fitted in one direction and care must be exercised when installing these rings. Without these rings lubricating oil in the upper cylinder would be burnt during combustion resulting in extremely high oil consumption. As the oil scraper rings wear their effectiveness in returning the oil to the sump reduces with high oil consumption as the consequence. Oil scraper ring wear may be the limiting factor when deciding cylinder overhaul periods for medium speed 4-stroke engines.

Manufacture

1. Statically cast in sand moulds to produce either a drum from which a number of piston rings would be manufactured or an individual ring.

2. Centrifugally cast to produce a fine grained non-porous drum of cast iron from which a number of piston rings will be machined. The statically cast rings, either drum or single casting, may be made out of round. The out of round blanks are machined in a special lathe that maintains the out of roundness. Rings manufactured in this way are expensive but ideal.

Most piston rings are made from circular cast blanks which are machined to a circular section on their inner and outer diameters. In order that the rings may exert radial pressure when fitted into the cylinders they are split in tension. Tensioning is done by cold deformation of the inner surface by hammering or rolling. The finished ring would be capable of exerting a radial pressure from 2 to 3 bar and have a Brinel hardness from 1600 to 2300 (S.I. units). Large diesel engine cylinder liners have a hardness range similar to the above.

Piston ring defects and their causes

1. Incorrectly fitted rings. If they are too tight in the grooves the rings could seize causing overheating, excessive wear, increased blow-past, etc. If they are too slack in the grooves angular working about a circumferential axis could cause ring breakage and piston groove damage. If the butt clearance is too great, excessive blow-past will occur.

2. Fouling due to deposits on the ring sides and their inner diameters, this could lead to rings sticking, breakage, increased blowpast and scuffing.

3. Corrosion of the piston rings can occur due to attack from corrosive elements in the fuel ash deposits.

4. If the ring bearing surfaces are in poor condition or in any way damaged (this could occur during installation) scoring of cylinder liner may take place, if the ring has sharp edges it will inhibit the formation of a good oil film between the surfaces.

Due to uneven cylinder liner wear the piston ring diameter changes during each stroke, this leads to ring and groove wear on the horizontal surfaces. This effect obviously increases as differential cylinder liner wear increases. Oscillation of the piston rings takes place in the cycle about a circumferential axis approximately through the centre of the ring section, and if the inner edges are not chamfered they can dig into the piston groove lands. Keeping the vertical clearance to a working minimum will reduce the oscillatory effect.

When considering piston rings perhaps the most destructive force at work is hammering. This is caused by relative axial movement between piston and ring as a result of gas loading and inertia when the piston changes direction at BDC. The hammering results in enlargement of the piston ring groove and may result in ring breakage. Cast steel, forged steel or aluminium pistons usually have ring groove landing surfaces protected to minimise the effects of hammering. This can be either:

- Flame hardening - top and bottom on upper grooves.
- Chromium plating- top and bottom on upper grooves.
- The fitting of cast iron inserts.

In 2-stroke engines, piston rings have to pass ports in the cylinder wall. Each time they do, movement of segments of the rings into the ports can take place. This would be more pronounced if the piston ring butts are passing the ports. It is possible for the butts to catch the port edge and bend the ring. In order to avoid or minimise this possibility, piston rings may be pegged to prevent their rotation or they may be specially shaped.

Inspection of pistons, rings and cylinders

Withdrawal of pistons their examination, overhaul or renewal, together with the cleaning and gauging of the cylinder liner, is a regular feature of maintenance procedures. Frequency of which depends upon numerous factors, such as: piston size, material and method of cooling; engine speed of rotation; type of engine, 2- or 4-stroke; fuel and type of cylinder lubricant used.

With high speed diesel engines of the 4-stroke type running time between piston overhauls is generally greater than that for large slow running two stroke engines. This can be attributed to the facts that: the engine is usually unidirectional, hence reduced numbers of stops and starts with their attendant wear and large fluctuations of thermal conditions. Small bore engines are easier to cool, cylinder volume is proportional to the square of the cylinder diameter hence increasing the diameter gives greatly increased cylinder content and high thermal capacity. Thus overhaul time can vary between about 2000 to 20 000 h.

Pistons and cylinder liners on some engines can be inspected without having to remove the piston.

After scavenge spaces have been cleaned of inflammable oil sludge and carbon deposits, each piston can in turn be placed at its lowest position. The cylinder liner surfaces can then be examined with the aid of a light introduced into the cylinder through the scavenge ports. The cylinder liner surfaces should have a mirror-like finish. However, black dry areas at the top of the liner indicate blow past of combustion gases. Dull vertically striped areas indicate breakdown of oil film and hardened metal surface (this is caused by metal seizure on a micro scale leading to intense heating).

After inspection of a cylinder the piston can be raised in steps in order to examine both the piston and the rings. Heavy carbon deposits on piston crown and burning away of metal would indicate incorrect fuel burning and poor cooling. Piston rings should be free in the grooves, have a well oiled appearance, be unbroken and worn smooth and bright on the outer surface. If they are too worn then sharp burrs can form on the edges which enable them to act as scraper rings, preventing good oil film formation.

Large 2-stroke engines

The cylinder covers of loop scavenged engines tend to be relatively simple symmetrical designs to avoid the problems of differential expansion and the consequent stresses. Early Sulzer designs were two piece with a cast iron main component and a central cast steel insert containing the valves. In this design cooling water is introduced into the cylinder cover through nozzles which ensure

that the water flows tangentially thus minimising impingement on internal surfaces thus reducing the possibility of erosion Fig. 56. More modern engines, which operate at higher temperatures and pressures, have one piece forged steel cylinder covers. This design employs bore cooling which allows the cooling water to pass very close to the combustion chamber effectively maintaining safe surface temperatures. Fig. 57.

FIG. 56
TWO PIECE CYLINDER COVER: SULZER RND TYPE.

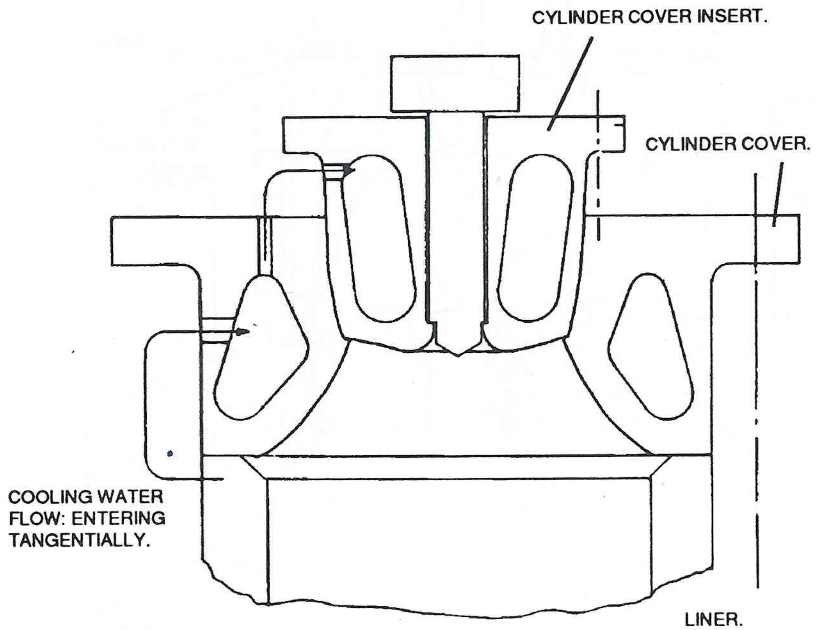
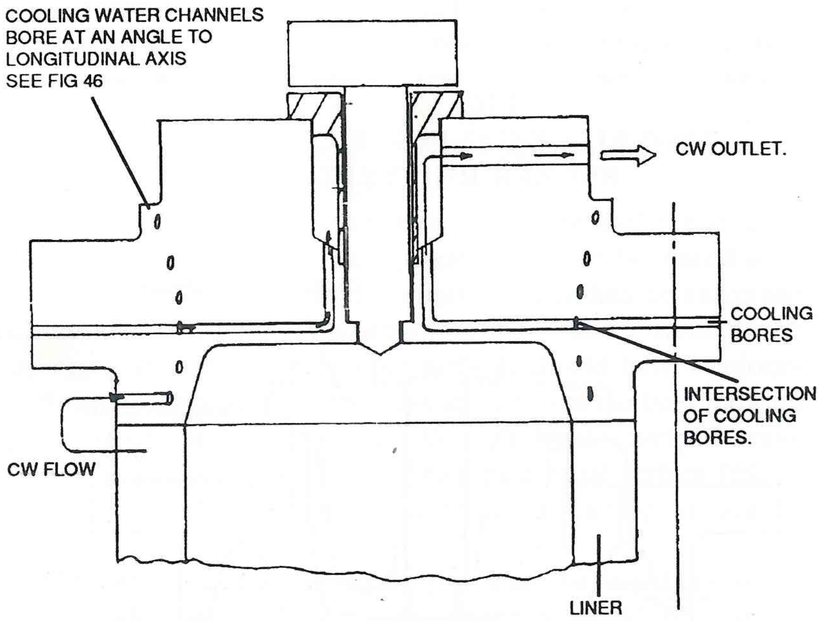
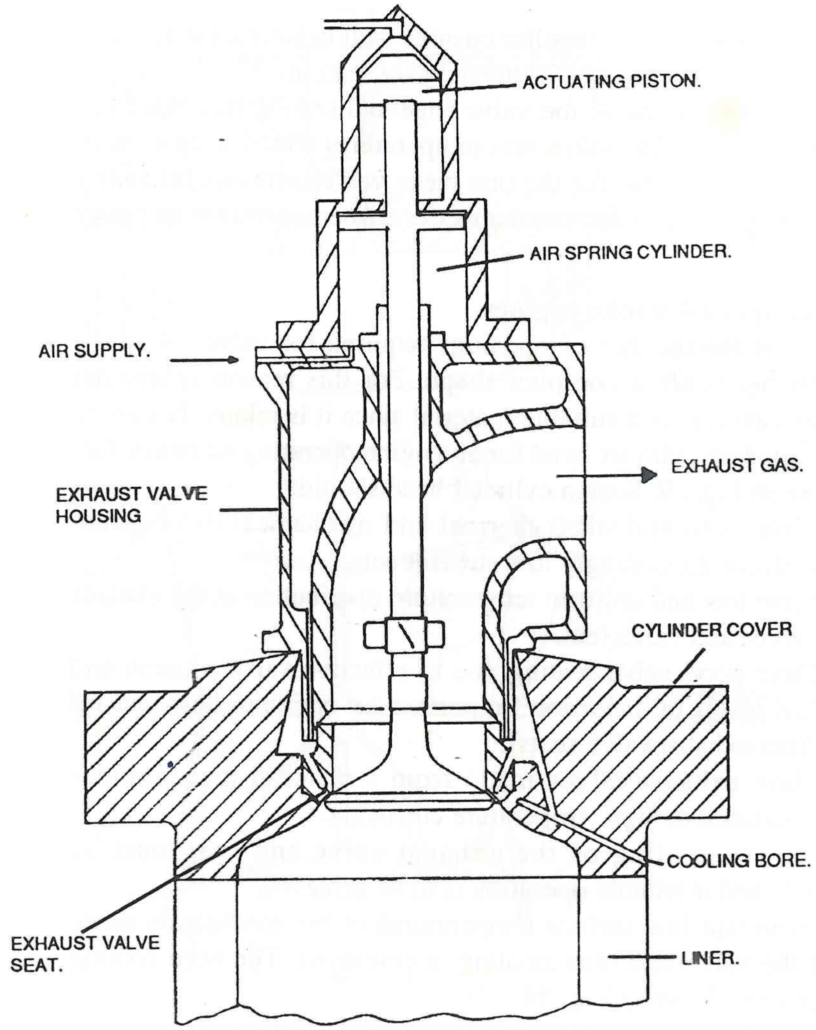


FIG. 57
ONE PIECE CYLINDER COVER
SULZER RND M.



With the exception of opposed piston configurations uniflow engines have a single central exhaust valve in removable casting installed in the cylinder cover. Fig. 58. The cylinder cover is manufactured from forged steel and cooling is accomplished through radial cooling bores close to the combustion chamber surface. The exhaust valve cage and seat ring are also bore cooled.

FIG. 58
HYDRAULICALLY ACTIVATED CENTRAL
EXHAUST VALVE FOR LARGE SLOW-SPEED
2-STROKE DIESEL ENGINE



Exhaust valves

In recent years manufacturers have adopted the hydraulically actuated valves in favour of mechanical pushrods and rockers. The advantages claimed for this configuration are:

- There is no transverse thrust from hydraulic actuators. Thrust is purely axial resulting in less guide wear.
- Controlled landing speed ensures minimum stress on valve and seat.
- Valve rotation by impeller ensures well balanced thermal and mechanical stress and uniform valve seating.

The extensive cooling of the valve cage and seating ring results in relatively low exhaust valve seat temperatures which coupled with the choice of Nimonic for the one piece valve increases reliability and the intervals between overhauls even when operating on heavy fuel.

Medium speed 4-stroke engines

Because of the number of openings required for valves, 4-stroke cylinder heads are a complex shape. For this reason spheroidal graphite cast iron is a suitable material since it is relatively easy to cast. A modern cylinder head for an engine operating on heavy fuel is shown in Fig. 59. Such a cylinder head should:

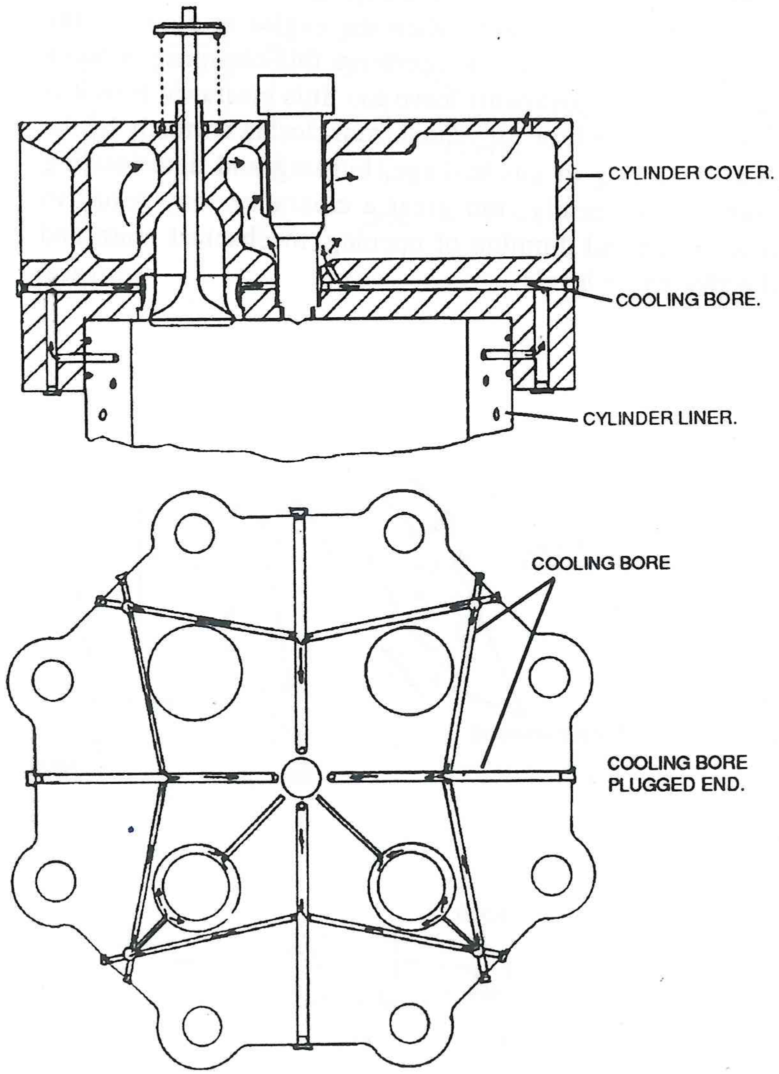
- Have even and small thermal and mechanical deformation with correspondingly low stress levels.
- Have low and uniform temperature distribution at the exhaust valves and valve seat.
- Have good valve seating due to effective valve rotation and low levels of distortion [important for optimum heat transfer from exhaust valve seats].
- Have exhaust valves made from a material that provides resistance to high temperature corrosion.

Effective cooling of the exhaust valve and seat must be accomplished if reliable operation is to be achieved.

To maintain low surface temperatures in the combustion space and at the valve seat bore cooling is employed. The bore cooling passages are shown in Fig. 59.

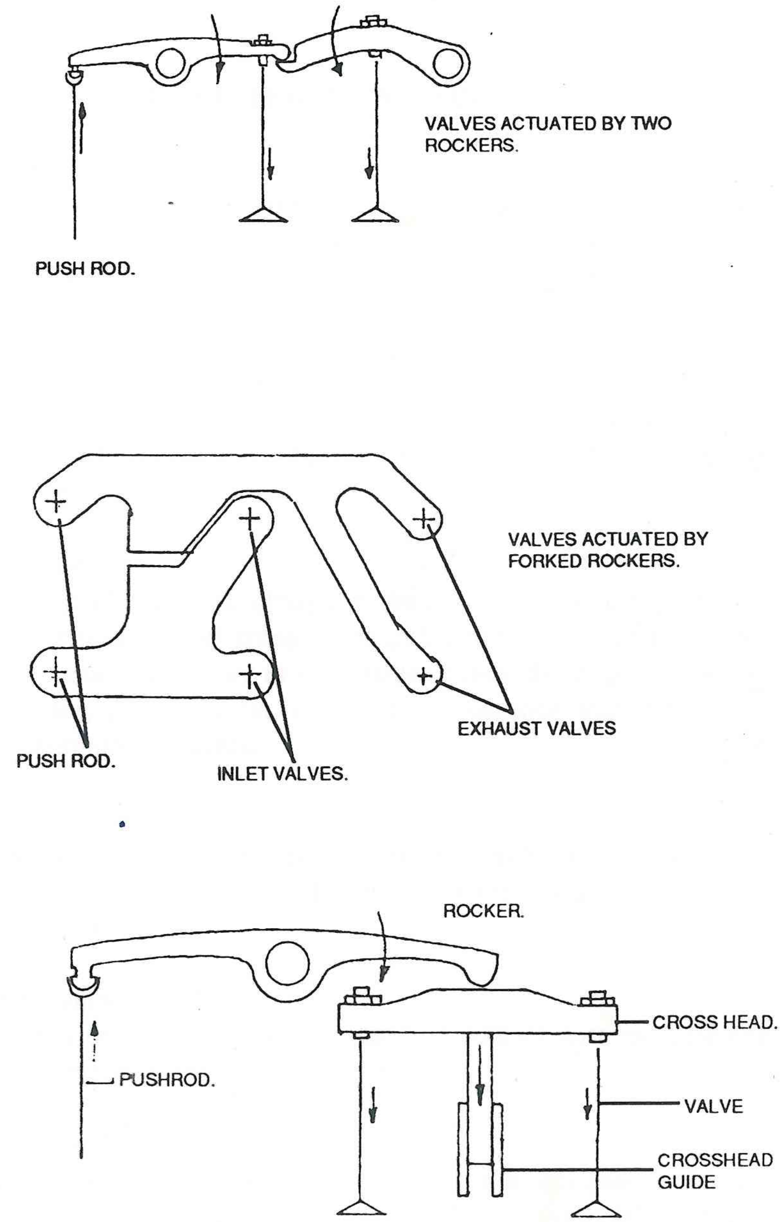
Four valves are usually employed on 4-stroke engines. This configuration allows the designer to maximise the cross-sectional area of inlet and exhaust ports and so improve the flow through the cylinder. This arrangement results in more complicated valve

FIG 59
4 STROKE DIESEL ENGINE CYLINDER COVER WITH BORE COOLING



actuation since two exhaust and two inlet valves must each be operated by one push rod. The various ways that the valves are actuated can be seen in Fig. 60. All of the designs shown control the valves together, it is important that, following maintenance, adjustments are made correctly. Clearance is allowed between the valve stem and the rocker arm when the engine is cold. As the engine attains normal running temperature this clearance is taken up by expansion. If adjustments leave too little clearance then it is likely that the valve will be prevented from closing correctly by the valve gear, resulting in gas leakage, burning and deteriorating performance. Conversely, too great a clearance may result in reduced valve lift and duration of opening, mechanical noise and reduced performance levels.

FIG 60
ALTERNATIVE METHODS OF VALVE ACTUATION



CHAPTER 3

FUEL INJECTION

DEFINITIONS AND PRINCIPLES

Atomisation

The break up of fuel into minute spray particles so as to ensure an intimate mixing of air and fuel oil is known as atomisation. The surface area/volume ratio of a fuel-oil droplet increases as its diameter decreases Fig. 61. The effect of this is that a smaller droplet can present a greater percentage of its molecules to contact with the available air than can a larger droplet. The smaller fuel-oil droplets then the more effective is the atomisation resulting in more rapid and complete combustion with maximum heat release from the fuel.

Turbulence

A swirl effect of air charge in the cylinder which in combination with atomised fuel spray gives intimate mixing and good overall combustion. Requires to be 'designed into' the engine by attention to liner, piston, ports, etc., details together with air pressure-temperature gradients.

Penetration

Ability of the fuel spray droplets to spread across the cylinder combustion space so as to allow maximum utilisation of volume for combustion.

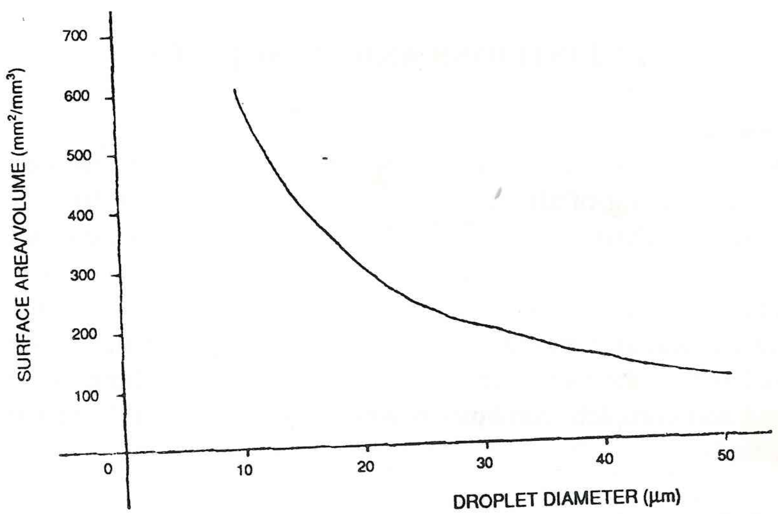
Impingement

Excess velocity of fuel spray causing contact with metallic engine parts and resulting in flame burning.

Sprayer Nozzle

The arrangement at the fuel valve tip to direct fuel in the proper direction with the correct velocity. If the sprayer holes are too short the direction can be indefinite and if too long impingement can

FIG 61
THE RELATIONSHIP BETWEEN FUEL DROPLET SIZE & SURFACE AREA/VOLUME RATIO



occur. If the hole diameters are too small fuel blockage (and impingement) can take place, alternatively too large diameters would not allow proper atomisation. In practice each manufacturer has a specific design taking into account method of injection, pressure, pumps, etc. Even with a particular engine different pressure, may be specified for different applications. For example, nozzles may be specified for different applications. For example, engines engaged in slow steaming, for reasons of economy, may be supplied with fuel valve sprayer nozzles with smaller holes of differing geometry than engines at higher powers. This measure improves the atomising and penetration performance of fuel valves at part load due to the restoration of fuel velocity through the nozzle. This yields improved economy at lower loads. The nozzles must be changed to the original size prior to operating at maximum power. As a generalisation the sprayer hole length: diameter ratio will be about 4:1, maximum pressure drop ratio about 12:1 and fuel velocity through the hole about 250m/s.

Viscosity
May be defined as internal fluid molecular friction which causes a resistance to flow.

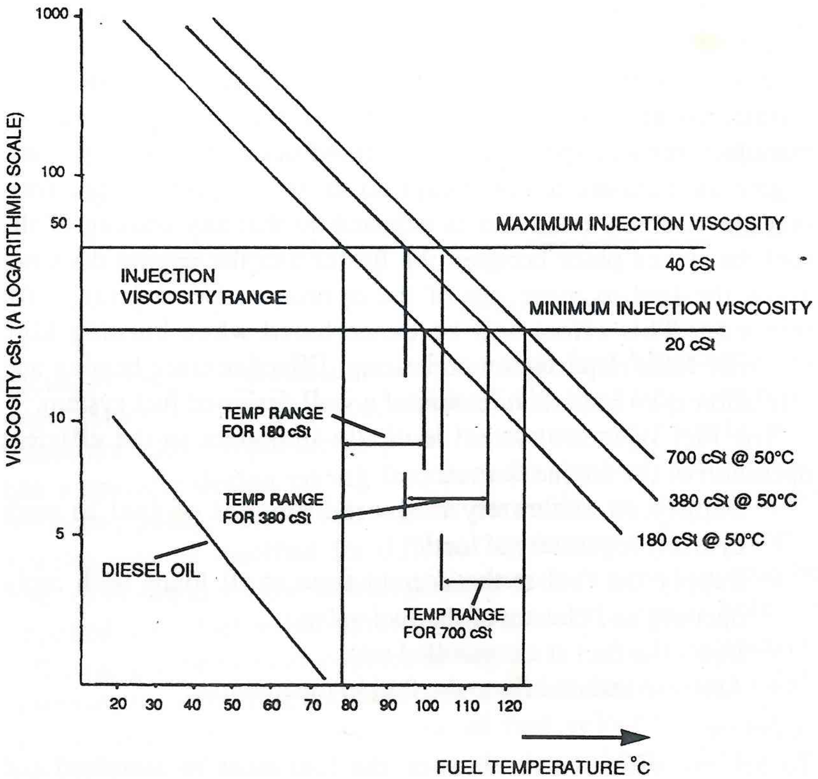
Pre-heating
With the almost universal use of residual fuel in marine diesel engines it is important, to ensure optimum fuel injection, that correct pre-heating is carried out. If the temperature of the fuel is too low then the viscosity will be high resulting in higher injection pressure and reduced atomising performance, excessive penetration and possible impingement on internal surfaces. Too high a temperature also has adverse effects by reducing penetration and causing deposits to be left on nozzle tip affecting atomisation. The relationship between viscosity and temperature is shown in Fig. 62. Careful control of fuel temperature is required to ensure that the fuel viscosity at the engine fuel rail is inside the range specified by the manufacturers. It is modern practice to utilise viscosity controllers to ensure that the fuel temperatures is maintained at the correct level. Although the viscosity controller manufacturer will specify that it be fitted in close proximity to the engine the builders do not always do so. It is important, therefore, that the viscosity controller is adjusted so that any cooling of the fuel that takes place between the heater and the engine does not allow the fuel to move out of the optimum viscosity range for injection. This effect will be exacerbated when burning high viscosity fuels requiring much heating. Effective trace heating and insulation is an important feature of a well designed fuel system.

The fuel injection system is vitally important to the efficient operation of the engine. It must:

- Supply an accurately measured amount of fuel to each cylinder regardless of load.
- Supply the fuel at the correct time at all loads with rapid opening and closing of the fuel valve.
- Inject the fuel at a controlled rate.
- Atomise and distribute the fuel in the cylinder.

Injection
To achieve effective combustion the fuel must be atomised and then distributed throughout the combustion space. It is the function of the fuel valve to accomplish this. Most fuel valves on marine diesel engines are of the hydraulic type. Fig. 63. The opening and closing of this type of valve is controlled by the fuel pressure delivered by the fuel pump which acts on the needle in the lower chamber. When the force is sufficient to overcome the spring, the needle lifts.

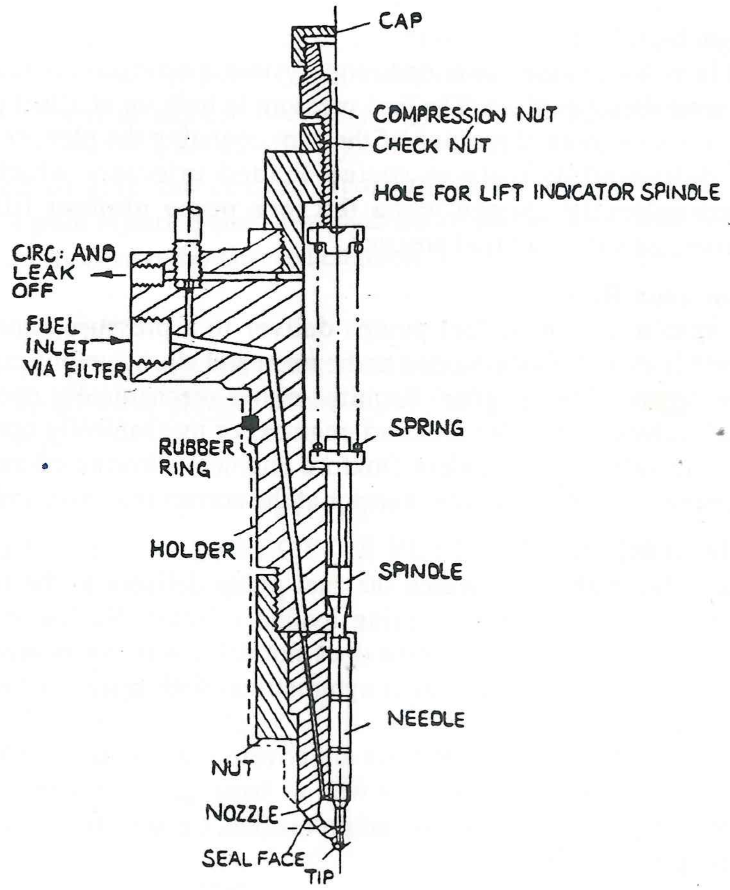
FIG 62
VISCOSITY TEMPERATURE CHART FOR MARINE FUELS.



Full lift occurs quickly as the extra area of the needle seat is exposed after initial lift. The full action of lift is limited by the needle shoulder which halts against a thrust face on the holder. The injector lift pressure varies with the design but may be about 140 bar average (some designs 250 bar).

A fuel valve lift diagram for such an injector is given in Chapter 1. By removal of the spring cap the valve lift indicator needle can be assembled in the adaptor. This particular design as sketched is not cooled itself but is enclosed in an injector holder with seal (face to face) at tapered nozzle end with rubber ring at the top.

FIG 63
FUEL VALVE INJECTOR (HYDRAULIC)



Coolant is circulated in the annular space between the injector holder and the holder itself. Direct cooling of the fuel valve as an alternative to this is easily arranged. Coolant connections on the main block would supply and return through drillings similar to that shown for fuel. The choice of oil or water for cooling depends on the engine and valve design and is also affected by the type of fuel. With hot boiler oil it is necessary to cool right to the injector tip so as to attempt to keep metal temperatures below 200°C. Hydraulic fuel valves usually have a lift of about 1 mm and the action is almost instantaneous.

There are three broad types of injection:

- Jerk injection.
- Common rail.
- Timed injection.

Jerk injection

This is by far the most common system employed on modern marine diesel engines. The fuel pressure is built up at a fuel pump in a few degrees of rotation of the cam operating the plunger. Fuel is delivered directly to spring loaded injectors which are hydraulically opened when the jerk pump plunger lift has generated sufficient fuel pressure.

Common Rail

A system in which fuel pumps deliver to a pressure main and various cylinder valves open to the main and allow fuel injection to the appropriate cylinder. Requires either mechanically operated fuel valves (*e.g.* older Doxford engines) or mechanically operated timing valves (*e.g.* modern Doxford engines) allowing connection between rail and hydraulic injector at the correct injection timing.

Timed Injection

As defined above in which the fuel pump delivers to the timing valve and thence to the spring loaded injector. No lost motion clutch is required as the cam does not drive a pump plunger but operates a valve. The cam is symmetrical with respect to engine dead centre.

Note: Many aspects of fuels are covered in Volume 8 (Chapter 2) and revision of oil tests as well as basic definitions relating to specific gravity, Conradson carbon residue, Cetane number, etc., is strongly advised.

Indicator Diagrams

Details have been given of some typical indicator diagrams showing engine faults in Chapter 1. Aspects of fuel injection faults included late and early injection (draw card), fuel valve lift diagrams, etc., as well as related details such as compression cards. Two further typical faults are as illustrated in Fig. 64.

Afterburning will show as indicated with a loss of power, increased cylinder exhaust temperature and possible discolouration of exhaust gases. Fuel restriction at filters, injectors, etc. or due to incorrect viscosity will result in a loss of power and reduced maximum pressure.

FUEL PUMPS

General

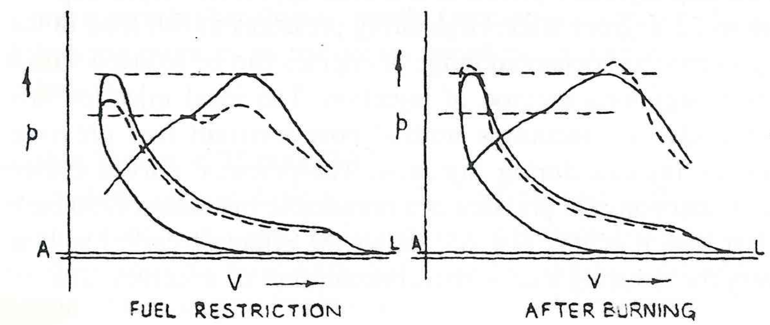
The physical energy demands of injection are great. Typical requirements include delivery of about 100ml of fuel in 1/30 second at 750 bar so as to atomise over an area of $40m^2$. A peak energy input can reach 230 kW. A short injection period at high pressure, so placed to give the desired firing pressure, is necessary. Generally pilot injection and slow injection of charge is difficult to arrange for modern turbo-charged engines.

Quantity control

The amount of fuel injected per stroke is usually accomplished by varying the effective plunger stroke of the fuel pump. This may be achieved by:

FIG 64

EFFECTS OF DEFECTIVE FUEL INJECTION



1. Varying the beginning of delivery.
2. Varying the end of delivery.
3. Varying the beginning and the end of delivery.

Control has been arranged for regulated end of effective stroke by helical groove with constant beginning of injection, as in the well known Bosch principle, described later. This method is regularly utilised for auxiliary engines and gives fuel injection early in the cycle at light load which gives higher efficiency but also leads to higher firing pressures. It has also been utilised with large direct coupled engines (*e.g.* B.& W.). Control in valve type pumps for large engines was usually with constant end and regulation of the start of injection by varying the suction valve closure (*e.g.* Sulzer and Doxford). Engine performance at low load with later injection is a compromise between economy and firing pressure. With turbo-charged engines the disadvantage of the constant end pump control is more noticeable as reduced firing pressure and efficiency is more marked at low loads due to reduced turbo-charger delivery and pressure.

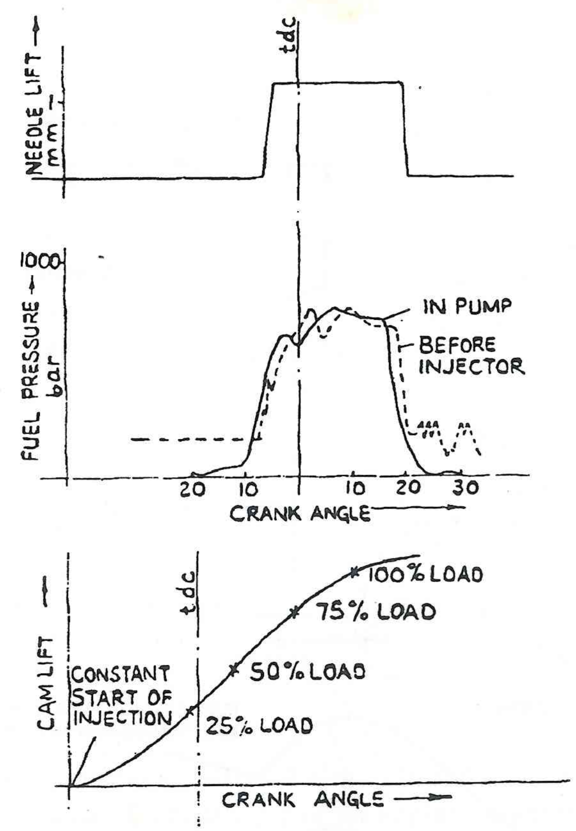
Because of the limitations of varying the fuel quantity delivered by only varying the beginning of delivery Sulzer redesigned their fuel pumps to include a suction valve and spill valve. Initially the spill valve only was controlled resulting in constant beginning with variable end of delivery. Latterly, however, in the interests of fuel economy and in common with other manufacturers, both valves are controlled to give variable beginning and end of delivery.

Injection Characteristics

The diagram given in Fig. 65. illustrates some features of fuel injection based largely on Sulzer RND practice.

The fuel valve injector lift diagram is shown with lift of about 1·3 mm and injection period at full load approximately 6 degrees before to 22 degrees after. High firing pressures at full load in the high powered turbocharged range of engines can be reduced with a constant beginning method of injection. The ideal injection law corresponds to a rectangle with almost constant fuel pressure before the injector during injection. The practical curves shown show almost constant pressure at a reasonable maximum (750 bar). The last sketch of this Fig. 65. shows the effect of earlier spill at delivery for reducing load with a constant start of injection. The

FIG 65
INJECTION CHARACTERISTICS



plunger at first moves very rapidly to build up pressure but is slowed during injection so giving minimum injection pressures at full load. The constant beginning of injection gives a flat combustion pressure over the full power range. At low loads the fuel pressure is higher than is usually the case because the plunger is delivering more in its maximum speed range, this gives better atomisation.

Variable Injection Timing (VIT)

The previous section dealt with only variable and with constant beginning of injection. It is modern practice for economy considerations to now vary both the beginning and end of injection. With constant beginning of injection the maximum firing

FIG 66
FUEL SAVINGS AVAILABLE BY UTILISING VARIABLE INJECTION TIMING (VIT)

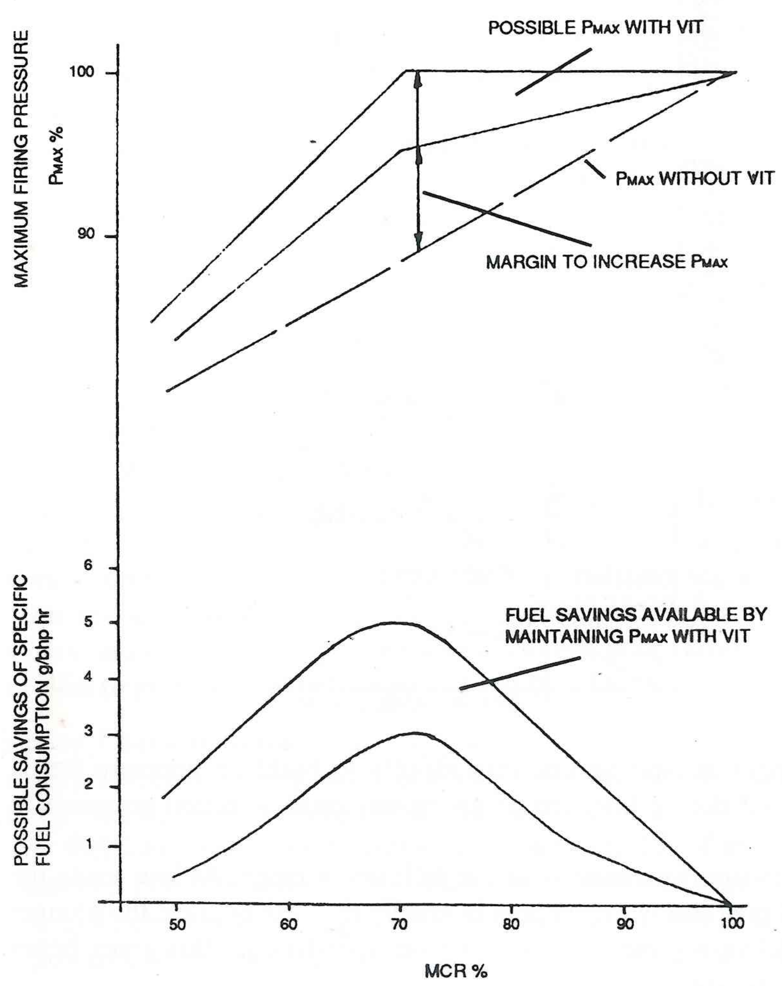
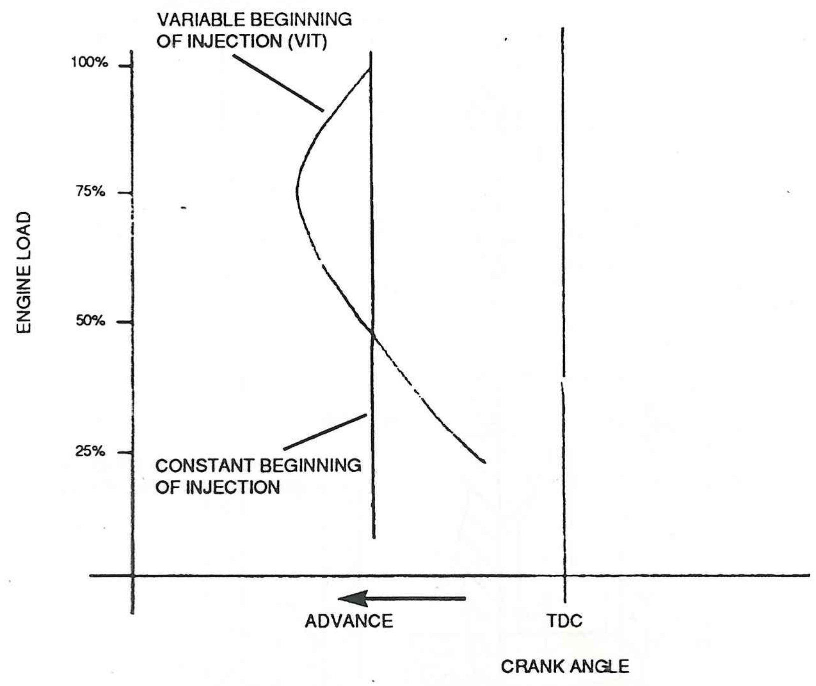


FIG 67
INJECTION TIMING VARIATION WITH ENGINE LOAD



pressure of the engine will fall almost linearly as the power of the engine is reduced. The b.m.e.p of the engine, however, reduces at a slower rate and since the thermal efficiency of the engine varies as the ratio of P_{MAX}/P_{MEP} then a reduction of firing pressure will result in a reduction of thermal efficiency of the engine. In order that the thermal efficiency and hence the specific fuel consumption can be maintained at optimum it is therefore necessary to maintain maximum firing pressures as the engine load is reduced. This is accomplished by advancing the timing of the fuel injection as the engine load is reduced Fig. 66. The advancement of the injection timing continues until about 65%-70%, thereafter the injection is retarded. Fig. 67. It can be seen that at about 25% engine load injection is retarded in relation to full load timing. This will reduce the "diesel knock" at low engine loads that is sometimes experienced on engines without variable injection timing [VIT].

FIG 68
BOSCH TYPE FUEL PUMP OPERATION

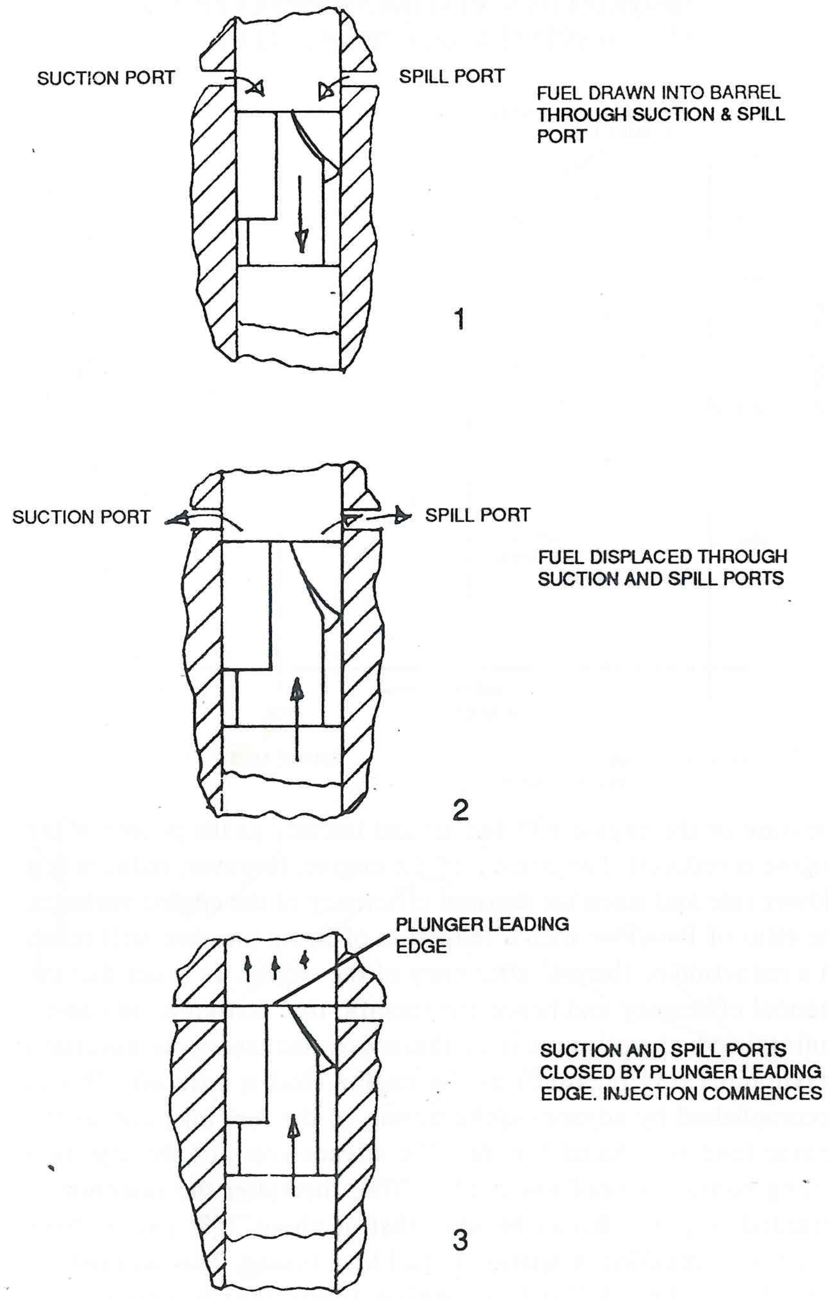
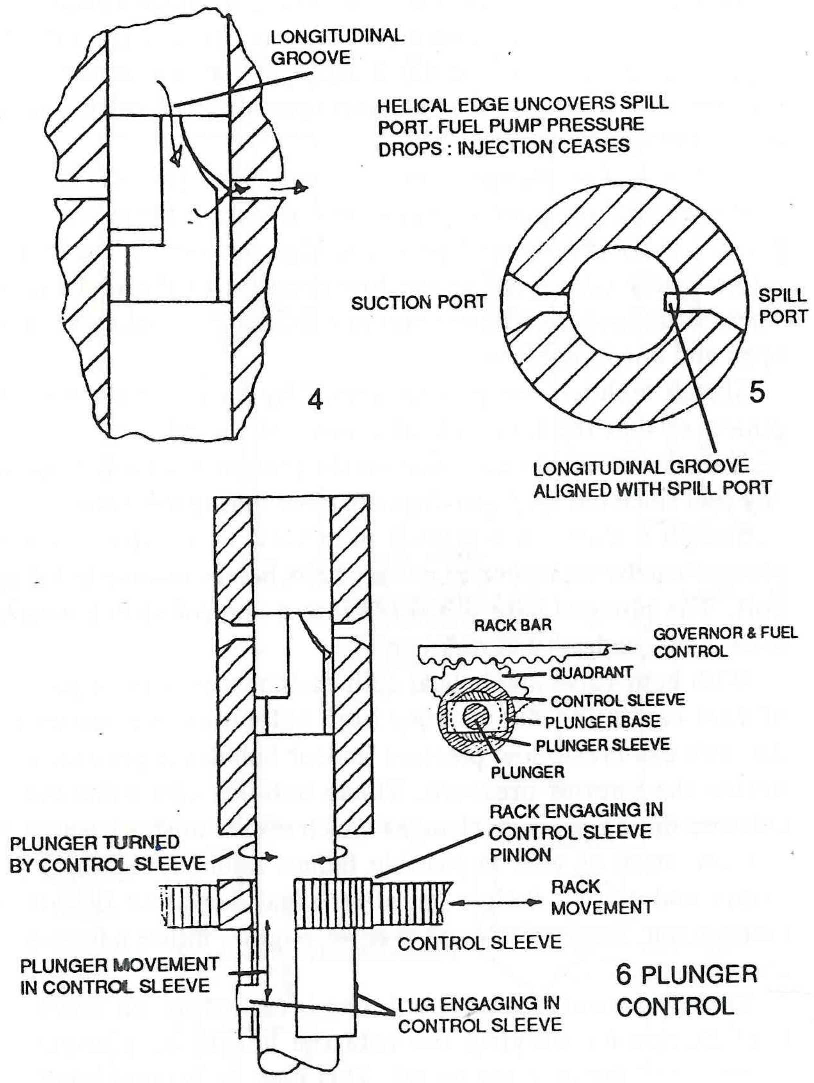


FIG 68
LONGITUDINAL GROOVE & PLUNGER CONTROL



Bosch Jerk Pump Principle

Fig. 68. shows a Bosch type fuel pump set at approximately 75% load.

The sketch numbered 1 show the plunger moving down. The pressure in the barrel falls and as the suction and spill ports open to the fuel rail the fuel flows into the barrel.

In Sketch 2 the plunger is moving upwards. The fuel is displaced from the barrel through the spill and suction ports. This displacement will continue until the plunger completely covers both ports.

Sketch 3 shows the plunger continuing to move upwards and just covering the spill and suction ports. This is the effective beginning of delivery and any further upward movement of the plunger will pressurise the fuel and open the fuel valve injecting fuel to the engine.

Sketch 4. The plunger continues to move upwards. Injection continues until the point when the helical edge of the groove on the plunger uncovers the spill port. The high pressure in the barrel is immediately connected to the low pressure of the fuel suction. There is no longer sufficient pressure to keep the fuel valve needle open and injection ceases.

Sketch 5 shows the plunger turned by means of the rack and pinion so that the longitudinal groove of the plunger is aligned with the spill port. In this position the plunger is unable to deliver any fuel since the spill port does not close during the cycle.

Sketch 6 shows a sectional plan view to illustrate how the plunger can be rotated so as to vary helix height relative to the spill port. The plunger base is slotted into a control sleeve which is rotated by quadrant and rack bar.

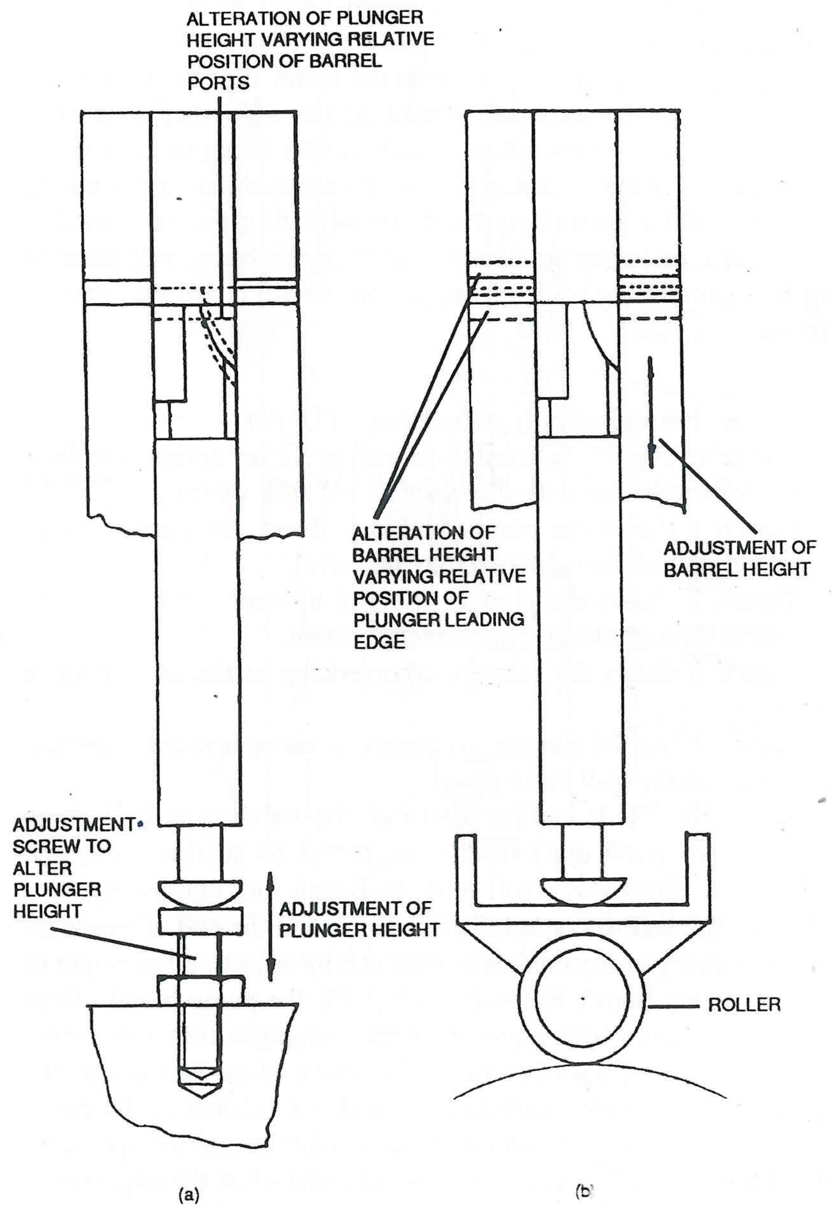
With both valve and helical spill designs there can be problems of fuel cavitation due to very high velocities. Velocities near 200 m/s can create low pressure vapour bubbles if pressure drops below the vapour pressure. These bubbles can subsequently collapse during pressure changes which results in shock waves and erosion attack as well as possible fatigue failure. A spring loaded piston and orifice design can absorb and damp out fluctuation. Götaverken, Stork, MAN and B & W. engines utilise a form of the above pump.

The adjustment of injection timing is carried out on Bosch type fuel pumps by varying the relative height of plunger and suction/spill ports in the barrel. This may be accomplished in a number of ways:

1. Adjusting the height of plunger relative to the barrel. Fig 69a.

2. Adjusting the height of the barrel relative to the plunger. Fig 69b.

FIG 69
ADJUSTMENT OF FUEL INJECTION TIMING BY VARYING RELATIVE POSITION OF PLUNGER AND BARREL



Adjustment of plunger height can be accomplished in some installations by adjusting the cam follower. Lowering the plunger has the effect of retarding the injection. Raising the plunger advances the injection.

In earlier B & W. designs adjustment of injection timing was carried out by raising the fuel pump barrel in relation to the plunger. In this design the fuel pump top flange has an external threaded portion projecting towards the barrel Fig. 70. This thread matches with the external thread of the adjusting ring. The adjusting ring has external gear teeth cut on its upper part which are engaged by the adjusting pinion. To adjust the injection timing the pinch bolts are released and the adjusting pinion turned to either raise or lower the barrel. Lowering the barrel will advance the injection timing while raising the barrel will retard the injection timing.

Jerk Fuel Pump [valve type fuel pump] Detail

The detail in Fig. 71. is based on an earlier Sulzer design. The fuel-pump delivery is controlled by suction and spill valves.

Sketch 1 shows the plunger moving down, the suction valve open and fuel-oil being drawn into the barrel.

Sketch 2 shows the plunger moving upwards and fuel being displaced through the still open suction valve.

Sketch 3 shows the delivery commencing as the suction valve closes.

Sketch 4. As the plunger continues to move upwards injection ceases when the spill valve opens.

From Fig. 71. it can be seen that the valve type fuel pump design lends itself quite readily to control on suction valve and spill valve. However, scroll type, or Bosch fuel pumps require a different method to control the beginning and the end of injection. The beginning of injection is carried out by adjusting the height of the fuel pump barrel. Referring to Fig. 72. the pump barrel has an outside threaded lower portion which engages into the timing guide operated by a toothed rack. Movement of the rack causes the pump barrel to move vertically up or down relative to the pump plunger. In this way the moment the plunger covers the spill port, and so commences injection, can be adjusted while the engine is in operation. The pump barrel is prevented from turning by a locating

FIG 70
ADJUSTMENT OF INJECTION TIMING BY RAISING OR LOWERING FUEL PUMP BARREL

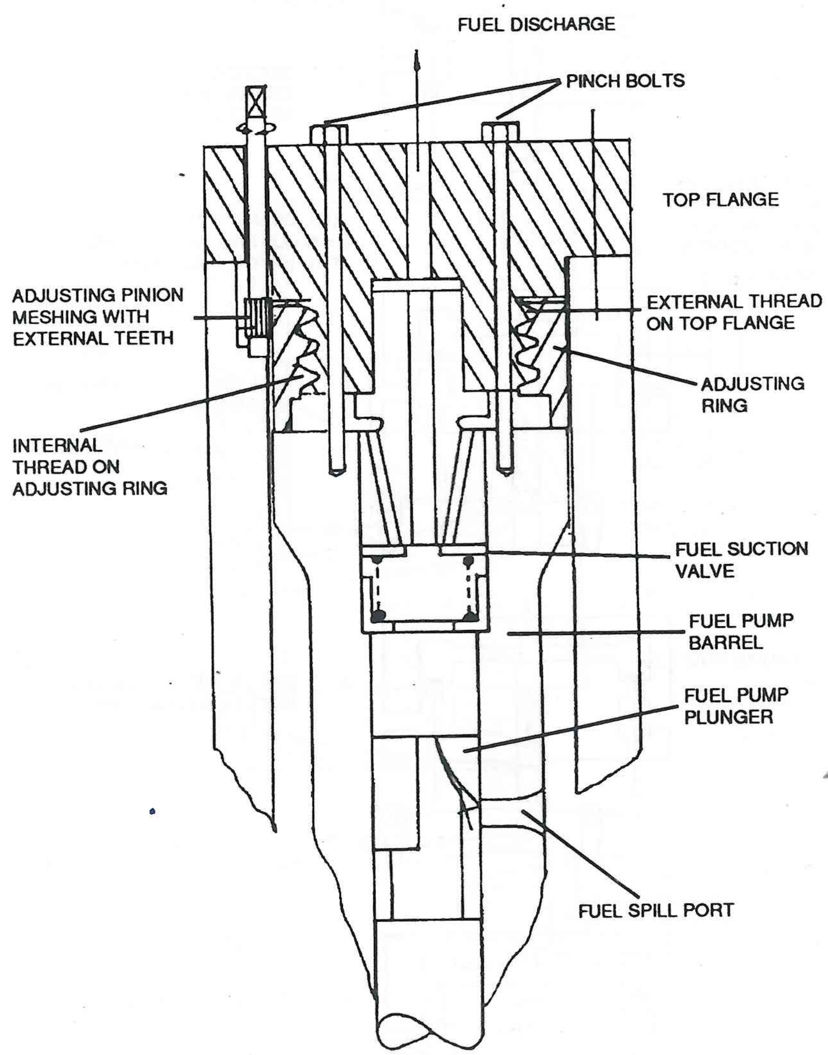


FIG 71
SULZER VALVE TYPE FUEL PUMP

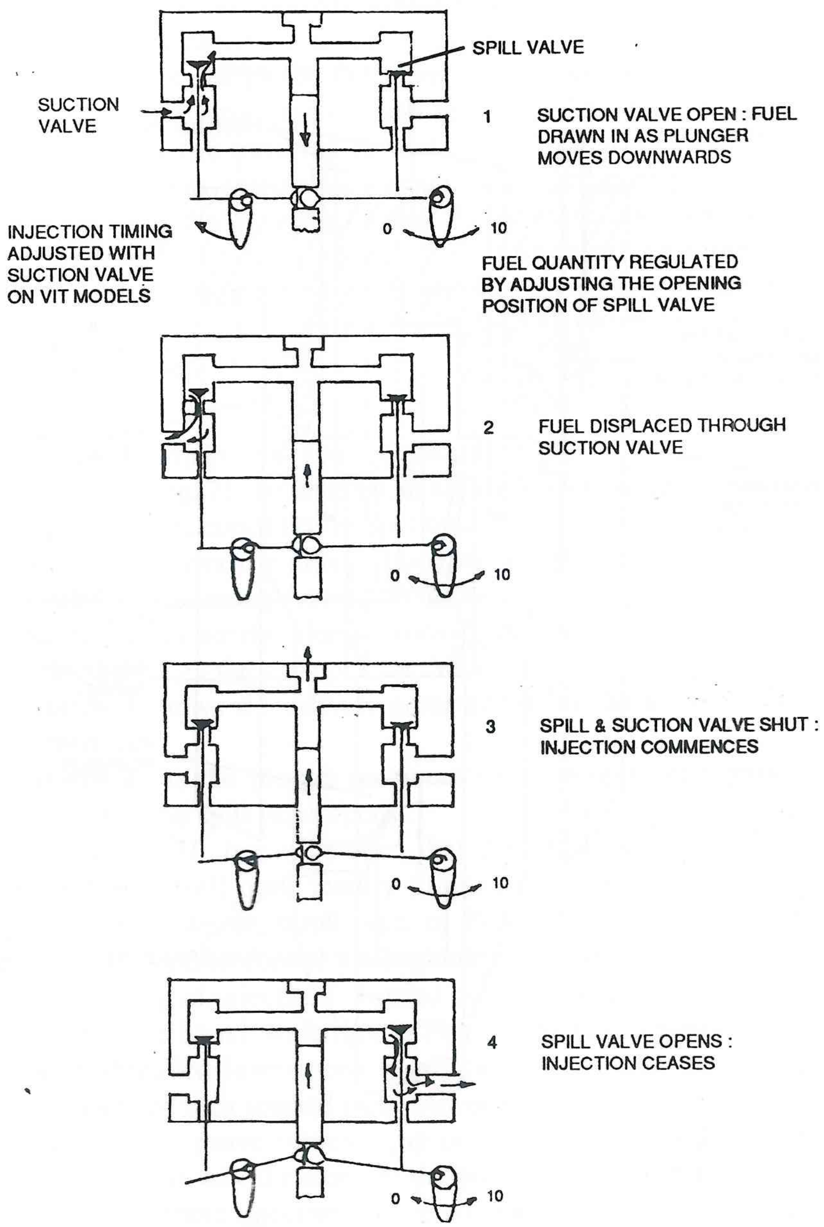
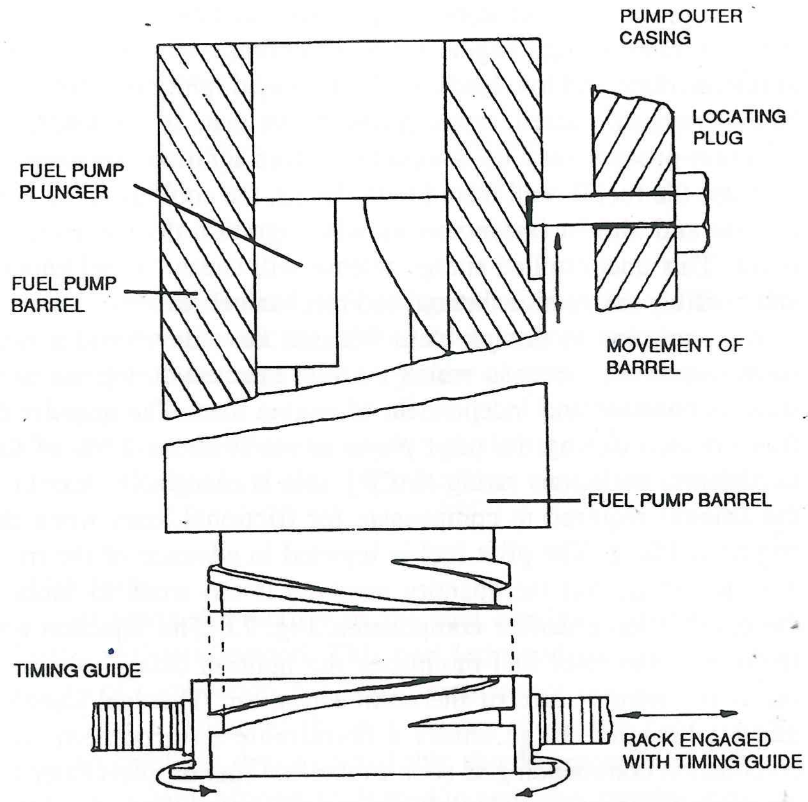


FIG 72
VIT BOSCH TYPE FUEL PUMP



plug. The end of injection, and therefore the quantity of fuel delivered, is regulated by rotating the plunger which varies the position of the helix edge relative to the spill port.

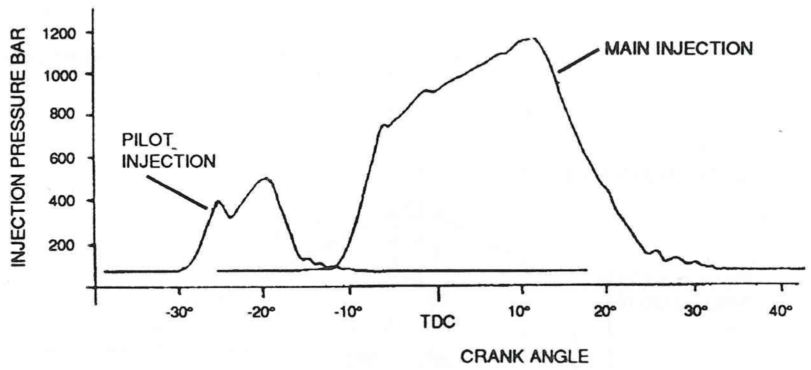
Two Stage Fuel Injection

Efforts are continually being made to improve the reliability and economy of medium speed engines which operate on heavier grades of fuel. One way of achieving this is by having a carefully controlled, reliable combustion process. This requires good atomisation with short injection periods but this results in high injection rates at high engine loads with commensurate high rates at intermediate and low loads where increased ignition delays may be experienced. Indeed, research has shown that, at low loads, the injection process may be completed before ignition commences. Because the fuel is well mixed with the air, the combustion is very intense and almost instantaneous when ignition does eventually occur. This uncontrolled energy release will cause "diesel knock" and possibly destructive thermal and mechanical stresses.

As a solution to this problem Wärtsilä have developed a two-stage injection process in which the fuel injected during the pilot stage is constant and independent of engine load. The quantity of fuel injected during the pilot phase is set at about 2·6% of the maximum continuous rating [MCR], this is marginally less than the amount required to compensate for frictional loses when the engine is idling. The pilot fuel is injected in advance of the main injection phase but the quantity involved is too small to damage the combustion chamber components. Fig. 73. The injection and ignition of the pilot fuel minimises the ignition delay because it raises the temperature of the combustion air. The fuel injected during the main stage enters a favourable environment with combustion commencing as soon as the first fuel droplets enter the combustion chamber. This eliminates the possibility of unburnt fuel being stored in the combustion chamber and hence the destructive uncontrolled release of energy.

Both fuel valves are supplied by the same fuel pump. The fuel pump, however, has two plungers to supply pilot and main injection. The main plunger of this fuel pump is of the conventional scroll, or Bosch type. The pilot plunger is positioned above the main plunger, but since the quantity of fuel injected in

FIG 73
PILOT AND MAIN INJECTION AGAINST CRANK ANGLE



the pilot stage is constant this plunger has no helix. As the pilot plunger covers the suction/spill port injection commences and ceases as the lower edge of the plunger uncovers the suction/spill port. Fig. 74.

This arrangement has the advantage that the emission of NO_x [oxides of nitrogen] are reduced and allows the use of low Cetane fuels.

Fuel Systems

In recent years the quality of the fuel available to the marine industry has deteriorated. This had led inevitably, not only to problems of combustion, but also to problems of storage.

To alleviate some of these problems the entire fuel system should be carefully designed and this should commence with the bunkering system. Modern fuels tend to be high viscosity and may have a high pour point so it is important that at the completion of bunkering that the fuel drains freely into the bunker tanks. If loading bunkers in cold climates it may be necessary to include insulation on the exposed bunker lines. An indication of a high pour point fuel may be a high loading temperature: the supplier ensuring that the fuel is easily pumped on board. If a waxy fuel is suspected then a pour point test should be carried out.

FIG 74
TWO STAGE FUEL PUMP

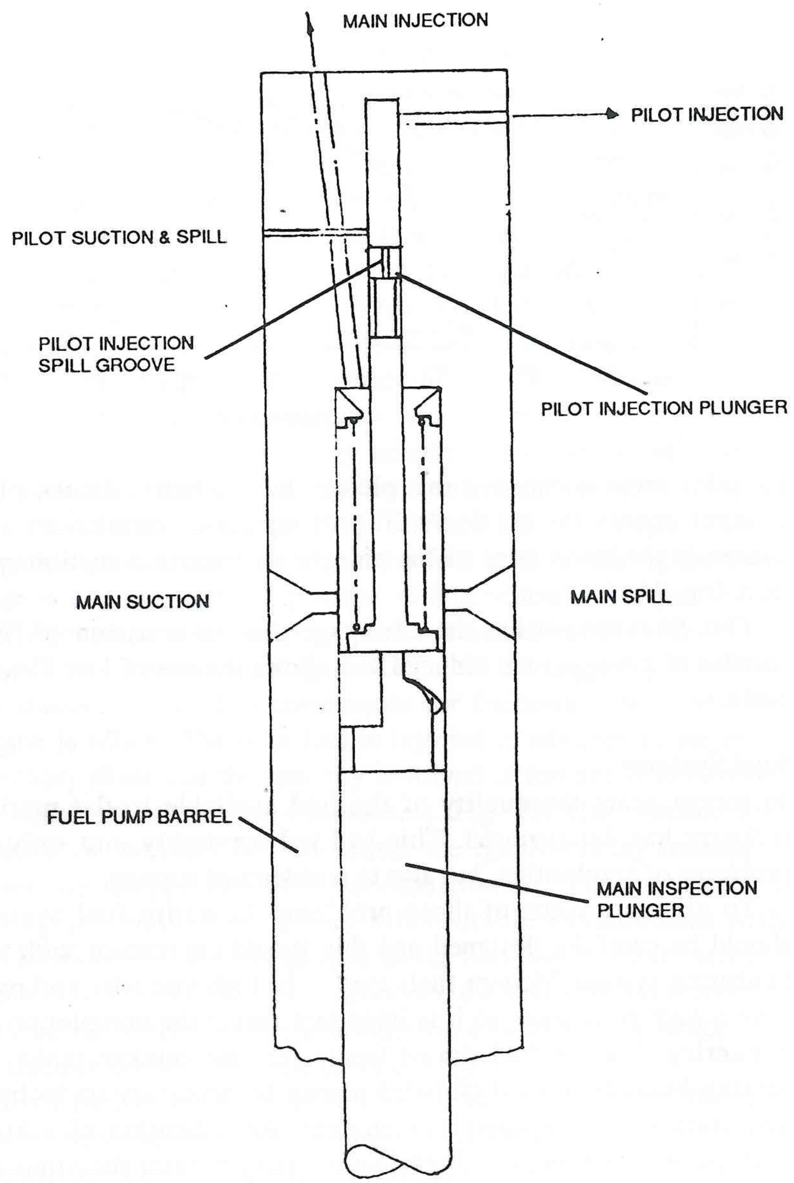
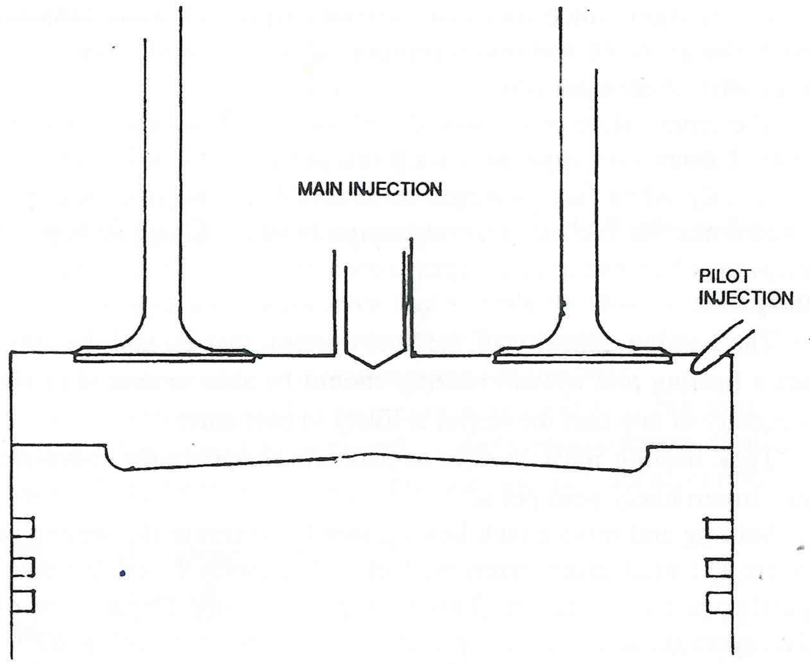


FIG 75
POSITION OF MAIN AND PILOT INJECTION IN CYLINDER



Due to the problems associated with incompatibility fuels from different sources should not be mixed. The importance of the segregation of fuels from different sources cannot be overstated and should be achieved, wherever possible, by transferring remaining fuel into smaller tanks prior to bunkering in order that the total quantity of fuel loaded can be received in empty tanks. Even if a vessel is equipped with adequate storage to ensure segregation mixing may occur in the settling and service tanks when fuels are changed over. If compatibility problems are suspected then fuel change-overs should be accomplished by running down the settling tank before admitting the next, possibly, incompatible fuel. Two service and two settling tanks would be the ideal situation, since fuel change-overs would be accomplished with the absolute minimum mixing. Designers and shipowners may wish to consider this.

The temperature of the stored fuel must be monitored to ensure that it does not approach its pour point. This is important, especially when fuel is stored in double bottoms, since it is not uncommon for fuels to have a pour point of 25°C and so become unpumpable even at temperatures that can be considered temperate.

The heating capacity of the fuel system, that is, tank heating, trace heating and system heating should be able to deal with the viscosity of any fuel the vessel is likely to encounter.

Tank heating must be able to maintain temperatures above the maximum likely pour point.

Settling and service tank heating should maximise the settling of water and solid matter from the fuel and maintain the correct post-purification temperature. This is important since Department of Transport regulations require that fuel is stored at, or below 60°C. However, the purification and clarification temperatures of high viscosity fuels may be substantially higher than this; 100°C for example. To comply with this regulation a post-purifier FUEL COOLER may be required.

When operating with high viscosity fuels it is necessary to employ high rates of heat transfer during fuel heating. This could cause thermal cracking of the fuel leading to carbon deposits on the heating surfaces resulting in reduced heating capacity. To maintain optimum heat transfer and heating steam consumption there should be a facility to enable the oil side of the heater to be cleaned periodically by circulating with a proprietary carbon remover. A Typical Fuel System is shown in Fig. 76.

Fuel management

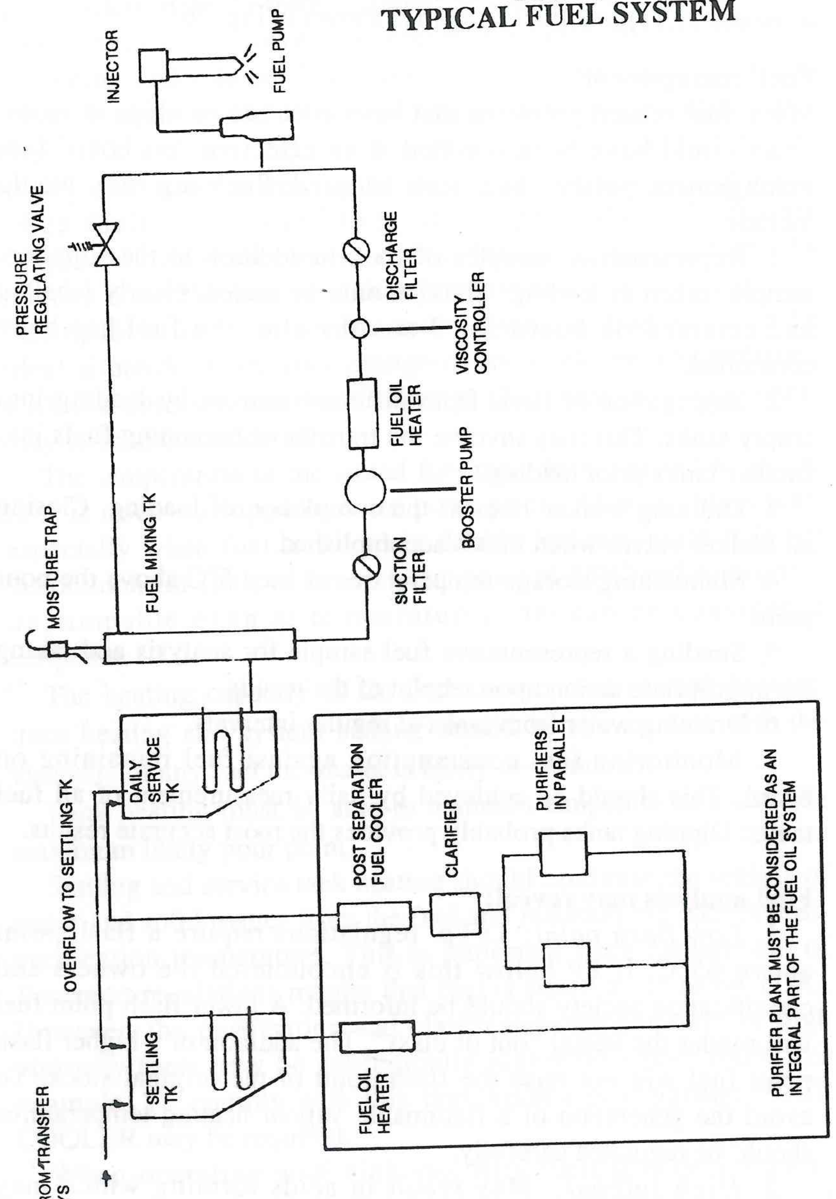
Many fuel related problems that have occurred on ships in recent years could have been avoided if an effective "on board fuel management policy" had been adopted. Such a policy would include.

1. Representative samples of fuel, in addition to the suppliers sample, taken at loading. These should be sealed, clearly labelled and retained on board for 3 months after the fuel has been consumed.
2. Segregation of fuels from different sources by loading into empty tanks. This may involve the transfer of remaining fuels into smaller tanks prior loading.
3. Draining bunker lines at the completion of loading. Closing all bunker valves when this is accomplished.
4. Maintaining storage temperatures at least 5°C above the pour point.
5. Sending a representative fuel sample for analysis and taking the appropriate action upon receipt of the results.
6. Draining water from tanks at regular intervals.
7. Monitoring fuel consumption against fuel remaining on board. This should be achieved by daily measurement of all fuel tanks: Dipping tanks probably provides the most accurate results.

Fuel analysis may reveal:

1. *Low flash point:* D.Tp. regulations require a flash point above 65°C. If FP below this is encountered the owners and classification society should be informed. A lower flash point fuel will render the vessel "out of class". The addition of a higher flash point fuel will not raise the flash point of the original stock. To avoid the generation of a flammable vapour heating temperatures should be regulated carefully.
2. *High sulphur:* May result in acids forming which may deplete the TBN of cylinder/system oil. Engine may require a higher TBN oil.

FIG 76
TYPICAL FUEL SYSTEM



3. *High water content:* May separate when heated. May, however, form a stable emulsion which is difficult to separate without the addition of emulsion breaking chemicals. If water contamination is salt water, not uncommon in the marine environment, serious problems associated with sodium-vanadium corrosion and turbo-charger fouling may be experienced. Water contamination also introduces the risk of bacteria into the fuel. Bacterial growth can occur at the oil/water interface which if allowed to proliferate can cause blockage of filters and fuel system. The problem of bacterial, or microbial attack, is greater in fuel which is unheated, especially diesel oil, since the temperatures involved when heating high viscosity fuels will pasteurise the fuel and thus kill off bacteria. Since prevention is better than cure draining the water from the oil is probably the best course of action.

4. *High vanadium:* May cause high temperature corrosion. The use of an ash modifying chemical additive to maintain the vanadium oxides in a molten state will prevent adhesion to high temperature components.

5. *Instability and incompatibility:* Instability refers to tendency of the fuel to produce a sludge by itself. Incompatibility is the tendency of the fuel to produce a sludge when blended with other fuels. These sludges form when the asphaltene content of the fuel can no longer stay in solution and so precipitates out, sometimes at a prodigious rate. The deposited sludge blocks tank suctions, filters, and pipes and quickly chokes purifiers. In engines the blockage of injector nozzles, late burning and coking can result in damage to pistons, rings and liners.

6. *High aluminium content:* This contamination is a result of carry over of "catalytic fines" from the refining process of the initial feedstock oil. These "fines" are an aluminium compound ranging in size from 5 mm to 50 mm and are extremely abrasive. Very low levels of aluminium indicate the presence of catalytic fines in the fuel which, if used, will lead to high levels of abrasive wear in the fuel system, piston, rings and liner in an extremely short period of time. 30 ppm of aluminium is generally considered as the maximum allowable level in fuel oil bunkers before purification. Because of the small size of these compounds they are difficult to remove completely by centrifuge. The purification

plant, in correct operation, will reduce the aluminium content to about 10 ppm before it is used in the engine. It has been found that if the aluminium content is above 30 ppm difficulties will be experienced attaining a safe level of 10 ppm after purification.

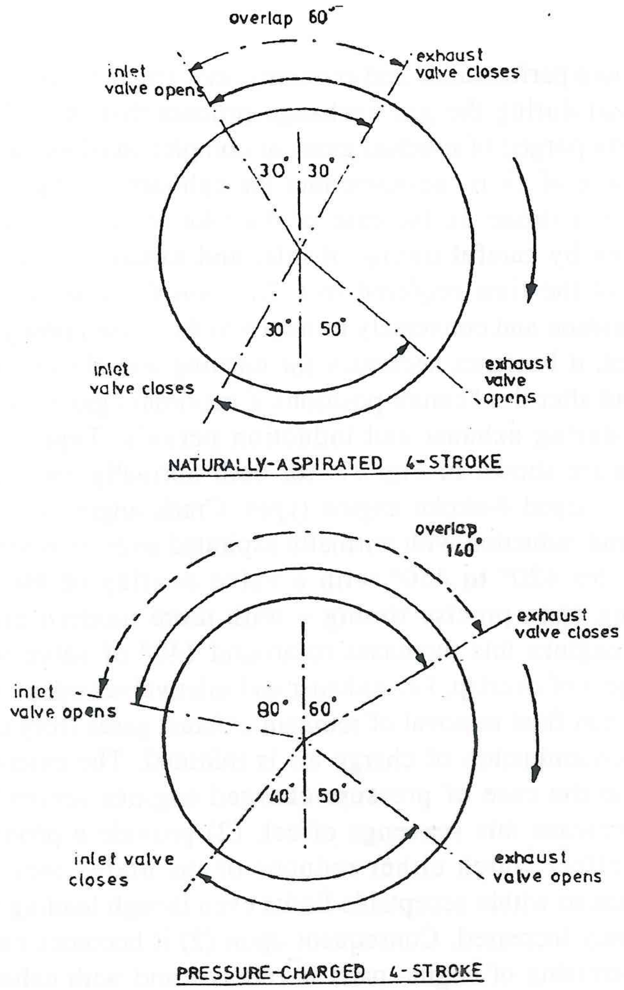
CHAPTER 4

SCAVENGING AND SUPERCHARGING

If maximum performance and economy, etc. are to be maintained it is essential during the gas exchange process that the cylinder is completely purged of residual gases at completion of exhaust and a fresh charge of air is introduce into the cylinder for the following compression stroke. In the case of 4-stroke engines this is easily carried out by careful timing of inlet and exhaust valves where, because of the time required to fully open the valves from the closed position and conversely to return to the closed position from fully open, it becomes necessary for opening and closing to begin before and after dead centre positions if maximum gas flow is to be ensured during exhaust and induction periods. Typical timing diagrams are shown in Fig. 77. for both normally aspirated and pressure charged 4-stroke engine types. Crank angle available for exhaust and induction with normally aspirated engines is seen to be of the order 420° to 450° with a valve overlap of 40° to 60° depending upon precise timing – with more modern pressure-charged engines this increases to around 140° of valve overlap. Basic object of overlap, *i.e.* exhaust and inlet valves open together, is to assist in final removal of remnant exhaust gases from cylinder so that contamination of charge air is minimal. The extension of overlap in the case of pressure-charged engines serves to: (1) further increase this scavenge effect, (2) provide a pronounced cooling effect which either reduces or maintains mean cycle temperature to within acceptable limits even though loading may be considerably increased. Consequent upon (2) it becomes clear that thermal stressing of engine parts is relieved and with exhaust gas turbocharger operation prolonged running at excessively high temperatures is avoided. This latter would have an adverse effect on materials used in turbocharger construction and could also contribute toward increased contamination.

In some cases an apparent anomaly exists between the temperature of exhaust gas leaving the cylinder and the temperature at inlet to turbocharger, being as much as 90° higher. This is

FIG 77
TYPICAL TIMING DIAGRAMS



partially explained by the fact that over the latter part of the gas exchange process the relatively cold scavenge air will have a depressing effect on the temperature indicated at cylinder outlet which will tend to indicate a mean value over the cyclic exchange.

More probably the increase may be largely attribute to change of kinetic energy into heat energy and an approximately adiabatic compression of the gas column between cylinder and turbine inlet.

2-Stroke Cycle Engines

With only one revolution in which to complete the cycle the time available for clearing the cylinder of residual exhaust gases and recharging with a fresh air supply is very much reduced. Of necessity the gas exchange process is carried out around b.d.c. where the positive displacement effects of the piston cannot be exploited as is the case with the 4-stroke cycle. The total angular movement seldom exceeding 140° compared to well in excess of 400° with 4-stroke operation gives some indication of the need for high efficiency scavenging processes if cylinder charge is not to suffer progressive contamination and subsequent loss of performance with increased temperature and thermal loading. Prior to the introduction of turbocharging to 2-stroke machinery this necessitated a low degree of pressure-charging of 1·1 to 1·2 bar to ensure adequacy of the gas exchange process.

The scavenging of 2-stroke engines is generally classified as: (1) uniflow or longitudinal scavenge, (2) loop and cross scavenge. Fig. 78. Because of the simplicity of the arrangement uniflow scavenge as employed in poppet-valve or opposed piston type engines is generally considered as being the most efficient. In this case charge air is admitted through ports at the lower end of the cylinder and as it sweeps upwards toward the exhaust discharge areas, almost complete evacuation of residual gases is obtained. By suitable design of the scavenge ports or the provision of special air deflectors the incoming charge air can be given a swirling motion which intensifies the purging effect and also promotes the degree of turbulence within the charge which is required for good combustion when fuel injection takes place.

Cross and loop scavenge have both exhaust and scavenge ports arranged around the periphery of the lower end of the liner and in so doing eliminate the need for cylinder head exhaust valves etc and their attendant operating gear. This considerably simplifies engine construction and can lead to a reduction of maintenance. Because of simplified cylinder head construction the cylinder combustion space can be designed for optimum combustion conditions. Generally however, the scavenging efficiency is somewhat lower than with the uniflow system due to the more complex gas-air interchange and the possibility of charge air passing straight to exhaust with little or no scavenging effect. Careful attention to port design does however considerably reduce this problem.

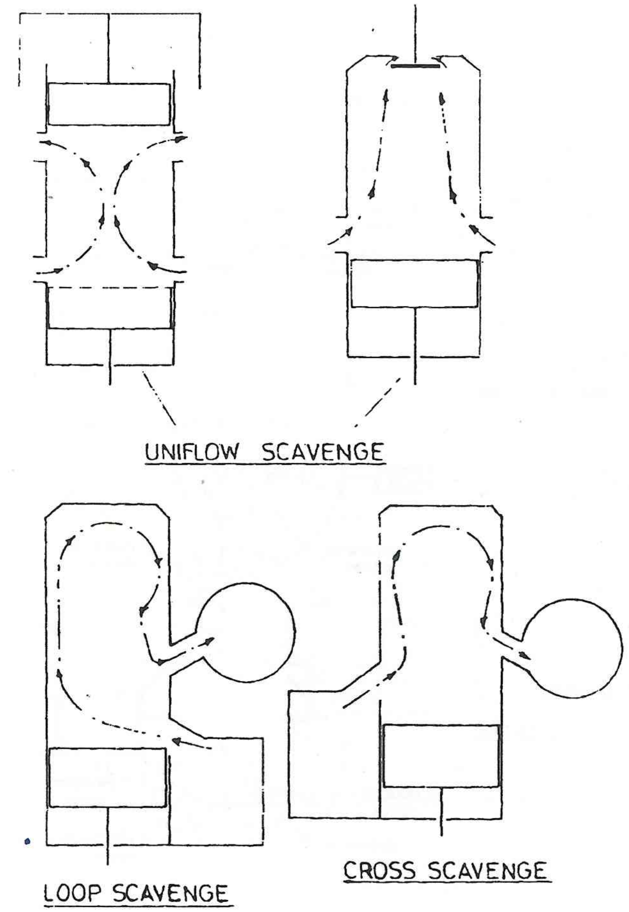
The gas exchange process itself may be divided into three separate phases: (1) blowdown, (2) scavenge and (3) post-scavenge.

During blowdown the exhaust gases are expelled rapidly – the process being assisted by amply dimensioned ports or valves arranged to open rapidly. At the end of this blowdown period when the scavenge ports begin to uncover, the cylinder pressure should be at or below charge air pressure so that the scavenge process which follows, effectively sweeps out the remaining residual gases. With scavenge ports closed the post scavenge period completing the gas exchange process should ensure that exhaust discharge areas close as quickly as possible to prevent undue loss of charge air so that the trapped air at beginning of compression has the highest possible density. Although some loss of charge air is unavoidable it should be borne in mind that the air supply is considerably in excess of that required for combustion and the cooling effect of the air passing through the system has the result of keeping mean cycle temperatures down so that service conditions are less exacting.

The increased cylinder pressures encountered with modern turbocharged machinery may result in exhaust opening being advanced so that sufficient time is given for cylinder pressure to fall to or below charge air pressure when the scavenge ports uncover. A complementary aspect of earlier opening to exhaust is the increased pulse energy obtainable from the exhaust gas which can be utilised to improve turbocharger performance. In many cases this is the main criterion which influences exhaust opening, since the loss of expansive working is more than offset by the gain in turbocharger output.

Obviously in the case of reversing engines there may be some slight penalty incurred if prolonged operation in the astern direction is considered. Fig. 79. shows the timing for some of the present generation of direct drive slow speed diesels.

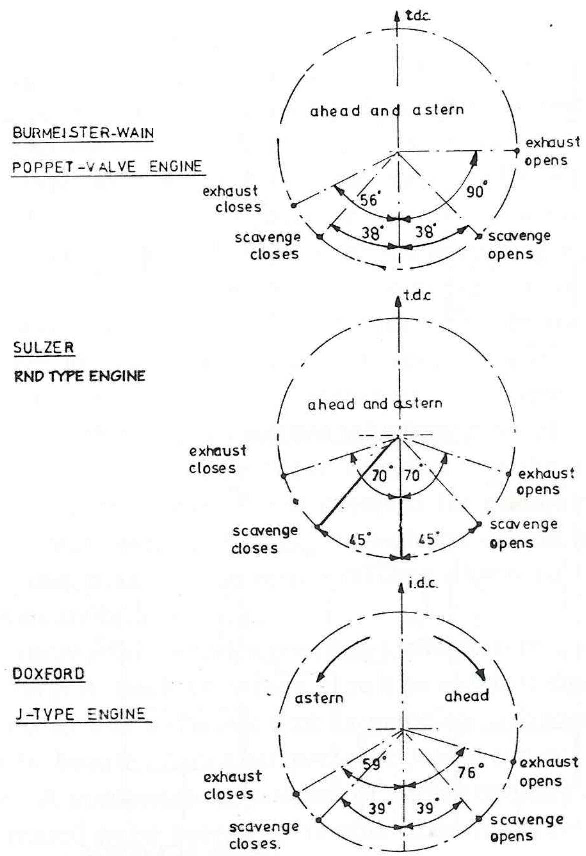
FIG 78
SCAVENGING OF 2-STROKE ENGINES



PRESSURE CHARGING

By increasing the density of the air charge in the cylinder at the beginning of compression a corresponding greater mass of fuel can be burned giving a substantial increase in power developed. The degree of pressure charging required, which determines the increase in air density, is achieved by the use of free running turbochargers which are driven by the exhaust gases expelled from

FIG 79
TIMING FOR SOME DIRECT DRIVE SLOW SPEED DIESELS



the main engine. About 20% of the energy available in the exhaust gas is utilised in this way. In the past it was usual practice to employ some form of scavenge assistance either in series or parallel with the turbochargers. This was accomplished by engine driven reciprocating scavenge pumps, under piston effect or independently driven auxillary blowers. Only the under-piston effect and auxillary blowers are used to any significant degree in modern practice.

FIG 80
PRESSURE-CHARGING

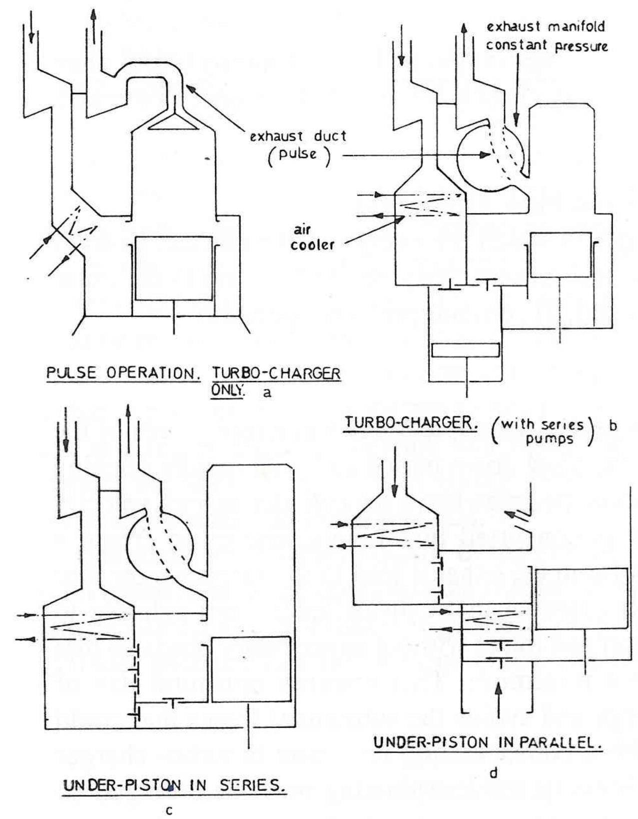


Fig. 80. (b) and (c). turbocharger provides charge air at 70 to 95% of required pressure with under-piston effect or series pump making up the balance. Slight increase in temperature of air delivered to engine since air cooling is carried out after the turbocharger only.

Fig. 80. (d). With parallel operation air supply to engine is increased by air delivery from pumps with proportionate increase in output resulting in greater exhaust gas supply to turbocharger and improved turbocharger performance.

The advantages of pressure-charging may be summed up as: (1) substantial increase in power for a given speed and size; (2) better mass power ratio, *i.e.* reduced engine mass for given output; (3) improved mechanical efficiency with reduction in specific fuel consumption; (4) reduction in cost per unit of power developed; (5) the increase in air supply has a considerable cooling effect leading to less exacting working conditions and improved reliability. Because of increasing power output and fuel economy diesel plant is now almost universally chosen for applications once dominated by steam turbine plant.

Constant Pressure and Pulse Operation

In general the manner in which the energy of the exhaust gases is utilised to drive the turbocharger may be ascribed to (1) the pulse system of operation and (2) constant pressure operation.

Pulse Operation

This makes full use of the higher pressures and temperatures of the exhaust gas during the blow-down period and with rapidly opening exhaust valves or ports the gases leave the cylinder at high velocity as pressure energy is converted into kinetic energy to create a pressure wave or pulse in the exhaust lead to the turbocharger. For pulse operation it is essential that exhaust leads from cylinder to turbine entry are short and direct without unnecessary bends so that volume is kept to a minimum. This ensures optimum use of available pulse energy and avoids the substantial losses that could otherwise occur with a corresponding reduction in turbo- charger performance. Of necessity, exhaust ducting must be arranged so that the gas-exchange processes of cylinders serving the same turbocharger do not interfere with each other to cause pressure disturbances that would affect purging and recharging with an adverse effect upon engine performance. With 2-stroke engines the optimum arrangement is three cylinder grouping with 120° phasing which gives up to 10% better utilisation of available energy than cylinder groupings other than multiples of three. Due to the small volume of the exhaust ducting and direct leading of exhaust to turbine inlet the pulse system is highly responsive to changing engine conditions giving good performance at all speeds. Theoretically turbocharging on the pulse system does not require

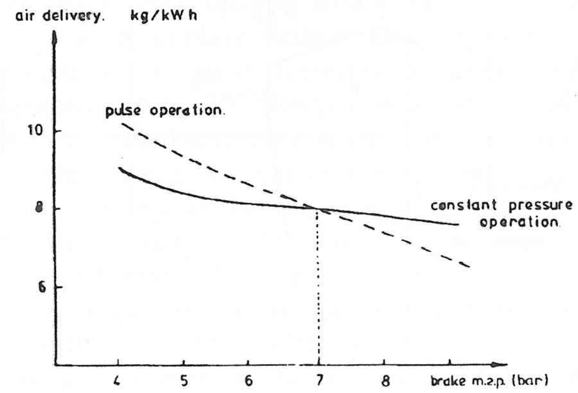
any form of scavenge assistance at low speeds or when starting. In practice however the use of an auxiliary blower or some other means of assistance is employed to ensure optimum conditions and good acceleration from rest.

Constant Pressure Operation

In this system the exhaust gases are discharged from the engine into a common manifold or receiver where the pulse energy is largely dissipated. Although the pulse energy is lost, the gas supply to the turbine is at almost constant pressure so that optimum design conditions prevail since, under normal conditions, gas flow will be steady rather than intermittent. Further, as engine ratings increase, the constant pressure energy contained in the exhaust gas becomes increasingly dominant so that sacrifice of pulse energy in a large volume receiver is of less consequence. Fig. 81. shows the results of tests carried out on a Sulzer type engine which indicates that up to b.m.e.p. of around 7 bar the advantage lies with the pulse system but as b.m.e.p. increases beyond this figure the constant pressure system becomes more efficient giving greater air throughout and some slight reduction in the fuel rate.

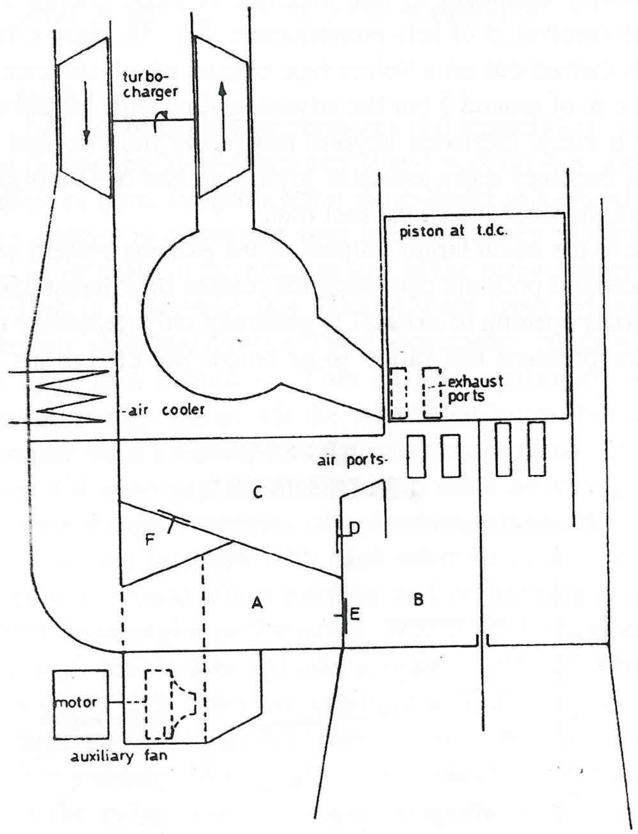
Due to the much larger volume of the exhaust system associated with constant pressure operation the release of exhaust gas is rapid and earlier opening to exhaust is generally only necessary to ensure cylinder pressure has fallen to or below the charge air pressure

FIG 81
AIR DELIVERY



when the scavenge ports begin to uncover. With a possible reduction in exhaust lead expansive working can be increased which is a further contributory factor in reducing the fuel rate. A major drawback to constant pressure operation is that the large capacity of the exhaust system gives poor response at the turbocharger to changing engine conditions with the energy supply at slow speeds being insufficient to maintain turbocharger performance at a level consistent with efficient engine operation. Some form of scavenge assistance such as under-piston scavenging is often utilised. To offset this however the number of

FIG 82
RND 90 SUPERCHARGING ARRANGEMENT



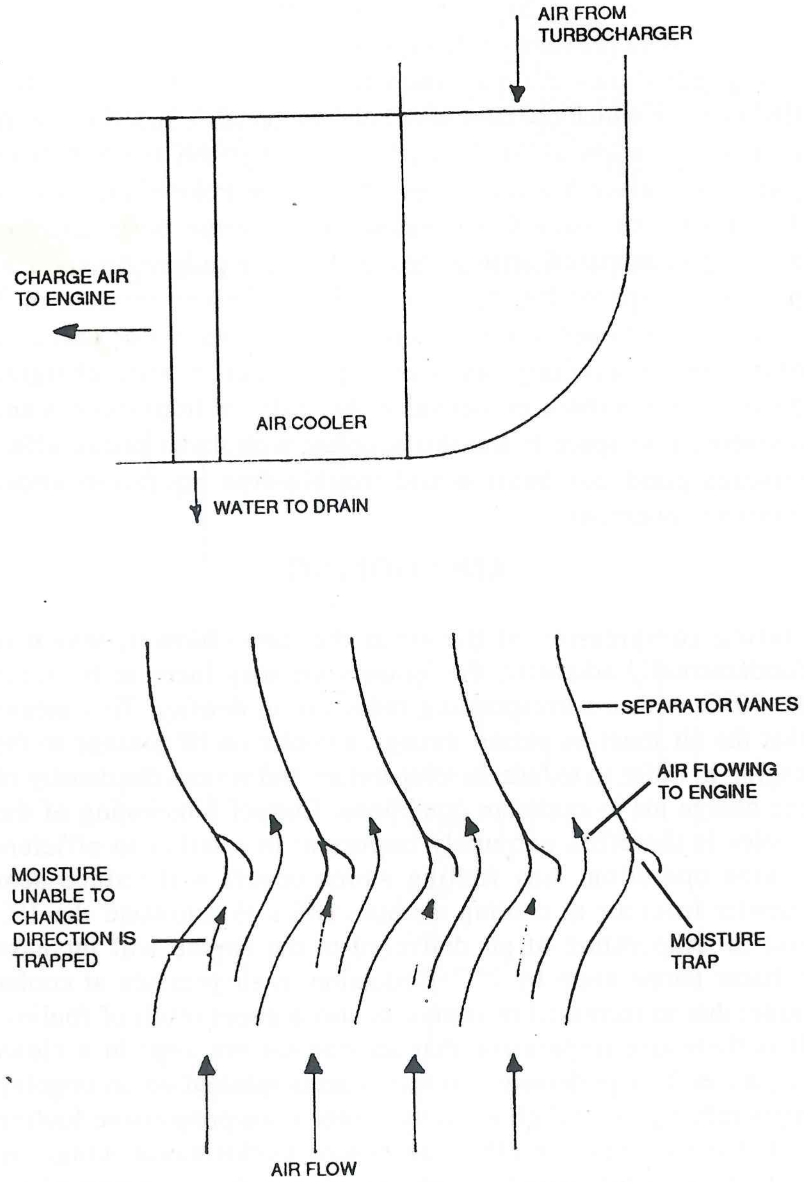
turbochargers required as compared to pulse operation can be reduced, a greater flexibility exists in the case of turbocharger location and exhaust arrangement and no de-rating of engine need be considered for cylinder groupings other than multiples of three. For this reason most large slow speed 2-stroke engines tend to be of the constant pressure configuration.

Fig. 82. shows the diagrammatic arrangement of the Sulzer RND engine which operates with constant pressure supercharge. In normal operation air is drawn into under-piston space B from common receiver A and compressed on downstroke of piston to be delivered into space C so that when scavenge ports uncover purging is initiated with a strong pressure pulse. As soon as pressure in spaces B/C falls to common receiver pressure in A scavenge continues at normal charge air pressure. For part load operation the auxiliary fan is arranged to cut in when charging pressure falls below a pre-set value. Air is drawn from space A and delivered into space F and this together with under piston effect ensures good combustion and trouble-free operation under transient conditions.

AIR COOLING

During compression of the air at the turbo-blower, which is fundamentally adiabatic, the temperature may increase by some 60-70°C with a corresponding reduction in density. This means that the air must be passed through a cooler on its passage to the engine in order to reduce its temperature and restore the density of the charge air to optimum conditions. Correct functioning of the cooler is therefore extremely important in relation to efficient engine operation. Any fouling which occurs will reduce heat transfer from air to cooling medium and it is estimated the 1°C rise in temperature of air delivered to the engine will increase exhaust temperature by 2°C. Reduction in air pressure at cooler outlet due to increased resistance is also a direct result of fouling. It is therefore imperative that air coolers are kept in a clean condition. It is preferable that this is accomplished on an ongoing basis rather than changing a dirty cooler since progressive fouling will have an adverse effect on engine performance. Ongoing cleaning can be carried out by spraying with a commercial air cooler cleaning solvent. Under conditions of high humidity precipitation at the cooler may be copious. Carry over of this water

FIG 83
WATER SEPARATOR



to the engine can have a number of detrimental effects. Water contamination of cylinder lubricating oil may reduce its viscosity and hence its ability to withstand the imposed loads leading to increased cylinder and piston ring wear. Water contamination may also lead to corrosion of engine components. To prevent the carry-over of water a water separator is fitted. Fig. 83. shows a water separator fitted on the outlet side of an air cooler. This separator utilises the difference in the mass of water and air. As the moist air flows into the vanes its direction is changed. Because of its lower mass the air is able to change direction easily to flow around the vanes. The water, however, because of its greater mass and, therefore momentum, is not able to change direction so easily and flows into the water trap to be removed at the drain. The water separator should also be sprayed with cleaning solvent when cleaning the air cooler. It must be noted that the vapour given off by cleaning solvents is harmful and by spraying into air coolers may contaminate the atmosphere throughout the engine. The air coolers should not be cleaned when personnel are working within the engine.

TURBOCHARGERS

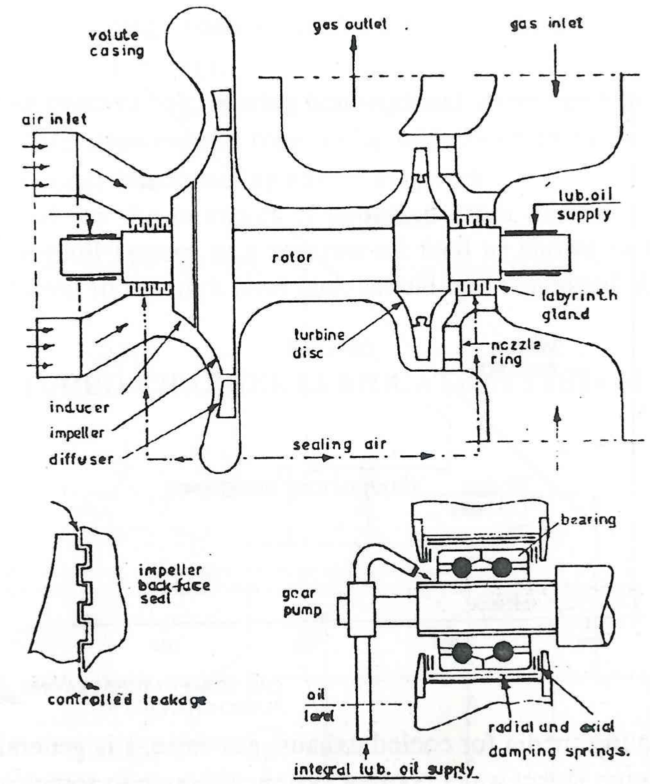
These are essentially a single stage axial flow turbine driving a single stage centrifugal air compressor via a common rotor shaft to form a self-contained free running unit. Expansion of the exhaust gas through the nozzles results in a high velocity gas stream entering the moving blade assembly. Because of the high rotational speeds perfect dynamic balance is essential if troublesome vibrations are to be avoided. Even with this, the effect of external vibrations being transmitted via the ship's structure to the turbocharger is a further problem to be resolved. This is done by mounting the bearings in resilient housings incorporating laminar spring assemblies to give both axial and radial damping effect. Another aspect of this arrangement is to prevent flutter or chatter at bearing surfaces when stopped so that incidental bearing damage is prevented. Lubrication of the bearings may be by separate or integral oil feed, but whatever arrangement is adopted it must be fully effective at a steady axial tilt of 15° and support a temporary tilt of $22\frac{1}{2}°$ as may occur in a heavy seaway. The bearings themselves may be a combination of ball and roller bearings or separate sleeve (journal) type bearings.

The various claims of superiority as to the effectiveness of the different types of bearings centre around the mechanical efficiency of the bearing configuration. The manufacturers of turbochargers equipped with rolling element bearing claim a distinct mechanical efficiency advantage across the whole operating range. On the other hand manufacturers of turbochargers equipped with sleeve type bearings claim comparable efficiency under full-load conditions but accede to lower efficiency at lower engine loads. With high speeds of operation the mechanical efficiency factor does seem to favour rolling element bearings. Against this however is the fact that periodic replacement of ball and roller assemblies is essential if trouble-free service is to maintained – this is due to the fact that rapid and repeated deformation with resultant stressing causes surface metal fatigue of contact surfaces with the result that failure will occur. The effects of vibration, overloading, corrosion or possible abrasive wear, etc, lead to premature failure which emphasises the need for isolation of bearings from external vibrations together with use of correct grade of lubricant and effective filtration, etc. Plain bearings should however have a life equal to that of the blower provided that normal operating conditions are not exceeded.

Referring to Fig. 84. it can be seen that the blower end of the turbocharger consists of a volute casing of light aluminium alloy construction which houses the inducer, impeller and diffuser which are also of light alloy construction. The function of the inducer is to guide the air smoothly into the eye of the impeller where it is collected and flung radially outward at ever increasing velocity due to the centrifugal effect at high rotational speed. At discharge from the impeller it passes to the diffuser where its velocity is reduced in the divergent passages thus converting its kinetic energy into pressure energy. The diffuser also functions to direct air smoothly into the volute casing which continues the deceleration process with further increase in air pressure. From here the air passes to the charge air receiver via the air cooler.

The turbine end of the turbocharger consist of casings which house the nozzle-ring turbine wheel and blading, etc. In older designs casings were water cooled but in turbochargers for modern large slow-speed 2-stroke engines, with relatively low exhaust gas

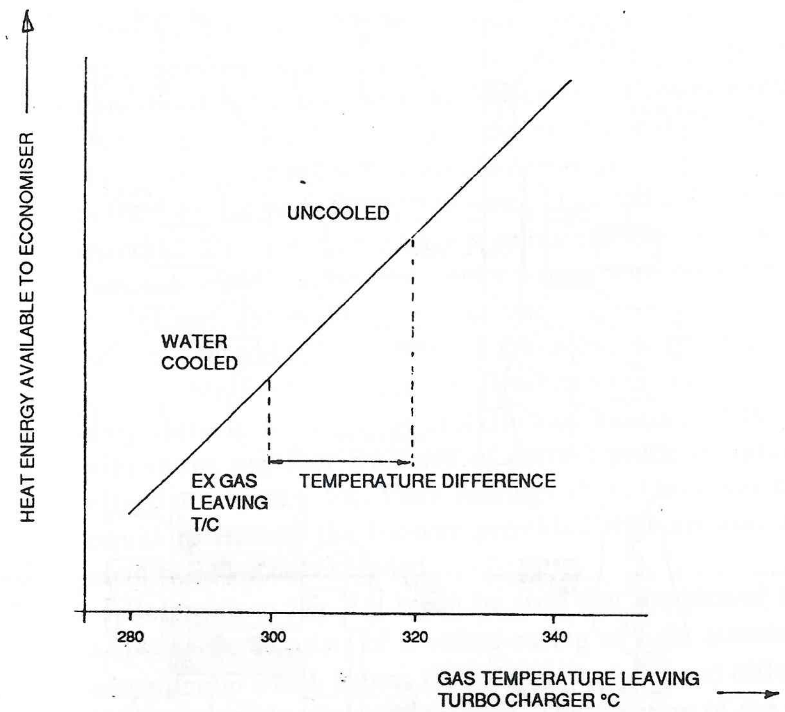
FIG 84
TURBO-CHARGER



temperatures the casings are uncooled. Uncooled designs retain more heat energy in the exhaust gas in the waste heat boiler so improving the overall plant efficiency. Fig. 85. shows the temperature advantage of uncooled designs.

The components in the high temperature gas stream, that is, the nozzle ring, turbine wheel, blades and rotor shaft are manufactured from heat resisting nickel-chrome alloy steel to withstand continuous operation at temperatures in excess of 450°C. Some degree of cooling may be given by controlled air leak-off past the labyrinth seal, between the back of the impeller and volute casing, which flows along the shaft towards the turbine end.

FIG 85
ADVANTAGE OF UNCOOLED TURBOCHARGER



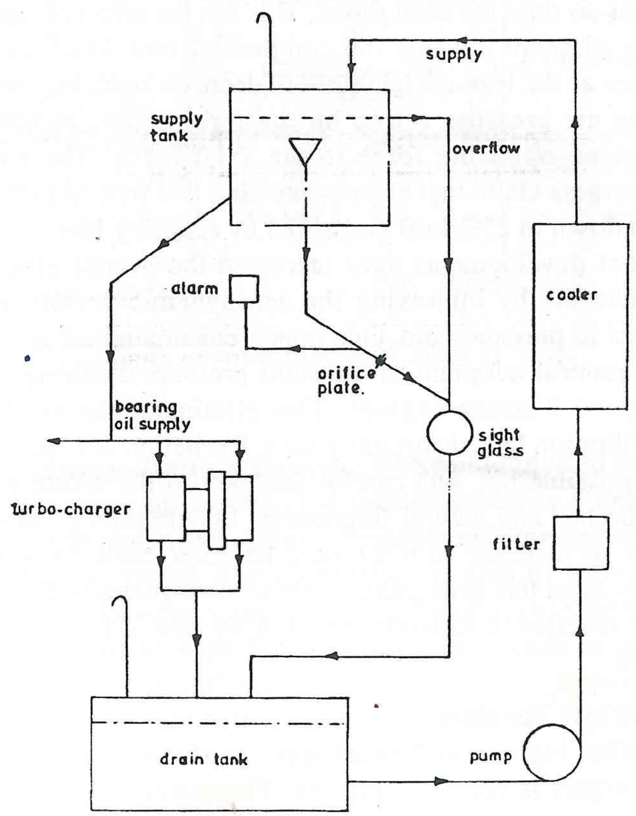
Cooling media for cooled exhaust gas casings is generally from the engine jacket water cooling system although in some cases sea-water has been employed. In both cases anti-corrosion plugs are fitted to prevent or inhibit corrosion on the water-side. With water cooled casings experience has shown that under light load conditions when low exhaust temperatures are encountered it is possible that precipitation of corrosive forming products – mainly sulphuric – will occur on the gas side of the casing. This results in serious corrosive attack which is more marked at the outlet casing because of lower temperatures. Methods of prevention such as enamelling and plastic coatings, etc. have been tried to alleviate this problem with varying degrees of success. A particularly effective approach to the problem is the use of air as the cooling media with the result that this particular instance of corrosive attack is virtually eliminated.

Some manufactures utilising sleeve type bearings mount them inboard of the compressor and turbine. This has several advantages:

1. A short, rigid shaft is possible.
2. It allows large volume turbine and compressor inlet casings, free of bearing housings.
3. The main casing, bearing housings and turbo-machinery form one module allowing the rotor to be withdrawn from the turbine casing without disconnecting engine ductwork.

The oil for the bearings is supplied from the main engine lubricating oil system or a separate oil feed as shown in Fig. 86. The oil level in the high level tank should be maintained about 6m

FIG 86
TURBO-CHARGER LUBRICATION SYSTEM



above the turbochargers. This will ensure that the oil pressure reaching the bearings should never fall below a pressure of around 1·6 bar. If level of oil falls below the mouth of the inner drain pipe it is quickly emptied and an alarm condition initiated. After an alarm it takes about ten minutes to empty the high level tank which is sufficient to ensure adequate lubrication of the turbochargers as they run down after the engine is stopped.

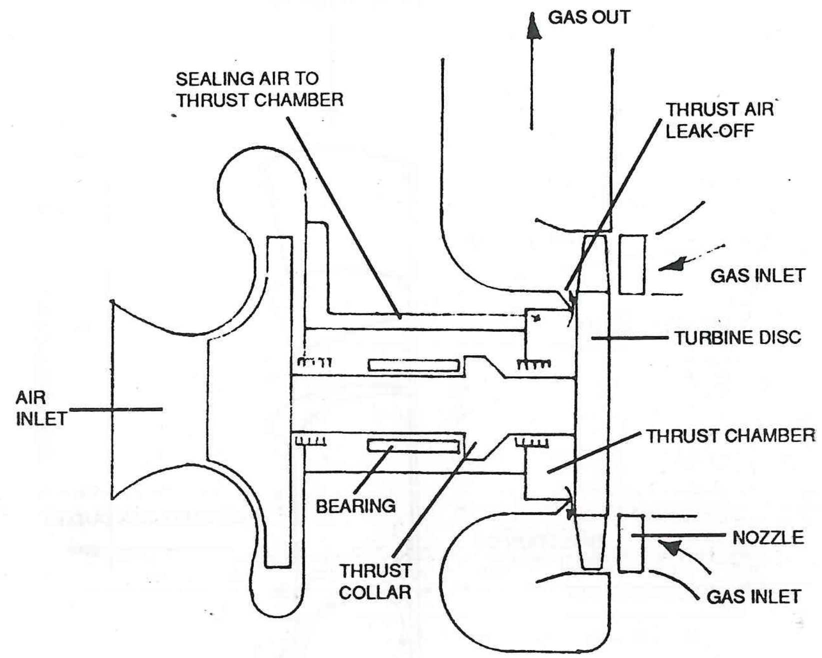
As discussed earlier sleeve type bearings suffer the disadvantage of having a lower mechanical efficiency at part load conditions. The effects of this can be minimised by careful design. To reduce friction the bearing length is reduced. A thrust bearing is incorporated into the main bearing but axial thrust is taken by this only at start-up, shut-down and very low loads. The main thrust being taken by sealing air acting on the turbine disc. Fig. 87. shows sealing air from the compressor outlet being fed to the chamber behind the turbine disc. This air flows past the leak-off labyrinth at a rate dependant upon the clearance. As the turbocharger load increases so does the axial thrust. This has the effect of moving the rotating element towards the compressor end which causes the clearance at the leak-off labyrinth to decrease reducing the flow of air. The air pressure acting on the turbine disc increases and imposes an opposing force to the axial force. The makers of turbochargers claim that engines utilising this type of turbocharger can run down to 25% load unassisted by auxiliary fans.

Recent developments have increased the overall efficiency of turbochargers by improving the aerodynamic performance and increases in pressure ratio. One improvement attained is as a result of the general adoption of constant pressure charging for large slow-speed 2-stroke engines. This eliminates the excitation of blade vibration by exhaust gas pulses. Excitation of blade vibration is still possible but with careful attention to the choice of nozzle vane number and natural frequencies of vibration of blades it is possible to dispense with the need for rotor blade damping wire. Not only does this give greater turbine aerodynamic efficiency, but greater resistance to contamination by heavy fuel combustion products.

Radial Flow Turbines

For smaller higher speed diesel applications the use of radial flow turbochargers is common. Fig. 88. The casings are uncooled but

FIG 87
TURBO-CHARGER WITH PLAIN BEARINGS



require insulation. Bearings are sleeve type and lubricated form the engine lubricating oil system. The turbine wheel is a one piece casting of a design which gives acceptable efficiencies over the entire operation range. The compressor is also of a one piece design of backswept vane design giving stable operating characteristics. At high air flows the efficiency tends to decrease due to losses at the turbine exit. A comparison between the efficiencies of axial and radial turbines can be seen in Fig. 89.

FIG 88
RADIAL FLOW TURBINE

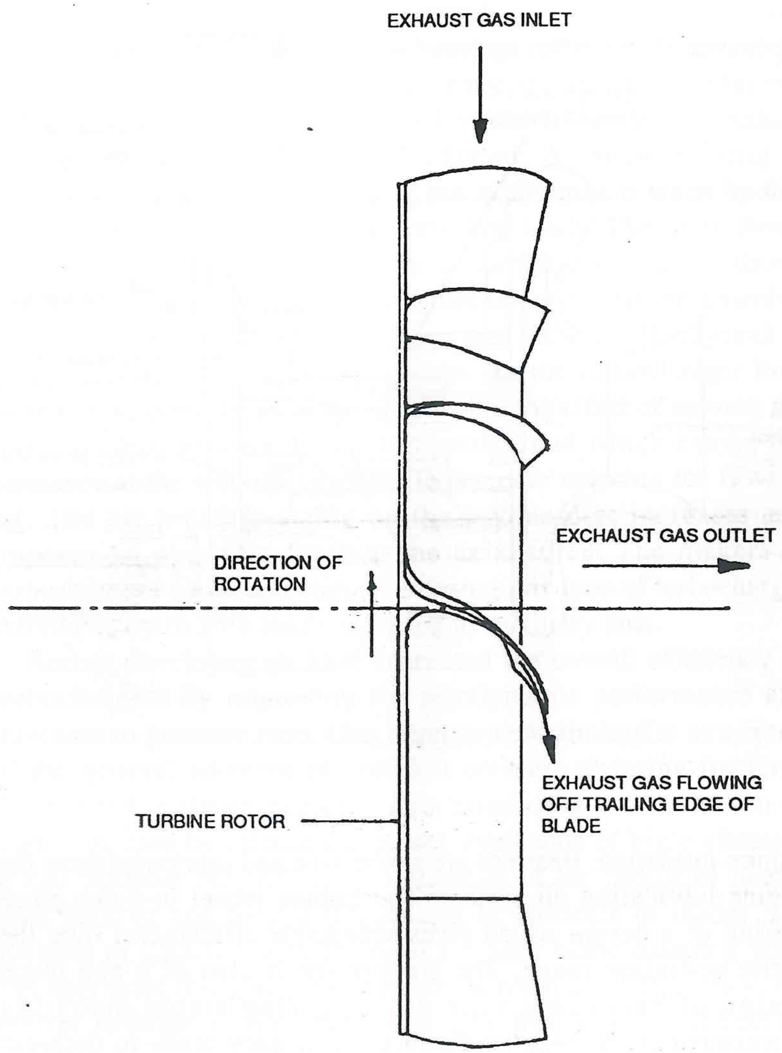
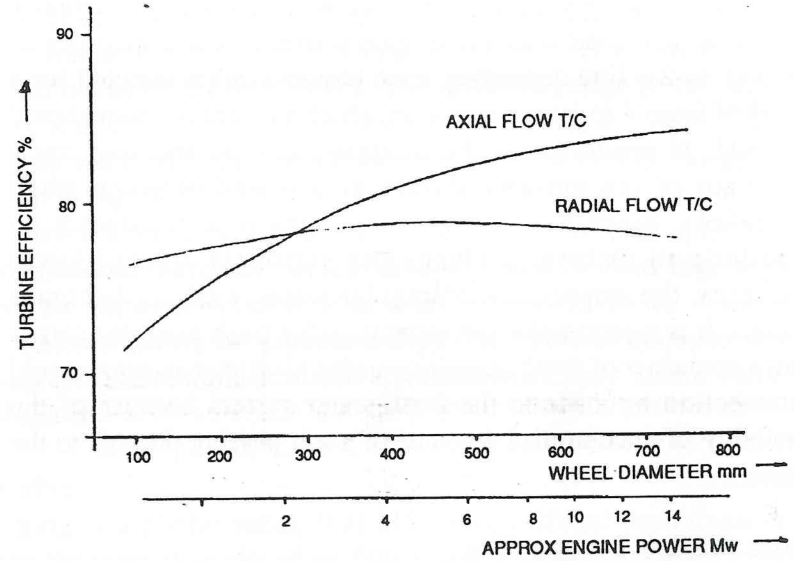


FIG 89
COMPARISON OF RADIAL & AXIAL FLOW T/C EFFICIENCY



Turbocharger Fouling

Contaminated turbines and compressors have poorer efficiency and lower performance which results in higher exhaust temperatures. In 4-stroke applications the charging pressure can increase due to the constriction of the flow area through the turbine resulting in unacceptable high ignition pressures. To maintain turbocharger efficiency it is important to ensure that all operating parameters are maintained to manufacturers recommiendations. If the compressor draws its air from the machinery spaces then steps must be taken to maintain as clean an atmosphere as possible since leaking exhaust gas and/or oil vapour will accelerate the deterioration of efficiency. In some installations the turbo-chargers draw air through ducts from outside the engine room.

TURBOCHARGER CLEANING

Water Washing – Blower Side

On the air side, dry or oily dust mixed with soot and a possibility of salt ingestion from salt laden atmosphere can lead to deposits which are relatively easy to remove with a water jet, usually injected at full load with the engine warm. A fixed quantity of liquid (1 to 2 1/2 litre depending upon blower size) is injected for a period of from 4 to 10 seconds after which an improvement should be noted. If unsuccessful the treatment can be repeated but a minimum of ten minutes should be allowed between wash procedures. Since a layer of a few tenths of a millimetre on impeller and diffuser surfaces can seriously affect blower efficiency the importance of regular water washing becomes obvious. It is essential that the water used for wash purposes comes from a container of fixed capacity – under no circumstances should a connection be made to the fresh water system because of the possibility of uncontrolled amounts of water passing through to the engine.

Water Wash – Turbine Side

This is generally carried out at reduced speed by rigging a portable connection to the domestic fresh water system and injecting water, via a spray orifice before the protective grating at turbine inlet, for a period of 15 to 20 minutes with drains open to discharge excessive moisture which does not evaporate off. Since water washing may not completely remove deposits, and can interact with sulphurous deposits with resultant corrosive attack, chemical cleaning may be used in preference. This effectively removes deposits at the turbine and moreover is still active within the exhaust gases passing to the waste heat system, so that further removal of deposits occurs which maintains heat transfer at optimum condition and keeps back-pressure of exhaust system to well within the limits required for efficient engine operation.

Dry Cleaning – Turbine Side

Instead of water, dry solid bodies in the form of granules are used for cleaning. About 1·5-2 kg of granules is blown by compressed air into the exhaust gas lines before the gas inlet casing or protection grid.

Agents particularly suited to blasting are natural kernel granules, or broken or artificially shaped activated carbon particles with a grain size 1·2 to 2·0 mm.

The blasting agents have a mechanical cleaning effect, but it is not possible to remove fairly thick deposits with the comparatively small quantity used. For this reason this method must be adopted more frequently than for cleaning with water. Dry cleaning is carried out every 24 to 50 hours. The main advantage of this type of turbine cleaning is that it can be carried out at full or only slightly reduced load. The cleaning equipment configuration is shown in Fig. 90.

Turbocharger manufacturers recommend that heavily contaminated machines, which have not been cleaned regularly from the very beginning or after overhaul, should not be cleaned by water-washing or granulate injection. This is because the dangers of incomplete removal of deposits may cause rotor imbalance.

Surging

Surging is a phenomenon that affects centrifugal compressors when the mass flow rate of air falls below a sustainable level for a given pressure ratio. Consider the system in Fig. 91. where a constant speed compressor supplies air though a duct. The outlet of the duct is regulated by a damper. With the damper fully open the pressure ratio across the compressor will be at its lowest value with the largest mass flow rate of air. As the damper is closed the resistance increases as does the pressure ratio but the mass flow of air decreases. If the damper is closed further a point will be reached where, because of the resistance, there will be such a low mass flow rate and high pressure ratio across the compressor that flow breaks down altogether. When this occurs the pressure downstream of the compressor is simply relieved to atmosphere, backwards, through the compressor. This is known as surging and is accompanied by loud sounds of "howling and banging". The

FIG 90
DRY TURBO-CHARGER CLEANING EQUIPMENT

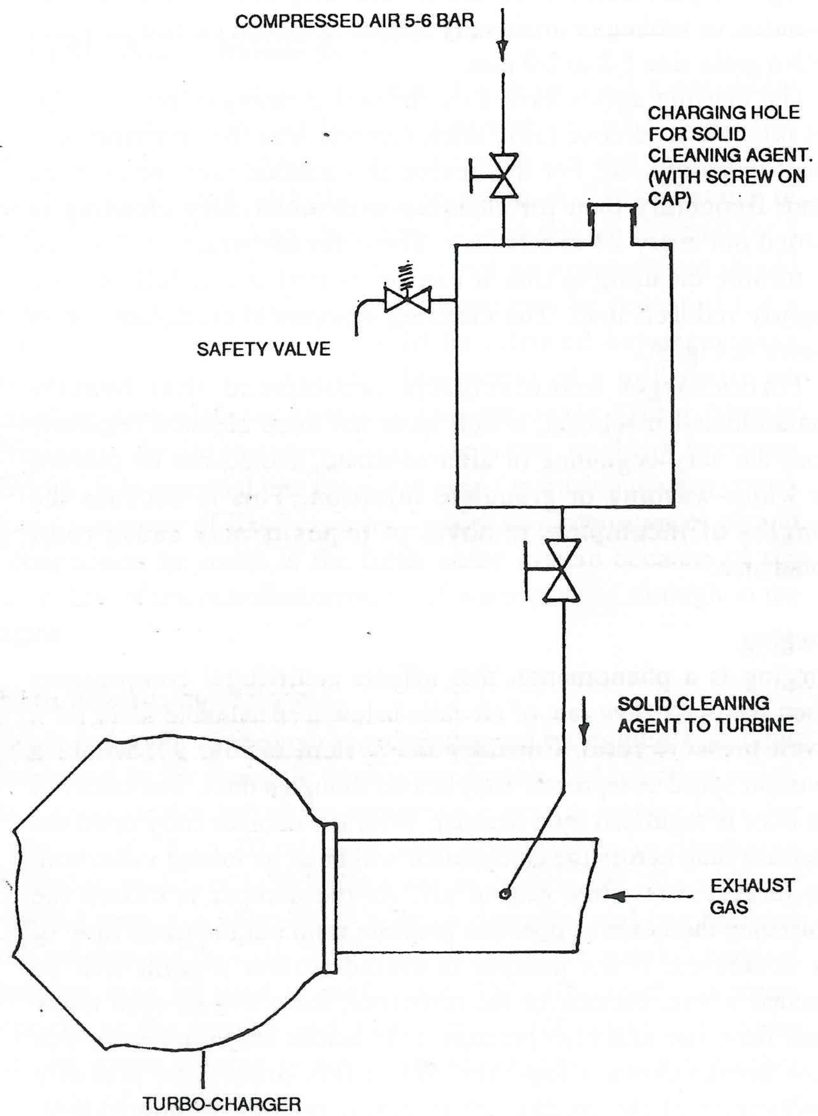
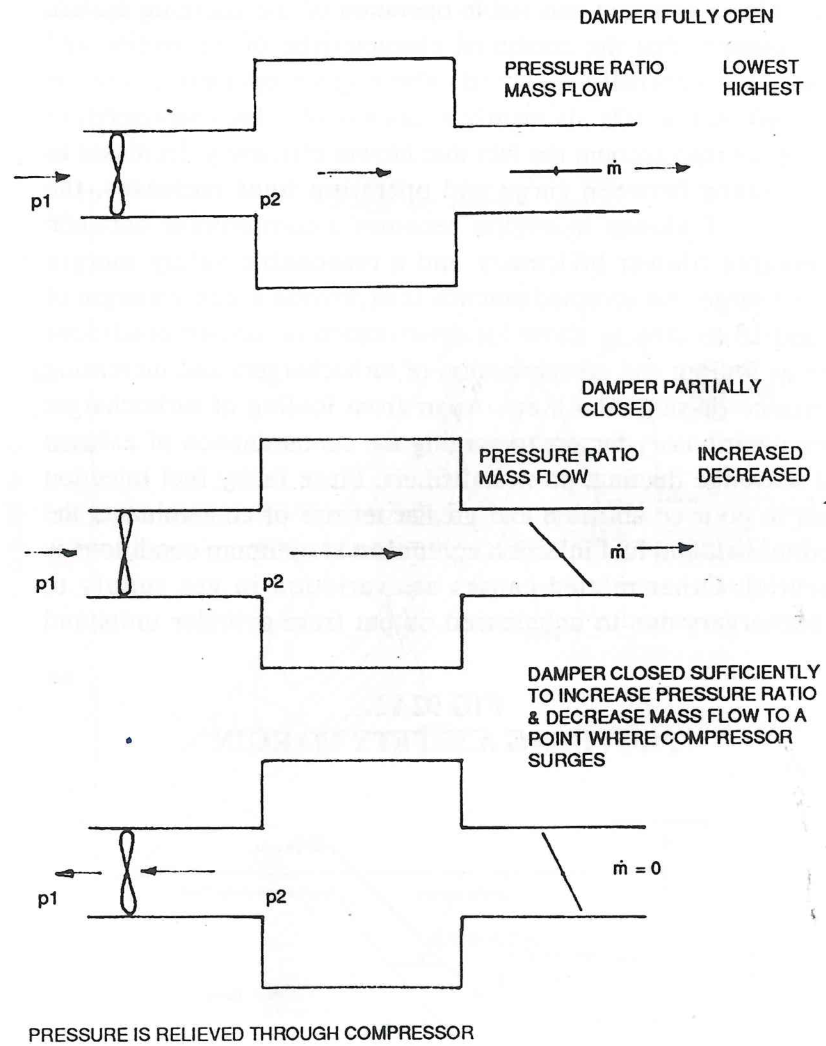


FIG 91
SURGING OF TURBO-CHARGER COMPRESSOR



events leading to the surging can be followed on a graph of pressure ratio against mass flow. This graph is known as a compressor map. Fig. 92b.

Surging may occur in heavy weather when the propeller comes out of the water and the governor shuts the fuel off almost instantaneously.

To obtain efficient and stable operation of the charging system it is essential that the combined characteristic of the engine and blower are carefully matched. The engine operating line, as indicated on Fig. 92a., is mainly a function of these characteristics and taking into account the fact that blower efficiency decreases as the distance between surge and operating lines increases, the matching of blower to engine becomes a compromise between acceptable blower efficiency and a reasonable safety margin against surge. An accepted practice is to provide a safety margin of around 15 to 20% to allow for deterioration of service conditions such as fouling and contamination of turbochargers and increasing resistance of ship's hull, etc. Apart from fouling of turbocharger other contributory factors to surging are contamination of exhaust and scavenge ducting, ports and filters. Since faulty fuel injection leads to poor combustion and greater release of contaminants the need to maintain fuel injection equipment at optimum conditions is essential. Other related causes are variation in gas supply to turbochargers due to unbalanced output from cylinder units and

FIG 92A
PROVIDING A SAFETY MARGIN

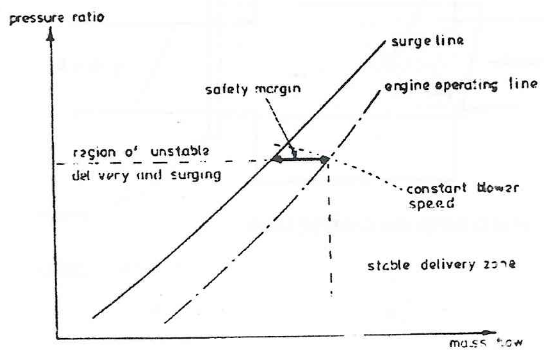
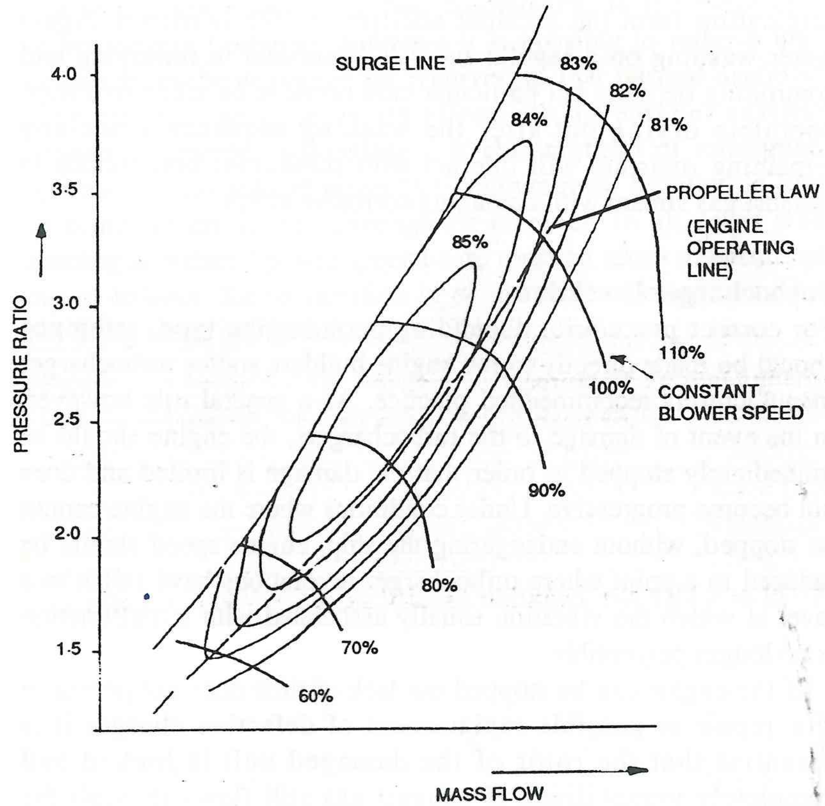


FIG 92B
COMPRESSOR MAP



mechanical damage to turbine blading, nozzles or bearings, etc.

During normal service the build-up of contaminants at the turbocharger can be attributed to deposition of air-borne contaminants at the compressor which in general are easily removed by waterwashing on a regular basis. At the turbine however, more active contaminants resulting from vanadium and sodium in the fuel together with the products of incomplete combustion deposit at a higher rate which increases with rising temperature. A further problem arises with the use of alkaline cylinder lubricants with the formation of calcium sulphate deposits originating form the alkaline additives in the lubricant. Again water washing on a regular basis is beneficial in removing and controlling deposits but particular care needs to be taken to ensure complete drying out after the washing sequence since any remaining moisture will interact with sulphurous compounds in exhaust gas stream with damaging corrosive effect.

Turbocharger Breakdown

For correct procedure, depending upon engine type, reference should be made directly to the engine builders and/or turbocharger manufacturers recommended practice. As a general rule however, in the event of damage to the turbochargers, the engine should be immediately stopped in order that the damage is limited and does not become progressive. Under conditions where the engine cannot be stopped, without endangering the ship, engine speed should be reduced to a point where turbocharger revolutions have fallen to a level at which the vibration usually associated with a malfunction is no longer perceptible.

If the engine can be stopped but lack of time does not permit *in situ* repair or possible replacement of defective charger it is essential that the rotor of the damaged unit is locked and completely immobilised. If exhaust gas still flows through the affected unit once the engine is restarted, the coolant flow through the turbine casing needs to be maintained but due to the lack of sealing air at shaft labyrinth glands the lubricating oil supply to the bearings will need to be cut off – with integral pumps mounted on the rotor shaft, the act of locking the shaft ensures this – otherwise contamination of lubricant together with increase in fouling will occur. For rotor and blade cooling a restricted air supply is

required and can be achieved by closing a damper or flap valve in the air delivery line from the charger, to a position which gives limited flow from scavenge receiver back to the damaged blower. Alternatively a bank flange incorporating an orifice of fixed diameter can be fitted at the outlet flange of the blower.

With only a single blower out of a number inoperative the power developed by the engine will obviously depend upon charge air pressure attainable. At the same time a careful watch must be kept upon exhaust condition and temperature to ensure efficient engine operation with good fuel combustion. In the event of all turbochargers becoming defective it is possible to remove blank covers from the scavenge air receiver so that natural aspiration supplemented by underpiston effect, etc. or parallel auxiliary blower operation is possible – if this method of emergency operation is carried out protective gratings must be fitted in place of blind covers at the scavenge air receiver. In all cases when running at reduced power special care must be taken to ensure any out of balance due to variation in output from affected units does not bring about any undue engine vibration.

CHAPTER 5

STARTING AND REVERSING

Starting Air Overlap

Some overlap of the timing of starting air valves must be provided so that as one cylinder valve is closing another one is opening. This is essential so as to ensure that there is no angular position of the engine crankshaft with insufficient air turning moment to give a positive start. The usual minimum amount of overlap provided in practice is 15°. Starting air is admitted on the working stroke and the period of opening is governed by practical considerations with three main factors to consider:

1. The Firing Interval of the engine.

$$\text{Firing Interval} = \frac{\text{Number of degrees in engine cycle}}{\text{Numbers of cylinders}}$$

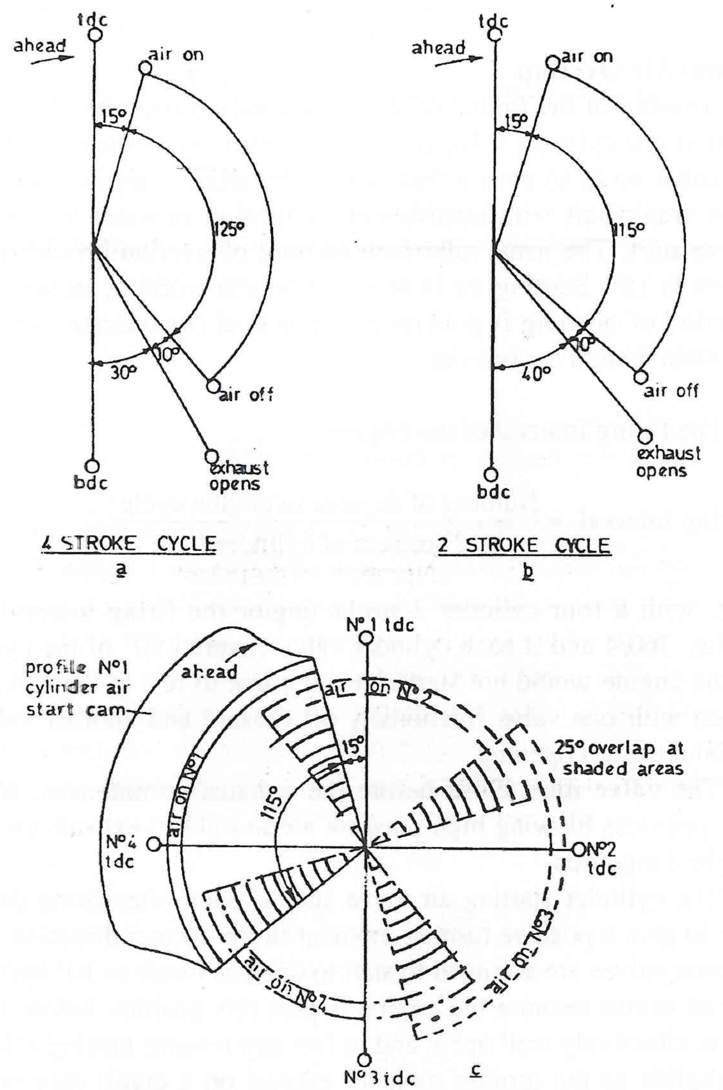
e.g. with a four cylinder 2-stroke engine the firing interval is 90°, *i.e.* 360/4 and if each cylinder valve covered 90° of the cycle then the engine would not start if it had come to rest in the critical position with one valve fractionally off closure and another valve just about to start opening.

2. The valve must close before the exhaust commences. It is rather pointless blowing high pressure air straight to exhaust and it could be dangerous.

3. The cylinder starting air valve should open after firing dead centre to give a positive turning moment in the correct direction. In fact some valves are arranged to start to open as much as 10° before the dead centre because the engine is past this position before the valve is effectively well open, and in fact any reverse turning effect is negligible as the turning moment exerted on a crank very near dead centre is small indeed.

Consider Fig. 93(a). for a 4-stroke engine. With the timings as shown the air starting valve opens 15° after dead centre and closes 10° before exhaust begins. The air start period is then 125°. The

FIG 93
AIR START CAM AND CRANK TIMING DIAGRAMS



firing interval for a 6-cylinder 4-stroke engine is 720/6=120°. The period of overlap is 5° which is insufficient. Although this example could easily be modified so as to give sufficient (say 15°) overlap by reducing the 15° after dead centre and the 10° before exhaust opening, it can become very difficult to arrange with very early

exhaust opening on turbocharged engines. A 7-cylinder 4-stroke engine is much easier to arrange.

Consider Fig. 93(b). for a 2-stroke engine:

This has an air start period of 115°. Firing interval for a 3-cylinder 2-stroke engine = 360/3 = 120°. This means no overlap. Modification can arrange to give satisfactory starting with this example but for modern turbocharged 2-stroke engines having exhaust opening as early as 75° before bottom (outer) dead centre it becomes virtually impossible. A 4-cylinder 2-stroke engine is much easier to arrange and would be adopted. Consider Fig. 93(c). which is a cam diagram for a 2-stroke engine with 4 cylinders. The air open period is 15° after dead centre to 130° after dead centre, *i.e.* a period of 115°. This gives 25° of overlap (115 - 360/4) which is most satisfactory. Take care to note the direction of rotation and this is a cam diagram so that for example, No. 1 crank is 15° after dead centre when the cam would arrange to directly or indirectly open the air start valve. The firing sequence for this engine is 1 4 3 2. This is very much related to engine balancing and no hard and fast rules can be laid down about crank firing sequences as each case must be treated on its merits.

It may be useful to note that for 6-cylinder, 2-stroke engines a very common firing sequence is 1 5 3 6 2 4 and similarly for 7- and 8-cylinders 1 7 2 5 4 3 6 and 1 6 4 2 8 3 5 7 respectively are often used.

The cam on No. 1 cylinder is shown for illustration as it would probably be for operating say cam operated valves, obviously the other profiles could be shown for the remaining three cylinders in a similar way. The air period for Nos. 1, 4, 3 and 2 cylinders are, shown respectively in full, chain dotted, short dotted and long dotted lines and the overlap is shown shaded.

Starting Air Valves

Each cylinder is fitted with a starting air valve which is operated

FIG 94
STARTING AIR VALVE

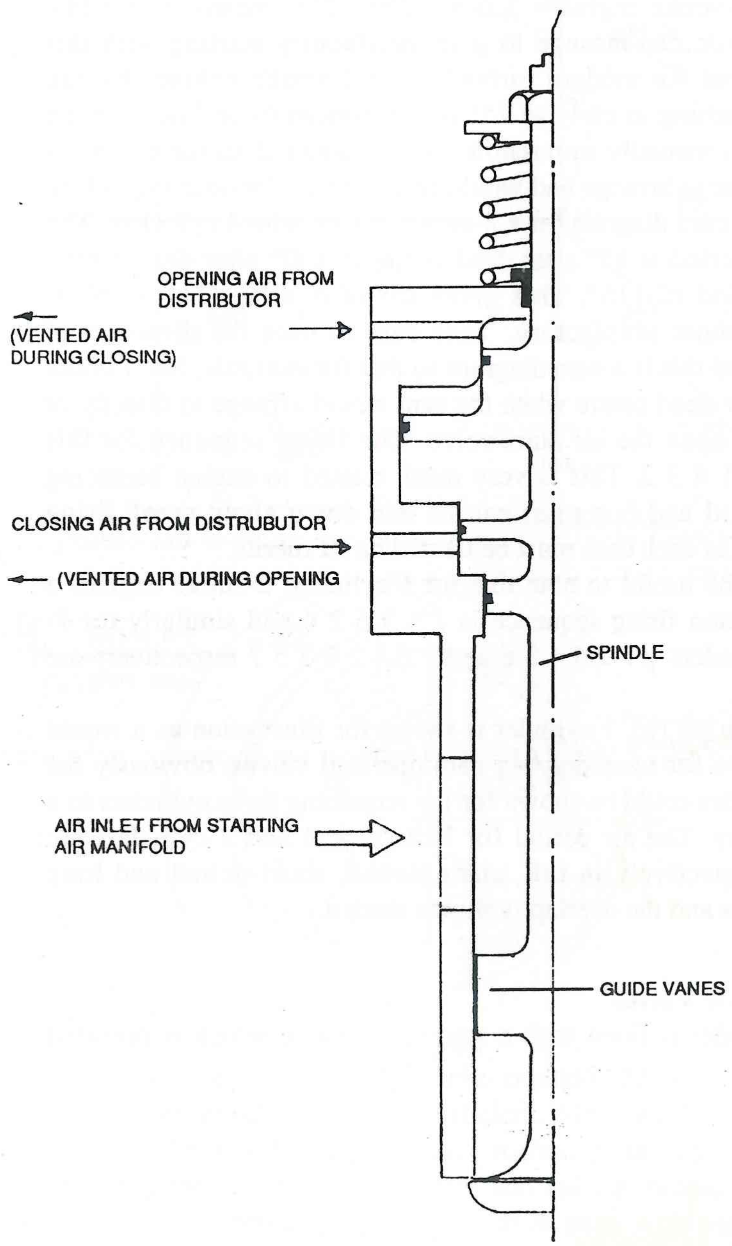
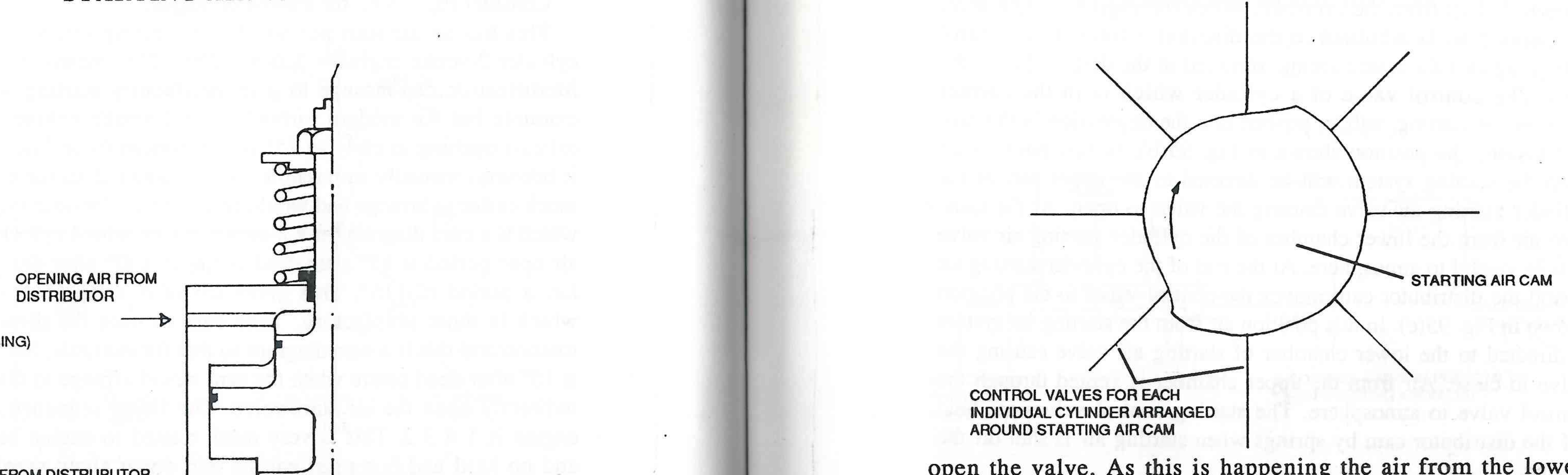


FIG 95A
STARTING CONTROL VALVE ARRANGEMENT
FOR EIGHT CYLINDER ENGINE (SULZER TYPE)



open the valve. As this is happening the air from the lower chamber is vented to atmosphere through the control valve. At the end of the starting air admission period control air is redirected to the lower chamber to close the valve while the upper chamber is vented to atmosphere through the control valve. The valve opens and closes quickly with air cushioning at the end of the closing motion to reduce shock on the valve seat. If the pressure in the cylinder is substantially higher than the starting air pressure, the valve will not open. This prevents hot gases entering the starting air manifold.

During engine operation the air inlet to the starting valve should be regularly checked. A hot inlet would indicate a leaking starting air valve allowing hot combustion gases to enter the air manifold which may lead to an explosion if starting air is admitted.

Starting Air Distributor

There are many designs of air distributor all with the same basic principles, *i.e.* to admit air to the pistons of cylinder relay valves in the correct sequence for engine starting. Valves not being supplied with air would be vented to the atmosphere via the distributor. Some overlap of timing would obviously be required.

One type of starting air distributor is shown in Fig. 95. This is based on the Sulzer design in which each cylinder has its own starting control valve. The starting control valves are arranged radially around the starting air distributor cam, which is driven via a vertical shaft from the camshaft. When the engine starting lever is operated air is admitted to the distributor forcing all control valves, against the return spring, [omitted in the diagram] onto the valves, against the return spring, [omitted in the diagram] onto the cam. The control valve of a cylinder which is in the correct position for starting, will be pushed into the depression in the cam and assume the position shown in Fig. 95(b). In this position air from the starting system will be directed to the upper part of the cylinder starting air valve causing the valve to open. At the same time air from the lower chamber of the cylinder starting air valve will be vented to atmosphere. At the end of the cylinder starting air period the distributor cam moves the control valve to the position shown in Fig. 95(c). In this position air from the starting air system is directed to the lower chamber of starting air valve causing the valve to close. Air from the upper chamber is vented through the control valve to atmosphere. The starting control valves are held off the distributor cam by springs when starting air is shut off the engine.

General Reversing Details

Most reversible engines are of the direct coupled 2-stroke type. The general trend in 4-stroke practice is to utilise an unidirectional engine coupled, via a reduction gearbox, to a controllable pitch propeller. The need for reversing mechanisms for 4-stroke engines is, therefore, reducing. For this reason the 2-stroke reversing mechanism principle will be considered in greatest detail.

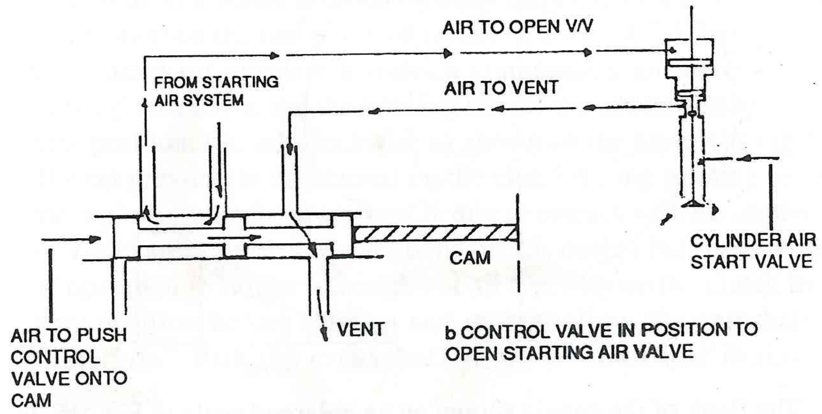
2-Stroke Reversing Gear

It is usually necessary to reposition the fuel cams on the camshaft, with jerk pumps, so that reversing can utilise one cam. This avoids the complication of moving the camshaft axially. This means that it is necessary to provide a lost motion clutch on the camshaft and the need for such a clutch will first be described.

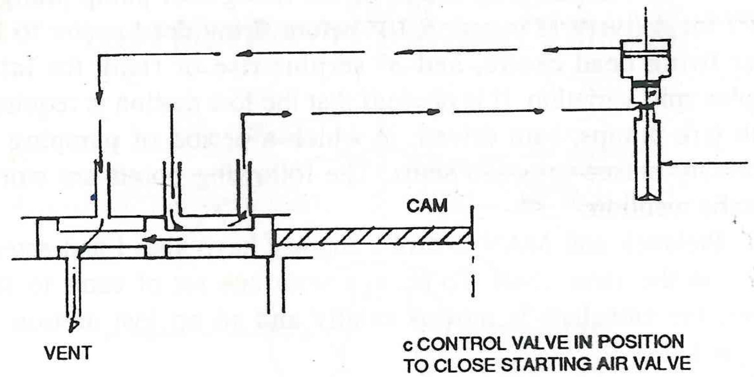
Referring to Fig. 96. the lost motion cam diagram:

Consider the engine position to be dead centre ahead with the cam peak centre line to be 55° after this position, anti-clockwise ahead rotation, for correct injection timing ahead. If now the

FIGS 95B, C
STARTING AIR DISTRIBUTOR



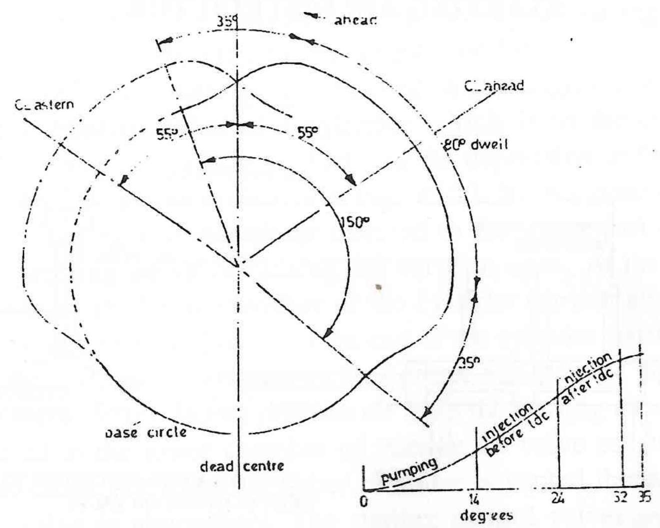
b CONTROL VALVE IN POSITION TO OPEN STARTING AIR VALVE



c CONTROL VALVE IN POSITION TO CLOSE STARTING AIR VALVE

engine is to run astern (clockwise) the cam is 55 + 55 = 110° out of phase. Either the cam itself must be moved by 110° or while the engine rotates 360° the cam must only rotate 250° (110° of lost motion). Note the symmetrical cam 75° each side of the cam peak centre line made up of 35° rising flank and 40° of dwell.

FIG 96
LOST MOTION CAM DIAGRAM



The flank of the cam is shown on an enlarged scale in Fig. 96. It will be noted that the 35° of cam flank is utilised for building up pressure by the pumping action of the rising fuel pump plunger (14°) for delivery at injection 10° before firing dead centre to 8° after firing dead centre, and 3° surplus rise of flank for later surplus spill variation. It is obvious that the lost motion is required with jerk pumps, cam driven, in which a period of pumping is necessary before injection starts. The following points are worth specific mention:

1. Pielstick and MAN 2-stroke engines have ahead and astern cams on the same shaft. To change from one set of cams to the other, the camshaft is moved axially and so no lost motion is required.

2. The dwell period is not normally necessary from the fuel injection aspect alone, *i.e.* about 30° lost motion would be adequate and is provided as such on British Polar and older Sulzer engines.

3. Dwell, in which the fuel plunger is held before return is often provided to give a delay interval. For example with older B & W. engines about 80° dwell gives a rotation (total) of the camshaft of

about one third of a revolution which allows an axial travel with a screw nut arrangement of reasonable size and pitch to change over for reverse running.

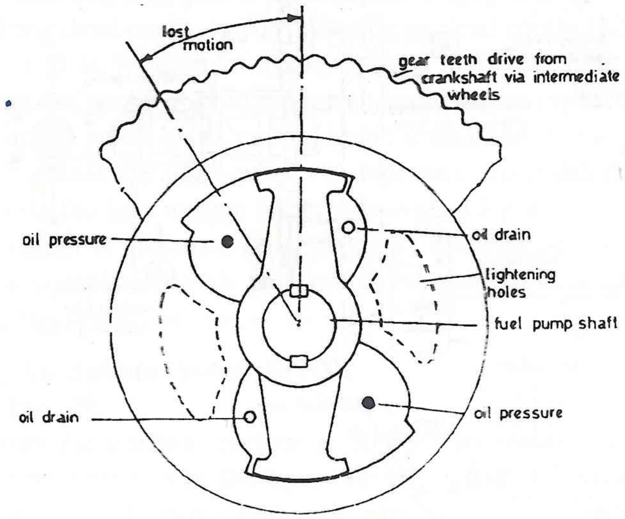
4. Older loop scavenged Sulzer engines have about 98° lost motion as the distributor repositioning for astern is from the same drive shaft as the fuel pumps, but via a vertical direct drive shaft.

Refer now to Fig. 97. the Lost Motion Clutch.

This design which is based on older Sulzer engine practice has a lost motion on the fuel pump camshaft of about 30°. When reversal is required oil pressure and drain connections are reversed. Oil flowing laterally along the housing moves the centre section to the new position, *i.e.* anticlockwise as shown on the sketch in Fig. 97. The oil pressure is maintained on the clutch during running so that the mating clutch faces are kept firmly in contact with no chatter.

There are a number of variations on this design but the principle of operation is similar although not all types rotate the clutch to its new position before starting and merely allow the camshaft to "catch on" with the crankshaft rotation when lost motion is completed.

FIG 97
LOST MOTION CLUTCH (EARLY SULZER)



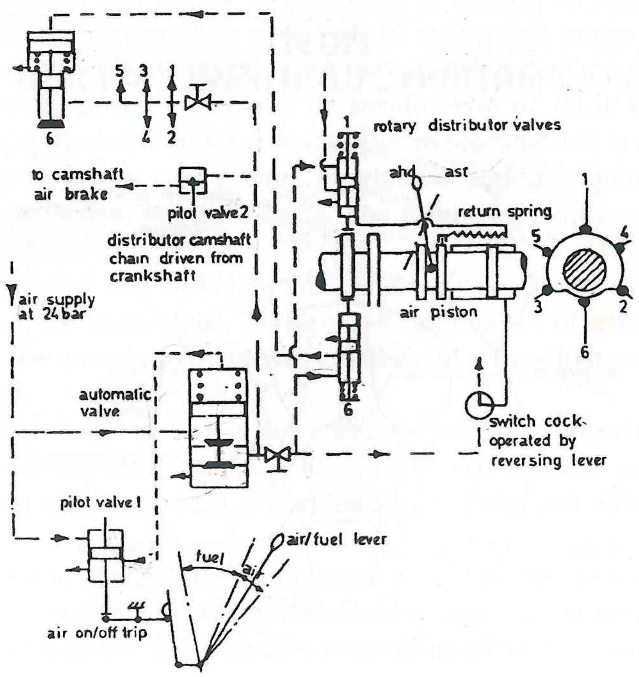
PRACTICAL SYSTEMS

Having described the basic principles of starting and reversing the actions are now combined to give a selection of systems as used on the various engine types.

Starting Air System (B & W.)

Consider first the air off position. Air from the storage bottle passes to the automatic valve which however remains shut as air passes through the pilot valve (1) to the top of the automatic valve piston. All cylinder valves and distributor valves are venting to atmosphere via the automatic valve. If now the lever is moved to the position shown in Fig. 98. then the air pressure on top of the automatic valve is vented through the pilot valve (1) by the linkage shown. This causes the automatic valve to open as the up force on

FIG 98
STARTING AIR SYSTEM (B & W)



the larger piston is greater than the down force on the smaller valve with the spring force. The lower vent connection is closed and air flows to all cylinder and distributor valves.

The cylinder valves are of the air piston relay type described earlier and in spite of main air pressure on them will be closed except for one valve (or possibly two). This distributor has the piston pilot valves mounted around the circumference of a negative cam. Only one distributor pilot valve can be pushed into the negative cam slot, *i.e.* No. 6. and hence air flows through the No. 6. distributor pilot valve only to the upper part of the piston for the No. 6. cylinder air starting valve, which will open. All other starting valves are shut and venting to atmosphere. The position shown for illustration is air on to No. 6. cylinder of a 6-cylinder engine running ahead. When the lever is moved forward on to fuel the whole system is again vented to atmosphere through the automatic valve.

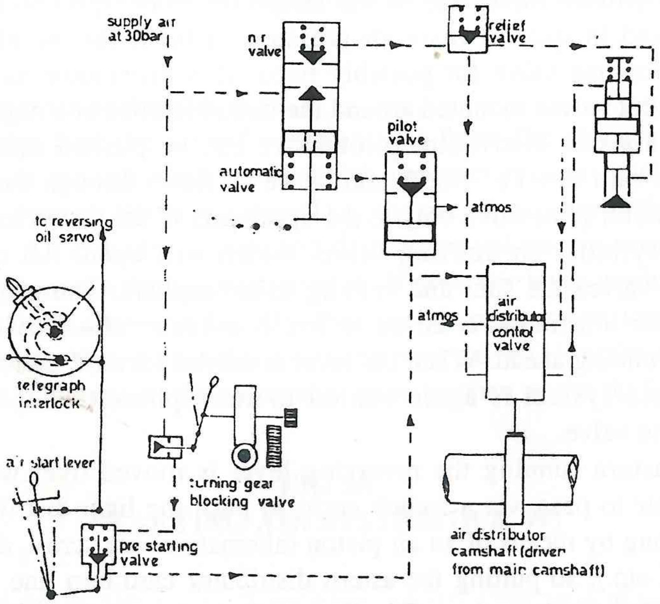
For astern running the reversing lever is moved over which allows air to pass, via a switch cock, to push the light distributor shaft along by means of an air piston (alternatives are scroll, direct linkage, etc.), so putting the astern distributor cam into line with the distributor pilot valves. Distributor pilot valves are kept out by springs during this operation. The air-fuel lever is then operated as previously described for the engine to run astern. Air start timing for a 2-stroke engine, upon which the above system is typical, is 5° before firing dead centre to 108° after firing dead centre (122° after for astern). B & W. engines also employ a revolving plug type of distributor on some engine designs. Again some types of these engines utilise an air brake on the main camshaft so that air pistons pressure against the pilot valve (2), operated from the reversing lever, while the lost motion is being travelled by the engine. The main camshaft is therefore kept stationary and just before the lost motion is complete the air pressure is released to atmosphere so releasing the brake.

Starting Air System (Sulzer RND).

Refer to Fig. 99.

Air from the starting receiver at 30 bar maximum flows to the pre-starting valve (via the open turning gear blocking valve shown), and directly to the automatic valve. At the automatic valve

FIG 99
STARTING AIR SYSTEM (SULZER RND)



air passes through the small drilled passage to the back of the piston and this together with the spring keeps this valve shut as the pilot valve is shut with air pressure on top and atmospheric vent below.

If the air starting lever is operated with control interlocks free, the opening of the pre-starting valve allows air to lift the pilot valve, vent the bottom of the automatic valve and cause it to open as shown. This allows air to pass to the cylinder valve via non-return and relief valves and also to the distributor. The distributor will allow air to pass to the appropriate cylinder valve causing it to open due to air pressure on the piston top. In this design when the piston top of the cylinder valve is connected to the atmosphere for venting, the bottom of the valve is connected to air pressure, this ensures a rapid closing action. The distributor of this engine is very similar in principle to that shown for the B & W. engine previously except that a positive cam is used by Sulzer.

A mechanical interlock is provided as a blocking device from the telegraph as shown. There is also a connection to the reversing

oil servo and an interlock connection from the reversing system to the air start lever via a blocking valve. These are described for the next sketch. Fig. 100.

Hydraulic Control System (Sulzer RND)

Consider a reversing action from ahead to astern.

Oil pressure from left of reversing valves to right of the clutch and under relay valve A. and air block valve.

The telegraph reply lever on the engine telegraph is first moved to stop and the fuel lever moved back to about notch $3_{1/2}$, the starting lever is mechanically blocked by the linkage shown Fig. 99.

The telegraph linkage to the reversing valve moves this valve and releases oil pressure from the lost motion clutch. This drop in pressure causes relay valve A. to move down by spring action which relieves pressure on the block piston (fuel) so cutting off fuel injection. The pressure on the block valve (cam) is also relieved which serves to also lock the starting lever.

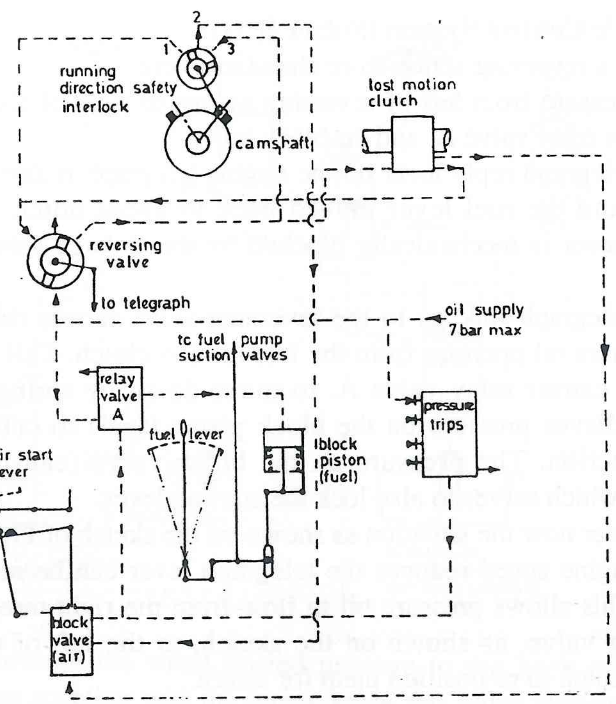
Consider now the situation as shown on the sketch of Fig. 100. When engine speed reduces the telegraph lever can be moved to astern. This allows pressure oil to flow from the right through the reversing valve, as shown on the sketch, to the left of the lost motion clutch to re-position them for astern.

When the servo has almost reached the end of its travel, pressure oil admitted to the block valve (air) releases the lock on the air start lever. (The mechanical lock on the air lever with the telegraph had been released when the telegraph lever was moved to the astern running position.) Pressure oil also acts on relay valve A. admitting oil to block piston (fuel) so allowing the fuel control linkages to the fuel pumps to assume a position corresponding to the load setting of the fuel lever.

If the pressure trips act in the event of low oil pressure (supply and bearings) or low water pressure (jacket or piston) then a trip piston moves up under preset spring pressure so connecting the oil pressure connection to drain. This pressure drop causes the block piston (fuel) to rise up under its spring force and shut off fuel injection.

Connections 1 and 3 from the running direction safety interlock to the reversing valve only allow fuel to the engine if the rotation agrees with the telegraph position. If not, the block piston (fuel) is

FIG 100
HYDRAULIC CONTROL SYSTEM (SULZER RD)



relieved of pressure via block valve (air) and relay valve A.

Movement of the air starting lever can now be carried out as both locks have been cleared and subject to no trip action and satisfactory correspondence between rotation direction and telegraph reply lever indication fuel can be admitted following the full sequence of air starting as described previously, and illustrated in Fig. 99. It is obvious that this system has a large amount of auto-control and is easily adjusted for bridge control.

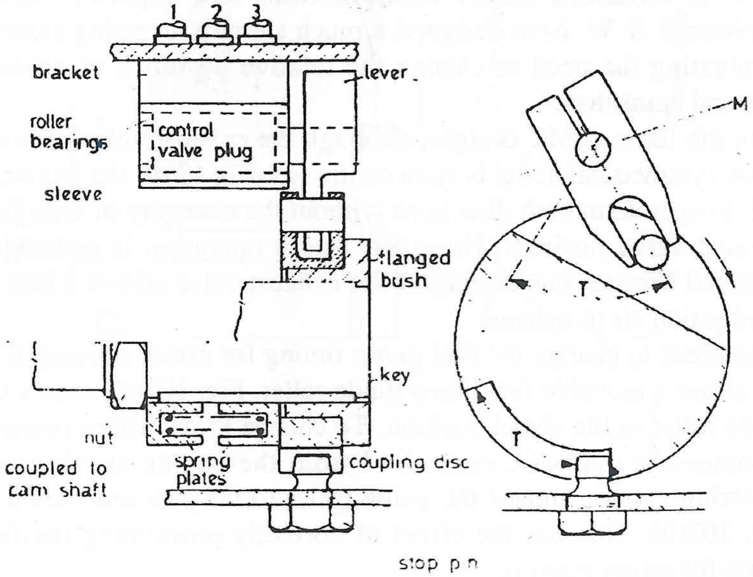
Control Gear Interlocks

There are many types of safety interlocks on modern IC engine manoeuvring systems. The previous few pages have picked out a number relating to the Sulzer RND engine and these will be adequate to cover most engine type designs as principles are all very similar.

Consider the interlock systems illustrated in Fig. 100. and 101. The telegraph and turning gear interlocks are straight mechanical linkages. In the former case rotation of the telegraph lever from stop position causes the pin to travel in the scroll and unlock the air start lever as well as re-position the reversing valve. The turning gear blocking valve can be seen to close when the pinion is placed in line with the toothed turning gear wheel of the engine. The interlock exerted on the block piston (fuel) is also a fairly simple principle working on the relay valve A. from the pressure trips and is as described previously.

Similarly the block valve (air) operates mechanically via the lever lock on air start lever and horizontal operating lever which rises to unlock under the oil pressure acting through the Servo on block valve (air) after the clutch reversals have taken place. (The pressure trips are merely spring loaded pistons moving against low oil or water pressure to relieve control oil pressure just like conventional relief valves.) It is perhaps appropriate here to describe one trip in detail and the direction safety lock will now be

FIG 101
SAFETY LOCK FOR CORRECT ROTATION (SULZER RND)



considered briefly. The function is to withhold fuel supply during manoeuvring if the running direction of the engine is not coincident with the setting of the engine telegraph lever. Refer now to Fig. 101.

At the camshaft forward end the shaft is coupled to the cam shaft and carries round with it, due to the key, a flanged bush and spring plates which cause an adjustable friction pressure axially due to the springs and nut. This pressure acts on the coupling disc which rotates through an angular travel T until the stop pin prevents further rotation. This causes angular rotation of a fork lever and the re-positioning of a control valve plug in a new position within the sleeve. Oil pressure from the reversing valve can only pass to the block valve (air) and unlock the air start lever and the fuel control if the rotation of the direction interlock is correct. If the stop pin were to break the fork lever would swing to position M and the fuel supply would be blocked.

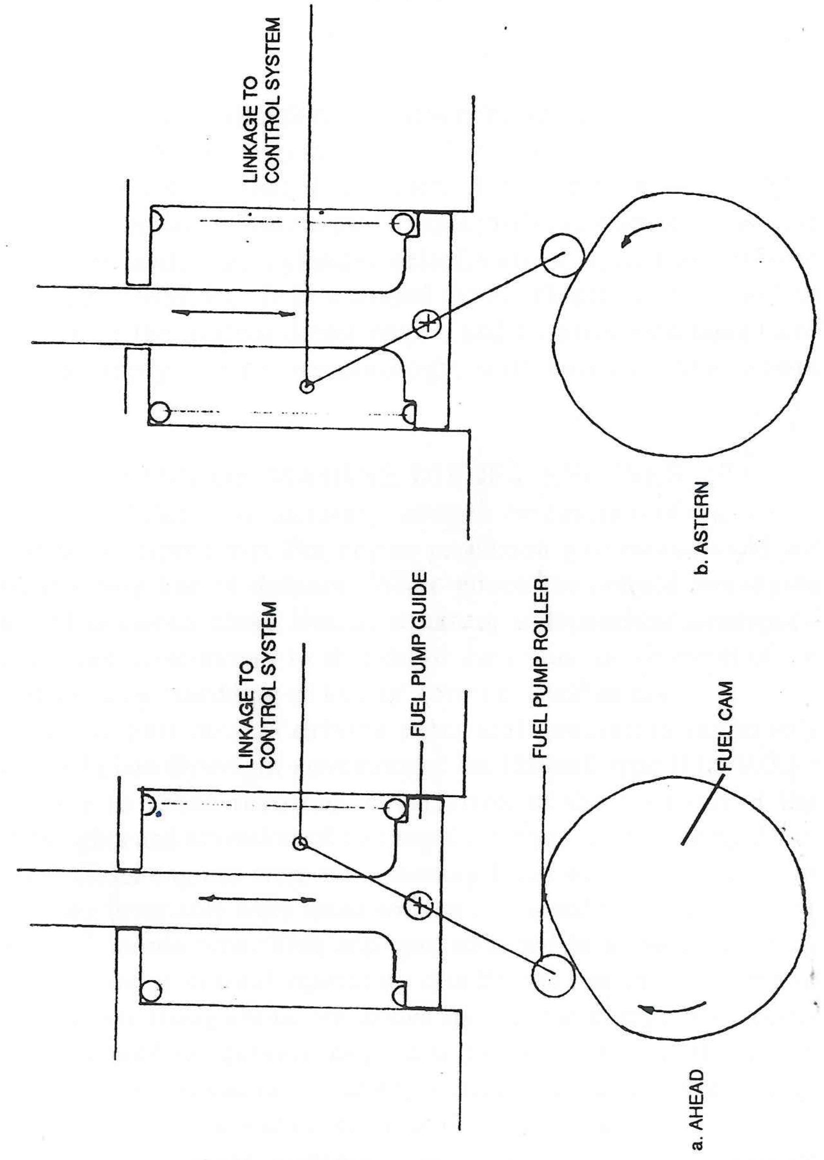
Modern Reversing Systems

In the previous sections reversal was carried out by utilising lost motion or by moving the camshaft axially to utilise a different set of cams. It can be seen that these methods involve added complication in the running gear and control systems of engines. In order to eliminate undue complications and improve engine response B & W. have designed a much simpler reversing system, eliminating the need to change the relative positions of exhaust cam and crankshaft.

In the latest L-MC designs, although the exhaust valve opening is not symmetrical about bottom centre (see Fig 13 d), the engine is able to operate in both directions without the necessity of changing exhaust valve position. However, astern operation is somewhat impaired because late closing of the exhaust valve allows a loss of combustion air to exhaust.

In order to change the fuel pump timing for astern running B & W. utilise a movable fuel pump guide roller. Fig. 102(a). shows the guide roller in the ahead position. To change to the astern position a pneumatic cylinder, controlled from the engine starting and reversing system, moves the guide roller to the position shown in Fig. 102(b). This has the effect of correctly positioning the fuel pump for astern running.

FIG 102
REVERSING MECHANISM OF MODERN B&W ENGINES



CHAPTER 6

CONTROL

The study and application of instrument and control devices has developed from the beginning of engineering itself. It is however in the last few years that this branch of engineering has assumed greater importance. Automatic Control in a simple sense has always been utilised, *e.g.* cylinder relief valves, speed governors, overspeed trips, etc. It is intended in this chapter to examine the control of the modern diesel engine and its associated equipment and to apply control terminology, with explanations, where required.

GOVERNING OF MARINE DIESEL ENGINES

A clear distinction is necessary between the function of a governor and an overspeed trip. For engine protection governors should not be the only line of defence. While governors control the engine speed between close limits, separate independent overspeed protection is necessary to shut down the engine in the event of the instantaneous shedding of load or governor malfunction.

In the past diesels driving electrical generators invariably utilised plain flyweight governors of the Hartnell type (Fig. 103.) a change in speed resulted in variation of the position of the flyweights and alteration of fuel supply. Larger, slow running direct drive diesel engines were not generally fitted with such a governor but they invariably were fitted with an overspeed trip, usually of the Aspinal inertia type. This trip was arranged to allow full energy supply under normal operating conditions but in the event of revolutions rising about 5% above normal the energy was totally shut off until revolutions dropped to normal again. At about 15% above normal revolutions the trip would stay locked, with energy shut off, and this would continue until re-set by hand.

A plain flyweight governor, which has to perform two separate functions (1) to act as a speed measuring device, (2) to supply the necessary power to move the fuel controlled system. Fig. 104. shows in block diagram form the arrangement. A closed loop

FIG 103
MECHANICAL GOVERNOR

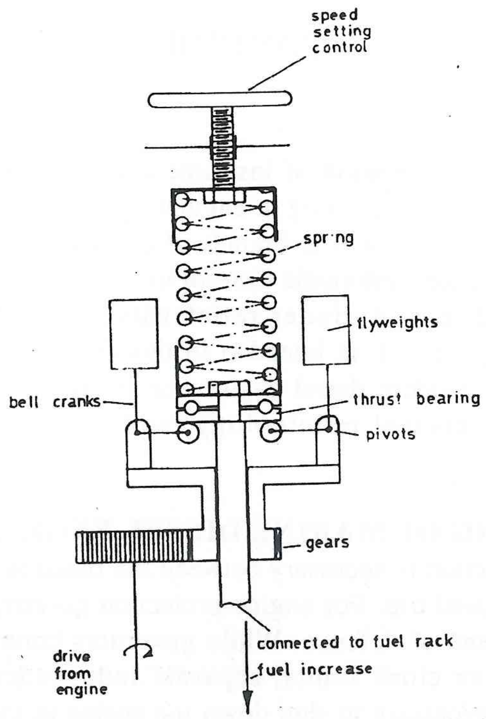
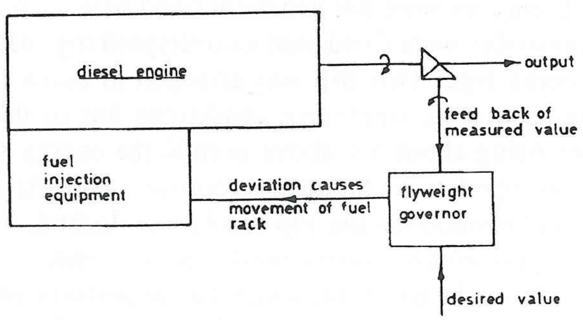


FIG 104
CLOSED LOOP CONTROL



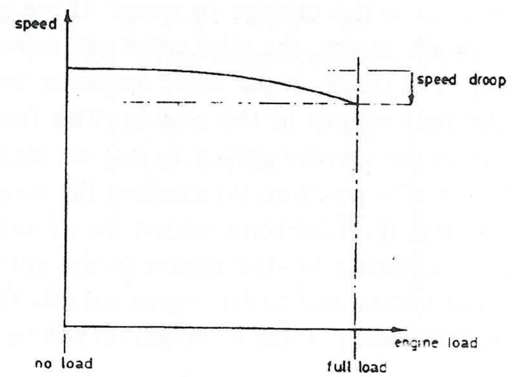
control system is one in which the control action is dependent on the output. The measured value of the output, in this case the engine speed, is fed back to the controller which compares this value with the desired value of speed. If there is any deviation between the values, measured and desired, the controller produces an output which is a function of the deviation. In this case the controller output would be proportional to the deviation, *i.e.* proportional control.

In control terminology deviation is sometimes called error, since it is the difference between measured and desired values, and desired value is sometimes called set value. Proportional control suffers from offset. In the example, if a speed change occurs the flyweights take up a new equilibrium position and the fuel supply will be altered to suit the new conditions. However, the diesel is now running at a slightly different speed to before. If the original speed was the desired value then the new speed is offset from the desired value.

In governor parlance the term speed droop, or just droop, is used to define the change in speed between no load and full load conditions. If speed droop did not exist then there would be one speed only for any position of the governor flyweights and this in turn means any fuel supply rate. In this case the diesel would hunt.

This is an isochronous condition, an engine fitted with an isochronous governor will hunt. However, the term isochronous has taken a new meaning as we will find later.

FIG 105
SPEED DROOP



Forces involved in the flyweight governor movement are, inertia, friction and spring. Considerable effort may therefore be required to cause movement, this would necessitate a change in speed without any alteration in governor position. This is bad control, the system is insensitive and various equilibrium speeds are possible. For simple systems these various equilibrium speeds are not an embarrassment, but if we require a better controlled system the two functions that the flyweight governor has to perform would be separated into (1) a speed measuring device and (2) a servo-power amplifier.

Fig. 106. shows the basic arrangement in block form. A load increase would cause a momentary speed droop. The speed measuring device would obtain a measured value signal from the diesel and compare this with a desired value from the speed setting control. The deviation would be converted into an output that would bring into action the servo-power amplifier which would position the fuel rack, increasing the supply of fuel to meet the increase in load.

Since the speed measuring device does not have to position the fuel rack – in fact it could be near zero loaded – it can be very responsive, minimising the time delay between load alteration and fuel alteration in the closed loop. The servo-power amplifier is usually a hydraulic device that simply, quickly and effectively provides the necessary muscle to move the fuel rack.

A proportional action governor is diagrammatically shown in Fig. 107. The centrifugal speed measuring unit is fitted with a conically shaped spring, unlike that shown in Fig. 103., this gives a spring rate which varies as the square of the speed. Fig. 108. This gives linearity to the speed measuring system, *i.e.* the response is directly proportional to the change in speed. If we consider an increase in load on the engine, the pilot valve will move down due to the speed drop. The piston in the servo-amplifier will move up and increase the fuel supply to the engine. The feedback link reduces the force in the speeder spring so that the flyweights can move outwards to a new position, thus raising the pilot valve and closing off the oil supply. If for some reason the oil supply system should fail then the spring loaded piston in the servo-cylinder would be moved down and fuel to the engine cut off. This is called fail safe. Any oil that leaks past the servo-piston will be drained off

FIG 106
BASIC ARRANGEMENT

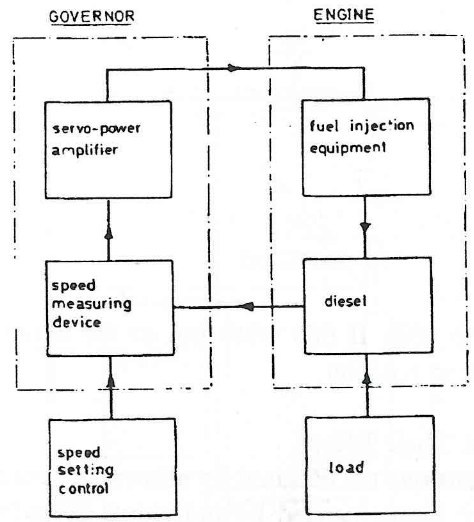


FIG 107
PROPORTIONAL ACTION GOVERNOR

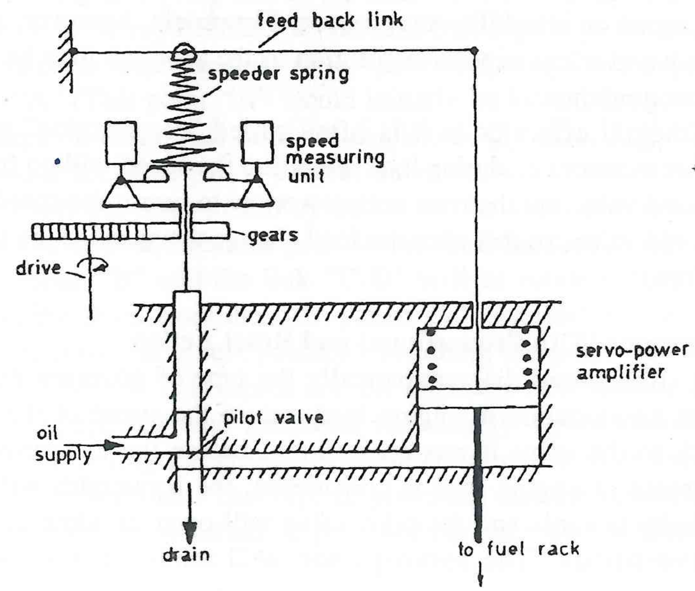
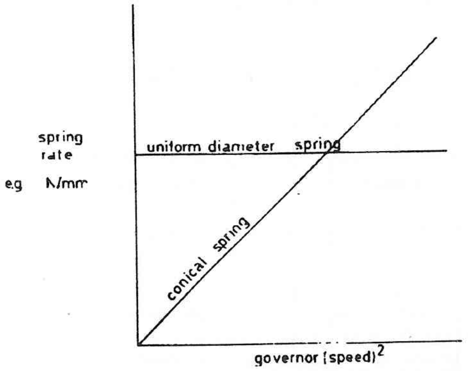


FIG 108
GOVERNOR (SPEED)



to the oil sump tank. If this were not so the servo-piston would eventually lock in position.

Flywheels and Their Effect

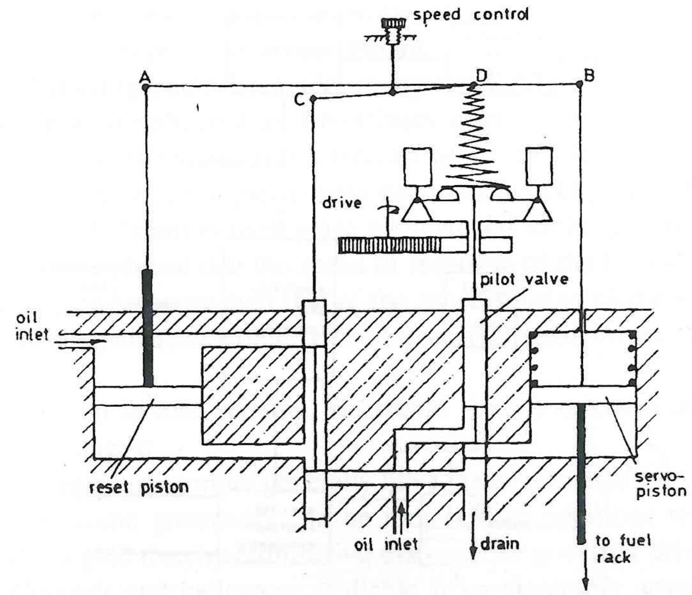
Flywheel dimensions are dictated by allowable speed variation due to non-uniform torque caused by individual cylinders firing. This of course is outside the control of the governor. If the speed has to remain nearly constant during changes of load it may be decided to fit a large flywheel, which increases the moment of inertia of the system and gives an integral effect – this must not be taken to extremes or instability may occur. Flywheels, however, are not cheap and a less expensive solution to the problem may be to fit a better governor.

Integral effect or as it is often called "reset action" reduces offset to zero, *i.e.* during load alteration the speed will go from the desired value but the reset action worked to return the speed to the desired value, so that after the load change the speed is the same as before.

Governor With Proportional and Reset Action

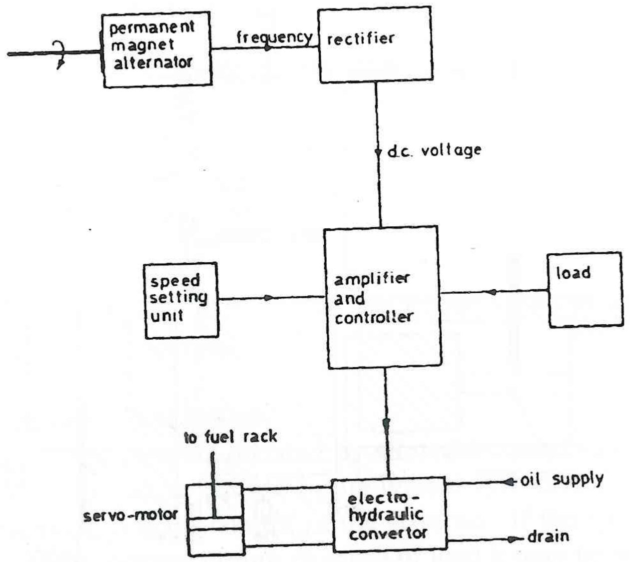
Fig. 109. shows diagrammatically the type of governor that will, after an alteration in engine load, return the speed of the engine back to the value it was operating at before the alteration. If an increase in engine load is considered, the flyweights will move radially inwards and the pilot valve will open to admit oil to the servo-piston. The servo-piston will move up the cylinder

FIG 109
GOVERNOR WITH PROPORTIONAL AND RESET ACTION



compressing the spring and at the same time it will cause (1) the fuel rack to be repositioned to increase fuel supply to the engine, (2) rotate the feedback link "A-B" anti-clockwise about the pivot point "A" (This point "A" would initially be locked due to equal pressures on either side of the reset piston), (3) rotate link "C-D" will move the reset piston control valve down and some oil will drain from the reset piston cylinder. As the reset piston moves down to a new equilibrium position the feedback link "A-B' will pivot about "B" and the link "C-D" will be rotated clockwise, closing the drain from the reset piston cylinder (and thus locking the reset piston in a new position), returning the point "D" to its original position. This means that the engine is now running at its original speed but with increased fuel supply. Speed droop that took place during the change of the relative positions of the two pistons was transient. This type of governor, that has proportional and reset action, is called in governor parlance an "isochronous governor".

FIG 110
ELECTRIC GOVERNOR



Electric Governor

This governor has proportional and reset action with the addition of load sensing. A small permanent magnet alternator is used to obtain the speed signal, the advantage to be gained is that there will be no slip rings or brushes with their attendant wear. The speed signal obtained from the frequency of the generated a.c. voltage impulses is converted into d.c. voltage which is proportional to the speed. A reference d.c. voltage of opposite polarity, which is representative of the desired operating speed, is fed into the controller from the speed setting unit. These two voltages are connected to the input of an electric amplifier. If the voltages are equal and opposite, they cancel and there will be two voltages are equal and opposite, they cancel and there will be no change in amplifier voltage output. If they are different, then the amplifier will send a signal through the controller to the electro-hydraulic converter which will in turn, via the servo-motor, reposition the fuel rack. In order that the system be isochronous the amplifier controller has internal feedback.

Load Sensing

The purpose of including load sensing into the governor is to correct the fuel supply to the prime mover before a speed change occurs. Load sensing governors are therefore anticipatory governors, *i.e.* they anticipate a change in speed and take steps to prevent, as far as possible, its occurrence.

Load sensing could be achieved by mechanical means but it would be a complicated and relatively costly system. For this reason load sensing governors tend to be of the electronic type. The output of, for example, a main generator would be monitored and if a load alteration took place a signal fed to the governor. It must be remembered that the speed of response of the load sensing element must be better than that of the speed sensing element. The speed sensing element would be used to correct small errors of fuel rack position.

In addition to load sensing, electronic governors claim several other advantages:

1. Electronic governors generally have faster response
2. Electronic governors can be mounted in positions remote from the engine thereby eliminating the need for governor drives.
3. Controls and indicators available from electronic governors make automation easier.
4. Control functions, for example, fuel limitation, acceleration and deceleration schedules and shut down functions such as low lubricating oil pressure can be built into the electronic governor.

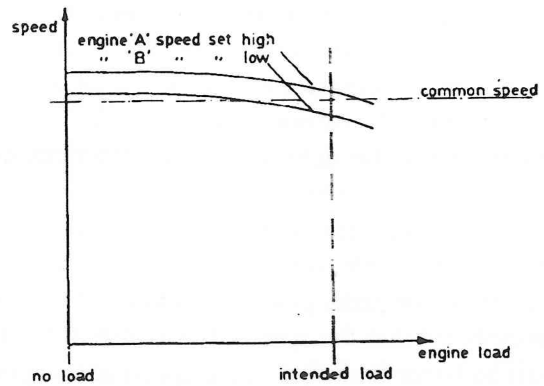
Geared Diesels

Two diesels geared together must run at the same speed, but if the governors of the two are not set equally then they will not carry equal shares of the load.

In Fig. 111. is shown the governor droop curves for two diesels A and B. Governor A has a higher speed setting than that of B, but since they must both run at a common speed the load carried by A will be greater than that of B. Actual load carried is given by the intersection of the common speed line and the droop curves. By adjusting the speed settings both droop curves could be made to coincide at the intended load, although this would be difficult to achieve in practice.

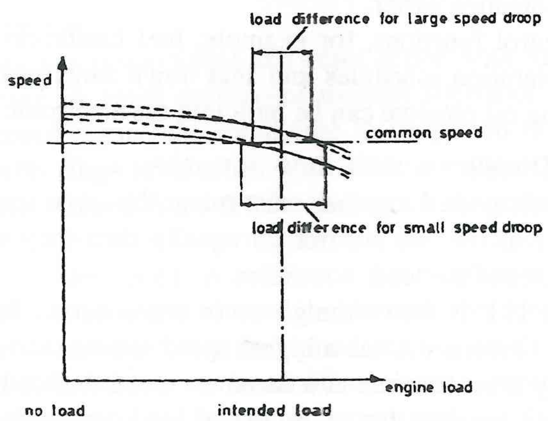
Shown in Fig. 112. are two sets of droop curves with the same difference in speed settings but with different amounts of speed

FIG 111
LOAD SHARING BETWEEN TWO ENGINES



droop. The difference in load sharing at the common speed is less for the larger speed droop curves than for the smaller. Hence speed droop and fine control over the desired level of speed are necessary for effective load sharing.

FIG 112
LOAD DIFFERENCE



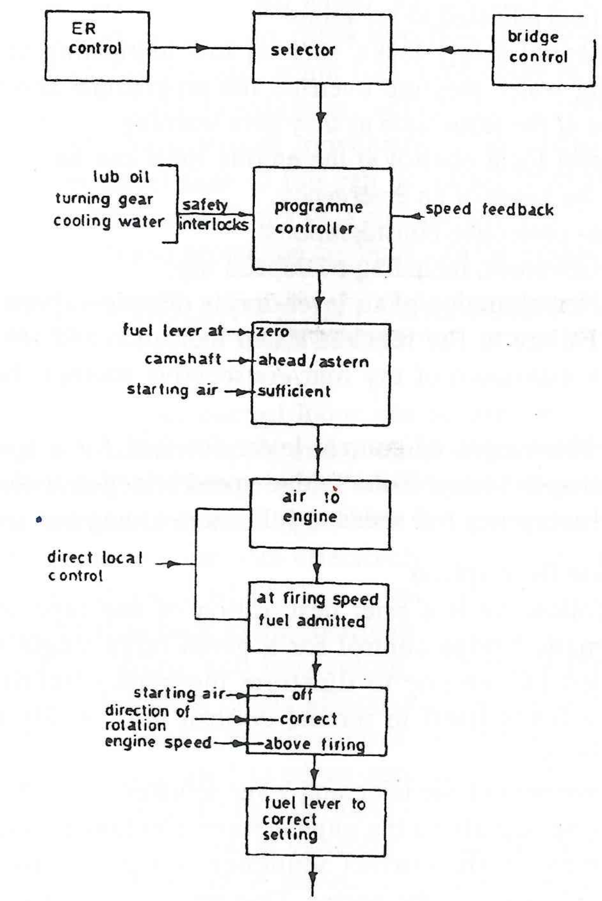
Bridge Control of Direct Drive Diesel Engine

Two consoles would be provided, one on the bridge the other in the engine room. For the bridge console the minimum possible alarms and instruments would be provided commensurate with

safety and information requirements, *e.g.* low starting air pressure and temperature, sufficient fuel oil, fuel oil pressure and temperature, etc. The engine room console would give comprehensive coverage and overriding control over that of the bridge.

In Fig. 113. for simplification all normal protective devices are assumed and subsidiary control loops are not considered. The selector would be in the engine room console and the operator can

FIG 113
ENGINE CONTROL PROGRAMME



select either engine room or bridge control, with one selected the other is inoperative. Assuming bridge control a programme would be selected, say half ahead, Then providing all safety blockages such as no action with turning gear in, etc. are satisfied, the programme can be initiated and could follow a sequence of checks and operations such as:

1. Fuel control lever at zero.
2. Camshaft in ahead position.
3. Sufficient starting air.
4. Starting air admitted.
5. Adjustable time delay permits engine to reach firing speed.
6. Fuel admitted.
7. Starting air off, checks on direction of rotation and speed.
8. Fuel adjusted to set value.

Essential safety locks, such as low lubricating oil pressure or cooling water pressure override the programme and will stop the engine at the same time as they give warning.

Direct local control at the engine itself can be used if required on in the event of an emergency.

Further protective considerations:

1. Governor, including overspeed trip.
2. Non operation of air lever during direction alteration.
3. Failure to fire requires alarm indication and sequence repeat with a maximum of say four consecutive attempts before overall lock.
4. Movement of control lever for fuel for a speed out of a critical speed range if the bridge speed selection within this range.
5. Emergency full ahead to full astern timing and setting.

Outline Description

The following is a brief description of one type of electronic-pneumatic bridge control for a given large single screw direct coupled I.C. engine to illustrate the main essentials. The I.C. engine lends itself to remote control more easily than turbine machinery.

Movement of the telegraph lever actuates a variable transformer so giving signals to the engine room electronic controller which transmits, in the correct sequence, a signal series to operate solenoid valves at the engine. One set of solenoid valves controls starting air to the engine while a second set regulates fuel supply,

the latter via the manual fuel admission lever, is coupled to a pneumatic cylinder whose speed of travel is governed by an integral hydraulic cylinder in which rate of oil displacement is governed by flow regulators. This cylinder also actuates a variable transformer giving a reset signal when fuel lever position matches telegraph setting.

With the engine on bridge control the engine control box starting air lever is ineffective and the fuel control rack is held clear of the box fuel lever. Engine override of bridge control is provided.

The function of the electronic controller is to give the following sequence for, say, start to half ahead: Ensure fuel at zero, admit starting air in correct direction, check direction, time delay to allow engine to reach firing speed, admit fuel, time delay to cut off air, time delay and check revolutions, adjust revolutions. Similar functions apply for astern or movements from ahead to astern directly. Lever travel time to full can be varied from stop to full between adjustable time limits of 1/2 minute and 6 minutes. Fault and alarm circuits and protection are built into the system.

PISTON COOLING AND LUBRICATING OIL CONTROL

Simple single element control loops can be used for most of the diesel engine auxiliary supply and cooling loops, however during the manoeuvring of diesel engines considerable thermal changes take place with variable time lags which the single element control may not be able to cope with effectively. (N.B. a single element control system is one in which there is only one measuring element feeding information back to the controller.)

For piston cooling and lubricating oil control the use of a cascade control system caters effectively for manoeuvring and steady state conditions. Cascade control means that one controller (the master) is being used to adjust automatically as required the set value of another controller (the slave).

In Fig. 114. the two main variables to consider are sea water inlet temperature and engine thermal load. For simplicity we can consider each variable separately:

1. Assuming the engine thermal load is constant and the sea water temperature varies. The slave controller senses the change in