temperature corrosion or cracking occurring in the lower portions of the liner. The solution to this problem is to either insulate the cooling water spaces that are at risk, or utilise a load controlled cylinder cooling system to maintain optimum cylinder liner temperature. Fig 131.

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. Fig. 45. The "O" rings are in groups of two, the space between them being open to the atmosphere. Leakage

FIG 43
CROSSHEAD WITH FLEXIBLE BEARING SUPPORTS

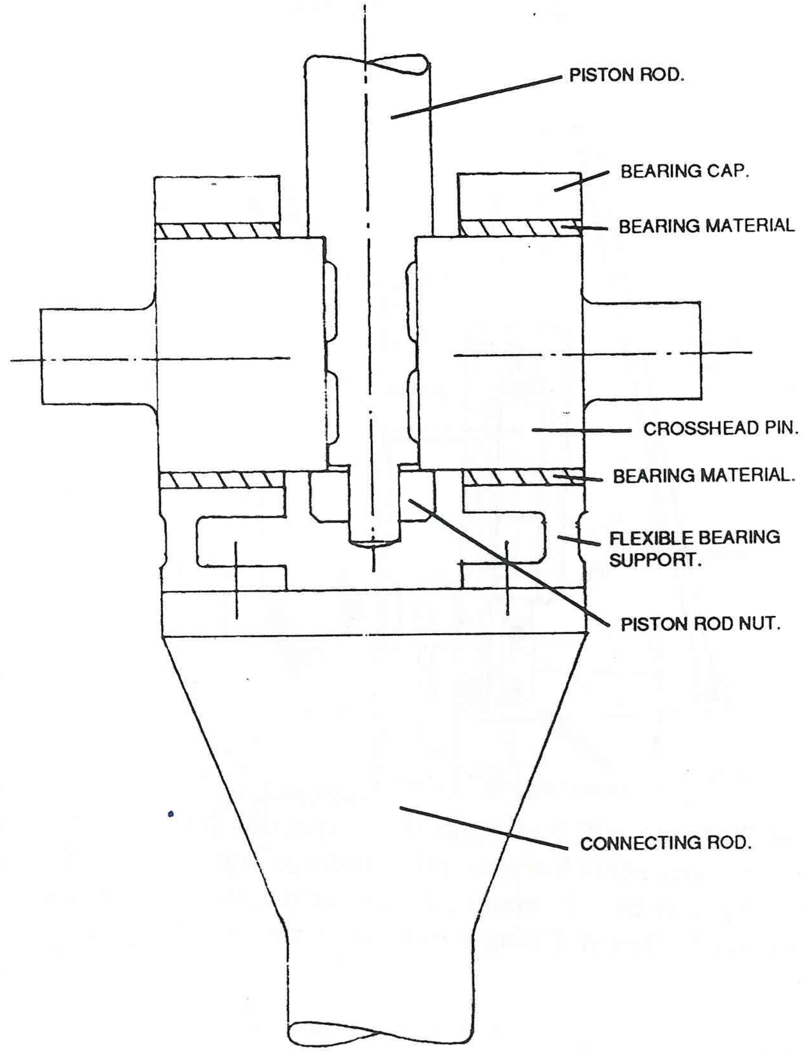
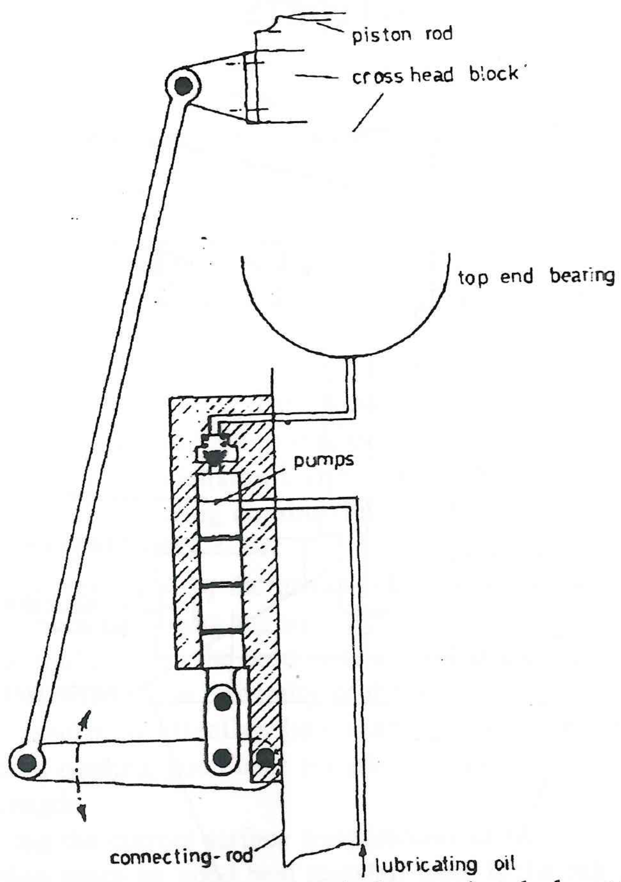


FIG 44
CROSS HEAD OIL PUMP



past an "O" ring will leak from this inspection hole not only alerting the engineers but preventing leakage into the scavenge space. Great care must be exercised to ensure that the "O" rings are not damaged when re-fitting a cylinder liner into the cylinder jacket.

Cylinder Lubrication

The principal objects of cylinder lubrication are:

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.

FIG 45
2 STROKE CYLINDER LINER

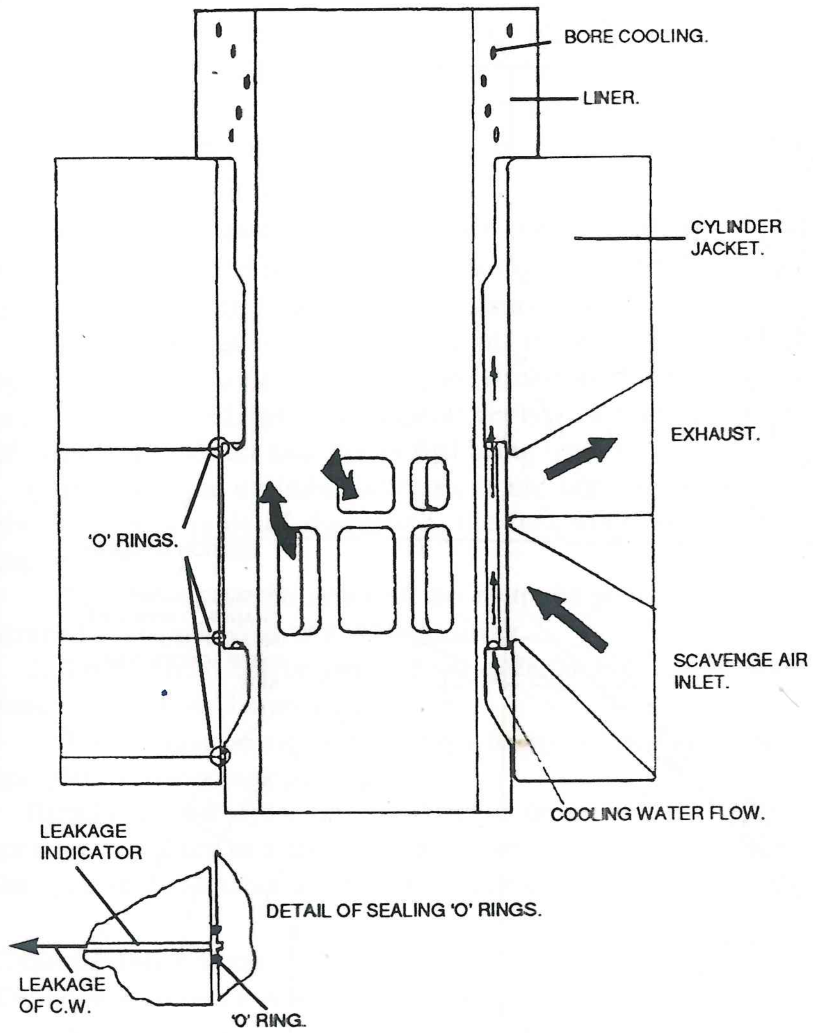
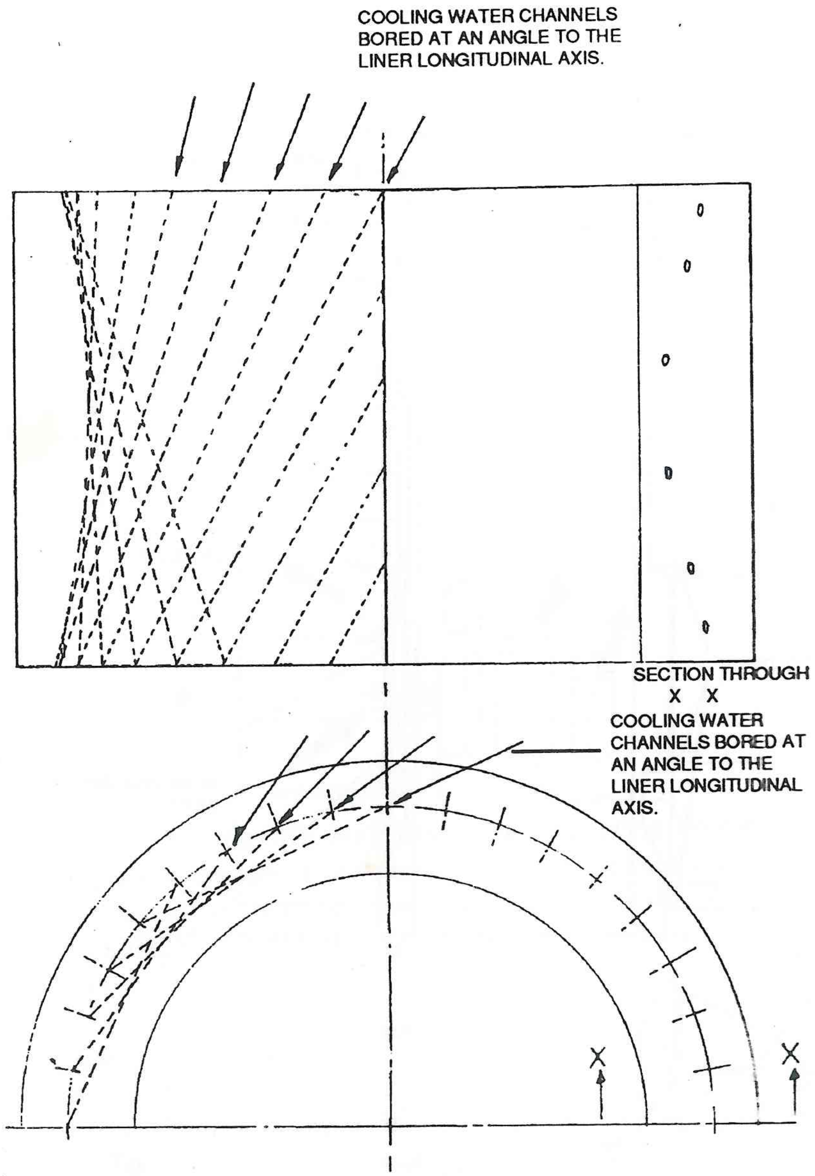


FIG 46

DIAGRAMMATIC VIEW OF CYLINDER LINER BORE COOLING



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 by blue smoke and increased oil consumption. As the oil burns it should leave as little and as soft a deposit as possible. Over lubrication should be avoided.

When the engine is new, cylinder lubrication rate should normally be greater than when the engine is run in. Reasons for this increased lubrication are: (1) surface asperities will, due to high local temperatures, cause increased oxidation of the oil and reduce its lubrication properties, (2) sealing of the rough surfaces is more difficult, (3) worn off 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.

Ideally, timed injection of lubricant delivering the correct measured quantity to a specific surface area at the correct time in the cycle is the aim, but is difficult to achieve in practice.

Cylinder Liner Wear

Cylinder liner wear can be divided into:

- Abrasive wear.
- Corrosive wear.

Abrasive wear

This occurs when abrasive particles enter the combustion space with scavenge air or as a result of poor quality or contaminated fuel. Instances of extremely high abrasive wear rates have occurred in the past due to the burning of fuel heavily contaminated by catalytic fines.

Corrosive wear

This is the more common cause of cylinder liner wear, caused when burning heavy fuel which contains significant amounts of sulphur. As the fuel burns the sulphur combines with oxygen to produce oxides of sulphur which 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. Fig. 47.

To minimise cylinder liner wear it is imperative that ship's engineers 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.
- Well maintained equipment.

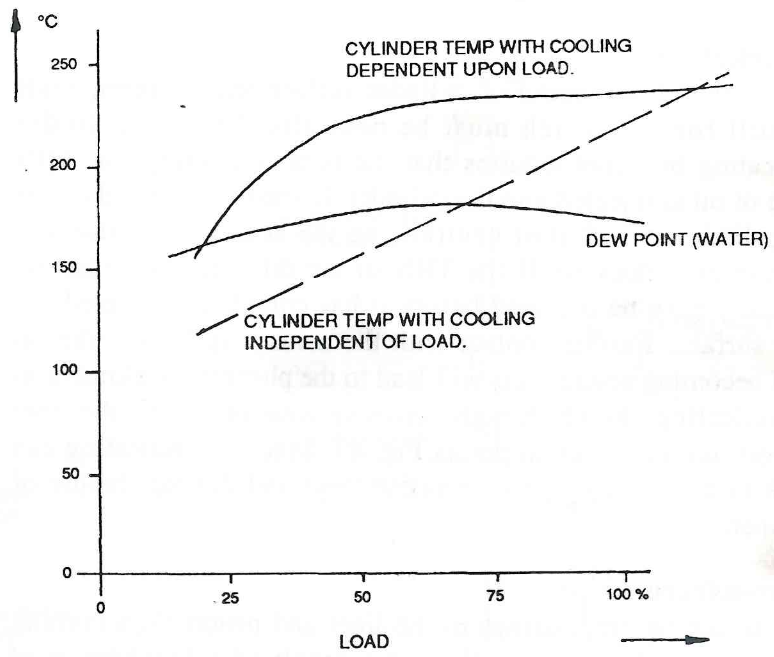
The deterioration of fuel quality that has taken place years coupled with the increased pressures and temperatures that occur during the combustion process have resulted in liners and piston rings operating under very severe conditions. Despite these adverse operating conditions cylinder liner wear rates have been reduced in recent years with large 2-stroke manufacturers claiming 0·03 mm/1000 hrs and medium speed 4-stroke engine manufacturers claiming wear rates of 0·02 mm/1000 hrs when operating on heavy fuel.

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.

FIG 47

TEMPERATURE OF CYLINDER LINER SURFACE THROUGHOUT. ENGINE LOAD RANGE.



- The development of load dependent temperature control of cooling water which maintains the cylinder liner temperature at optimum level. Fig. 131 [cooling water section].
- The use of good quality cast iron with sufficient hard phase content for cylinder liners.
- Careful design of piston ring profiles to maximise lubricating oil film thickness.
- Improvements in lubricating oil distribution across cylinder liner surface. This includes multi-level injection in 2-strokes engine and forced piston skirt lubrication in 4-stroke engines. Fig. 51.
- Improved separation of condensate from scavenge air.

Cylinder liner wear profile

Fig. 48. shows the wear profile of both a 4-stroke and 2-stroke engine cylinder liner. It can be seen that the greatest wear occurs in the upper part of the liner adjacent to 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.

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. Immediately the oil enters the cylinder it will start 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 Fig. 49. Severe cloverleafing can result in gas blow-by past the piston rings and ultimate failure of the liner.

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 instantaneous seizure and tearing taking place. 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 is persistent and as a result of inadequate lubriation.

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

FIG 48
LINER WEAR PROFILE

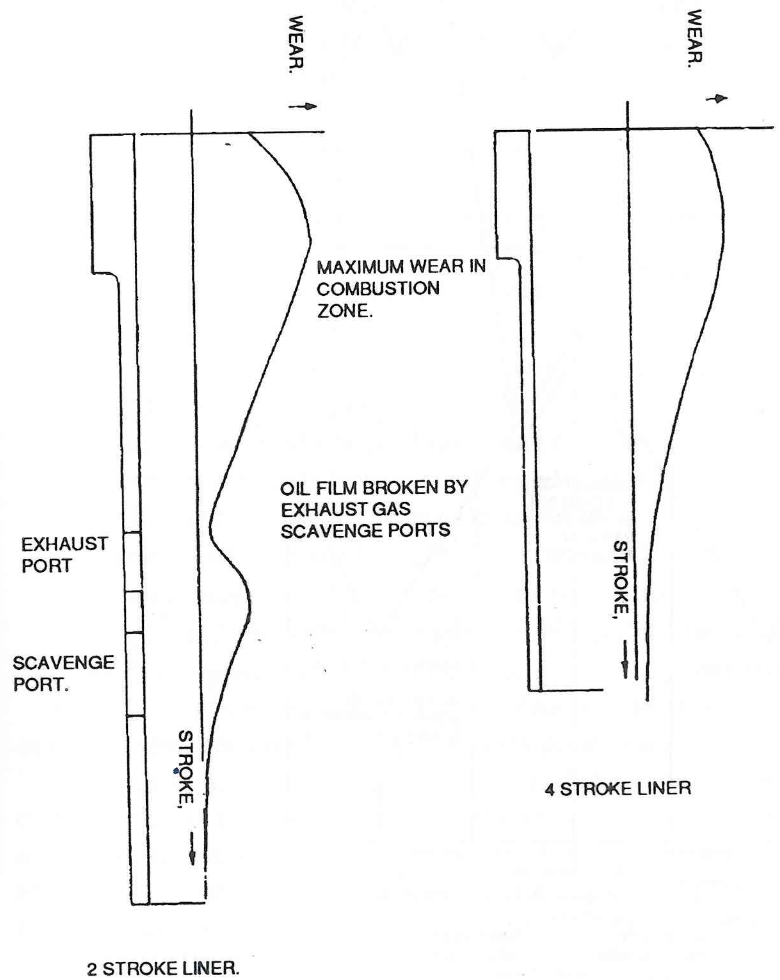
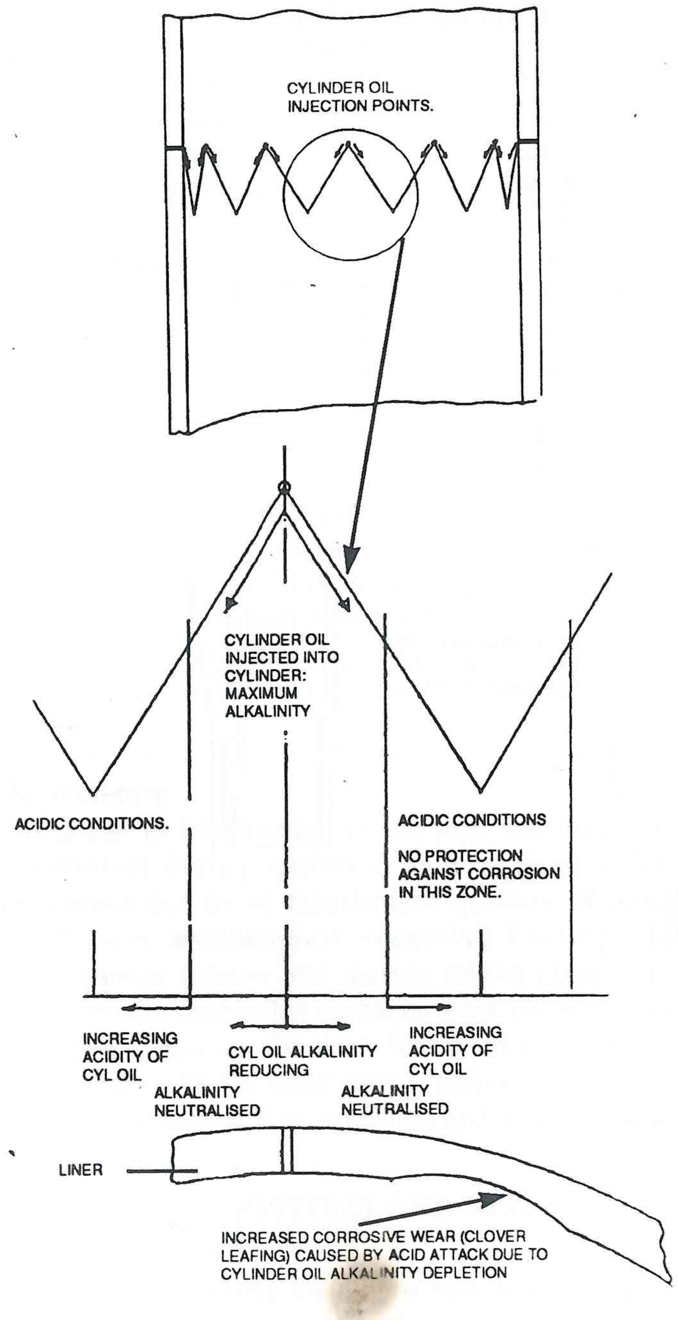


FIG 49
INCREASED CORROSIVE WEAR OF CYLINDER LINER. (CLOVERLEAFING).



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. Because 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.

- Piston crowns which are highly stressed mechanically and thermally are made from cast or forged steel. Cast iron inserts are fitted to the ring grooves to resist wear.
- Piston skirts are made from cast iron which has superior rubbing properties than either cast or forged steel. To reduce weight and reduce inertia loads aluminium is used in some medium speed 4-stroke applications.

Fig. 50. shows a piston for a large Sulzer engine. The crown is of forged steel and combines strength with good heat transfer.

Strength is achieved by using an overall thick section piston crown which is then bore cooled. Intensive cooling is achieved by the cocktail shaker effect of the water. With air present in the piston [this comes from the telescopic system, it being necessary to provide a cushion and prevent water hammer] together with water, the inertia effect coupled with the bore cooling leads to very effective cooling as the piston goes over TDC.

Fig. 51. shows an oil cooled piston of a large modern B.& W. engine. The piston crown of this piston is also manufactured from forged steel but in this case the section is relatively fine. Strength being achieved by the "strong back" principle which supports the

FIG 50
WATER COOLED PISTON WITH BORE COOLING

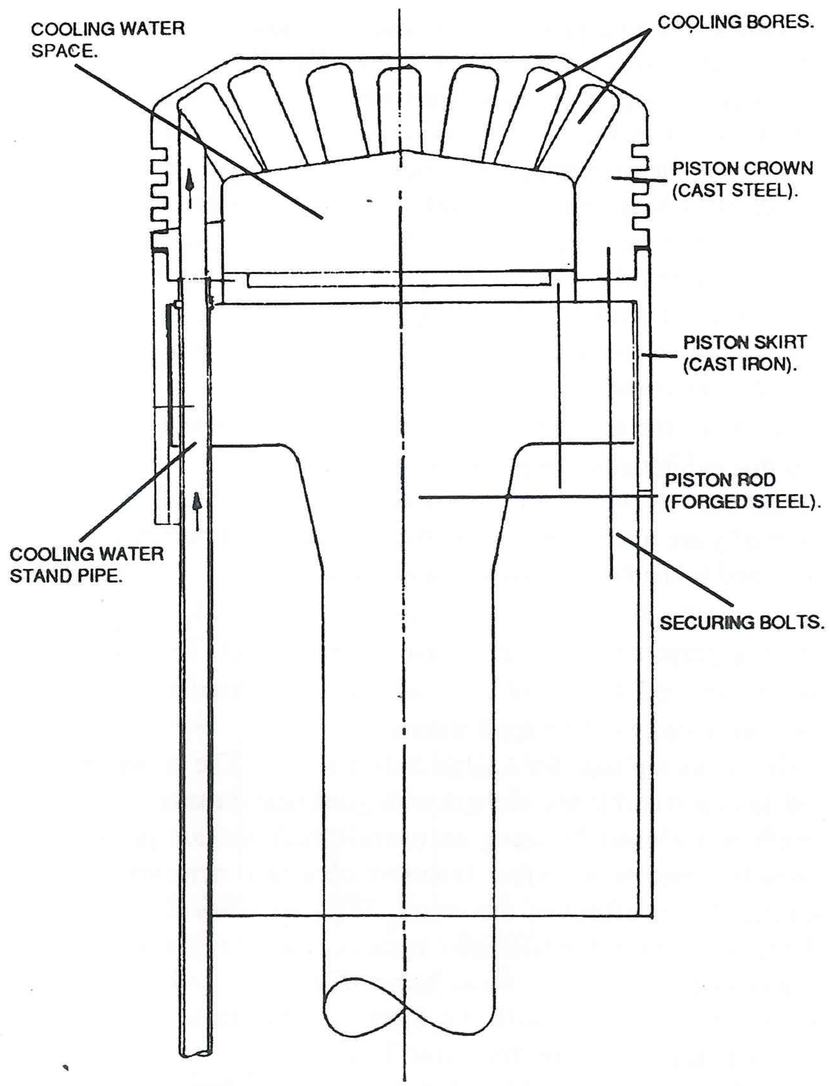
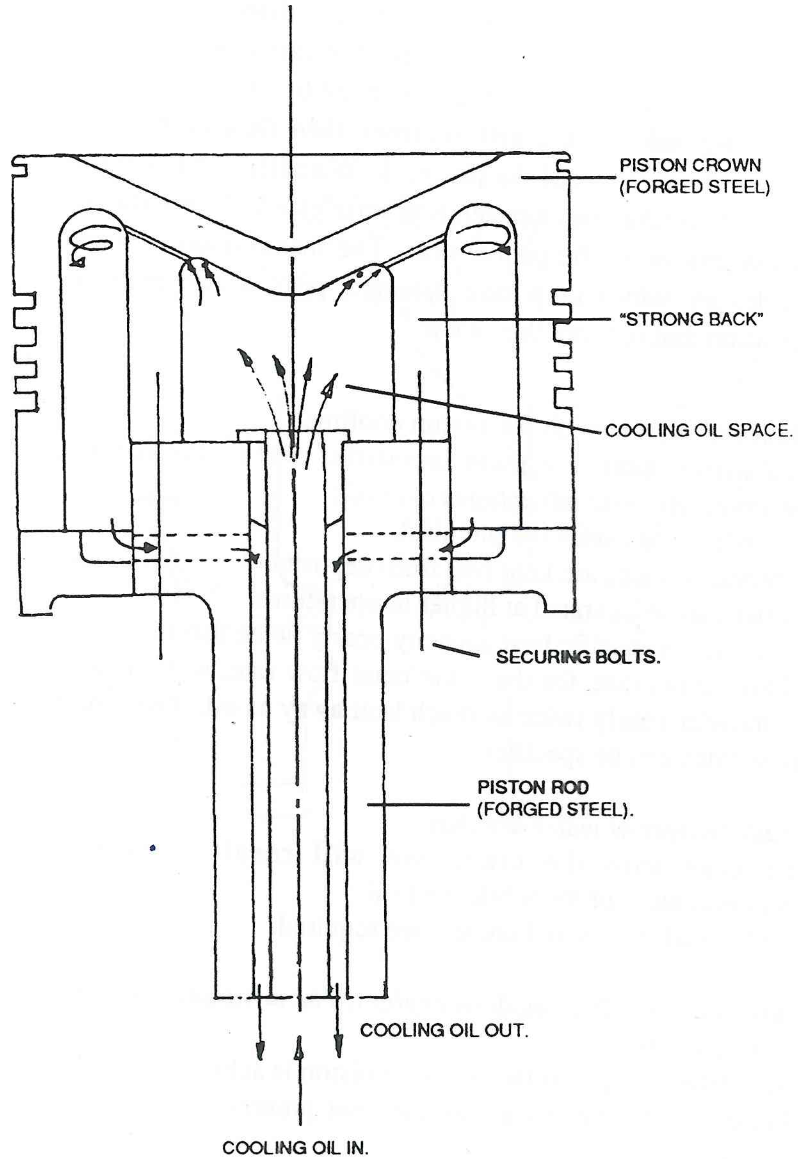


FIG 51
OIL COOLED PISTON FROM 2 STROKE ENGINE.



piston crown from inside. Bore cooling is employed but in this design the bores act as nozzles through which the oil flows radially, spraying onto the underside of the piston crown, before flowing to the drain.

Fig. 52. shows a piston from a medium speed 4-stroke engine. This design transmits the combustion forces directly onto the gudgeon pin. The piston crown is of forged or cast steel whilst the skirt is of nodular cast iron. Cooling is effected by oil flowing from the connecting rod into the piston crown then flowing radially outwards to effectively cool the piston. In Wartsilla engines some of this oil is then taken out through four nozzles which feed the oil distribution groove in the piston skirt. The manufacturers claim that this design, which they have patented, provides an even oil film formation that reduces liner wear.

The Choice of water or oil for piston cooling.

Distilled water, kept free from impurities and in the correct alkaline state, has some advantages over oil:

- It is relatively cheap and plentiful.
- Internal surfaces are kept free from deposits.
- Water can be operated at higher temperatures.
- Water has a specific heat capacity nearly twice that of oil. [This means that, for the same mass flow rate, water is able to transfer nearly twice as much heat away as oil, lower mass flow rates can be specified.]

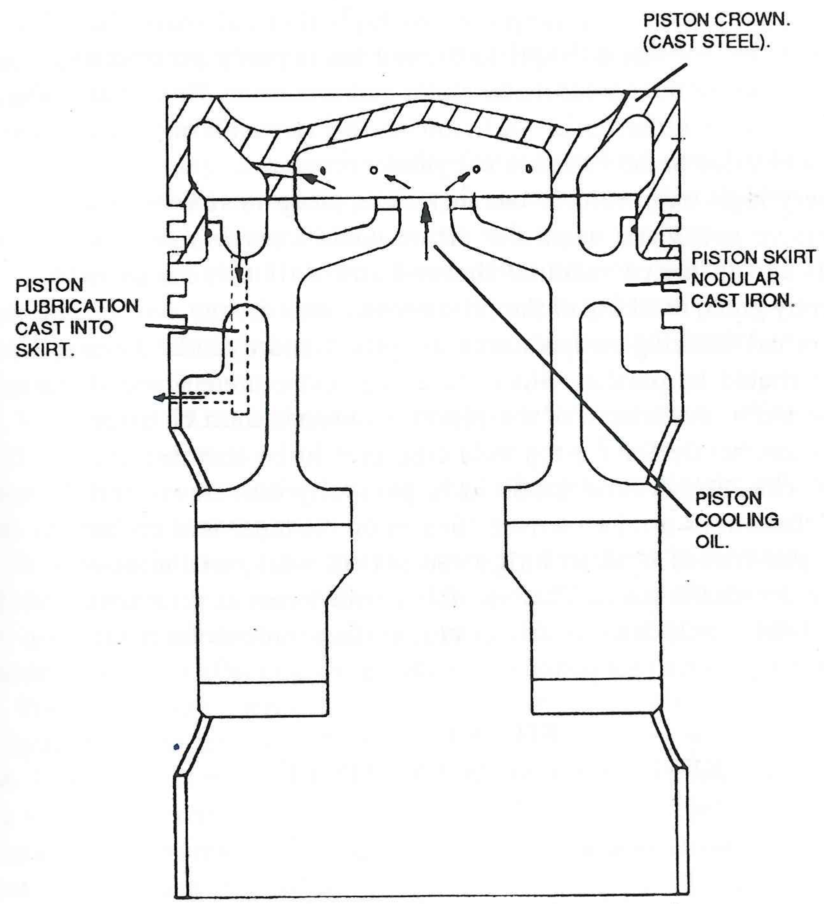
The disadvantages of water are that:

- Leakage into the crankcase will result in serious contamination of the lubricating oil.
- Additional pumps and coolers are required.

Oil is used extensively in modern engines. The advantages claimed for this medium are:

- Simplified supply of the oil to the piston is achieved.
- Leakage into the crankcase does not present contamination problems.
- The lower thermal conductivity results in a less steep temperature gradient over the piston crown.

FIG 52
COMPOSITE PISTON SUITABLE FOR A HIGH OUTPUT MEDIUM SPEED 4-STROKE DIESEL ENGINE.



The disadvantages of oil are:

- The temperatures must be kept relatively low in order to limit oxidation of the oil.
- If overheating occurs there is a a possibility that carbon deposits could form on internal surfaces and the danger that carbon particles could enter the lubricating oil system.

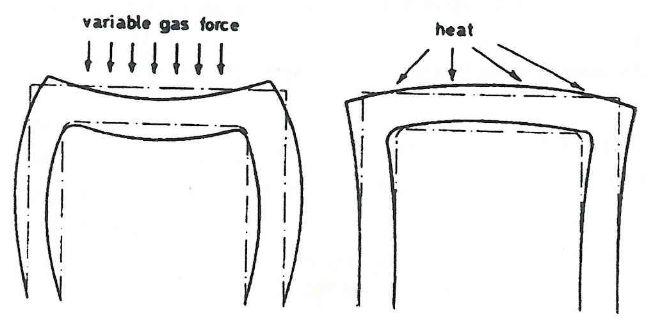
Failure of pistons due to Thermal Loads

When a piston crown is subjected to high thermal load, the material at the gas side attempts to expand but is partly prevented from doing so by the cooler metal under and around it. This leads to compressive stresses in addition to the stresses imposed mechanically due to the variation in cylinder pressures.

At very high temperatures the metal can creep to relieve this compressive stress and when the piston cools a residual tensile stress is set up hence residual thermal stress. If this stress is sufficiently great, cracking of the piston crown may result.

At normal working temperatures the piston and cylinder liner surfaces should be parallel. Since there is a temperature gradient from the top to the bottom of the piston, allowance must be made during manufacture for the top cold clearance to be less than the bottom. The temperature gradient is generally non-linear and thermal distortions produce tensile stresses on the inner wall of the piston, gas forces tend to bulge the piston wall out thereby reducing the tensile stress. This variable tensile stress at very high thermal loads could lead to cracks propagating through from the inside of the piston to the piston ring grooves.

FIG. 53
EFFECT OF GAS AND HEAT



Piston Rings

Properties required of a piston ring:

1. Good mechanical strength, it must not break easily.
2. High resistance to wear and corrosion.
3. Self lubricating.
4. Great resistance to high temperatures.
5. Must at all times retain its tension to give a good gas seal.
6. Be compatible with cylinder liner material.

The above properties are the ideal and therefore difficult to achieve in practice. Materials that are used to obtain as many of the desired properties as possible are as follows:

1. Ordinary grey cast iron, in order that it may have good wear resistance and self lubricating property it must have a large amount of graphite in its structure. This however reduces its strength.

2. Alloyed cast iron, elements and combinations of elements that are alloyed with the iron to give finer grained structure and good graphite formation are: Molybdenum, Nickel and Copper or Vanadium and Copper.

3. Spheroidal Graphitic iron, very good wear resistance, not as self lubricating as the ordinary grey cast iron. These rings are usually given a protective coating, *e.g.* chromed or aluminised, etc. to improve running-in.

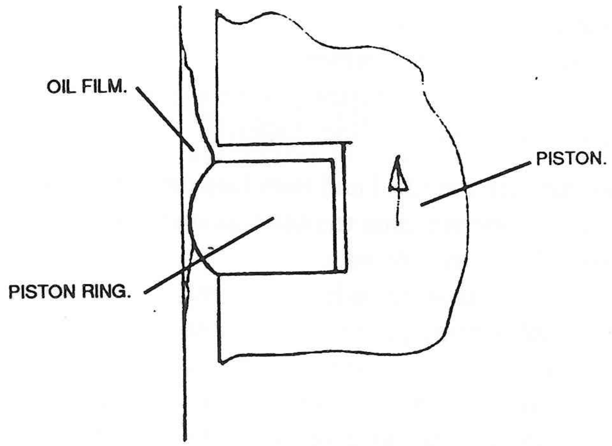
It is possible to improve the properties by treatment. In the case of the cast irons with suitable composition they can be heat treated by quenching, tempering or austempering. This gives strength and hardness without affecting the graphite.

Piston rings are often contoured to assist in the establishment of a hydrodynamic lubricating oil film and so reduce liner wear Fig. 54. It is common practice for manufacturers to specify a ring pack in which the first and second compression rings, subjected to higher temperatures and pressures, differ from the lower rings. It is important when installing new rings that the manufacturers recommendations are followed since ring failure may result if incorrect rings are fitted.

In addition to compression rings 4-stroke medium speed engines also employ oil control, or oil scraper rings Fig. 55. Unlike compression rings, which help promote the formation of an oil film, oil scraper rings scrape the oil from the cylinder liner and return it to the sump. Many designs of oil scraper rings can only be

FIG 54

2 STROKE ENGINE PISTON RING PROFILE.



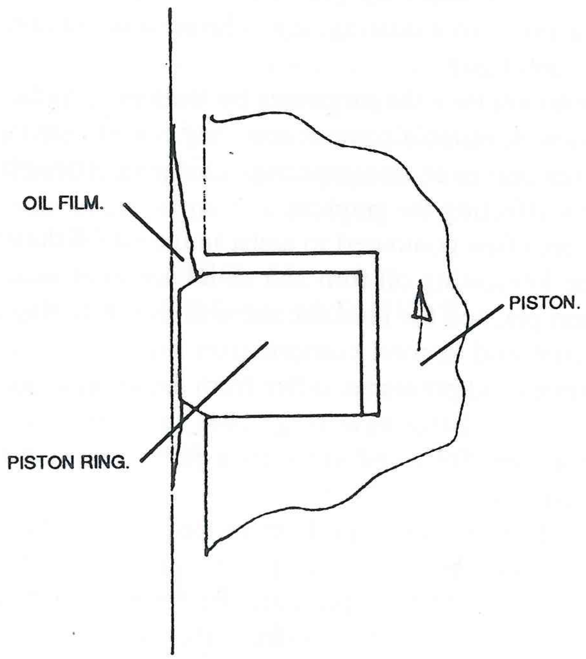


FIG 55

4 STROKE PISTON RINGS.