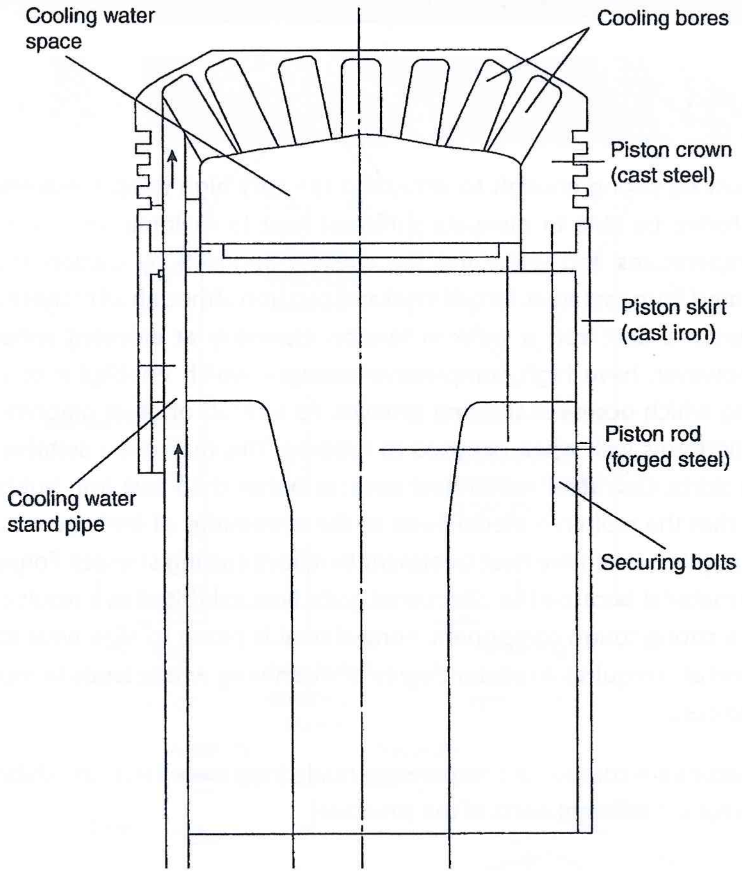
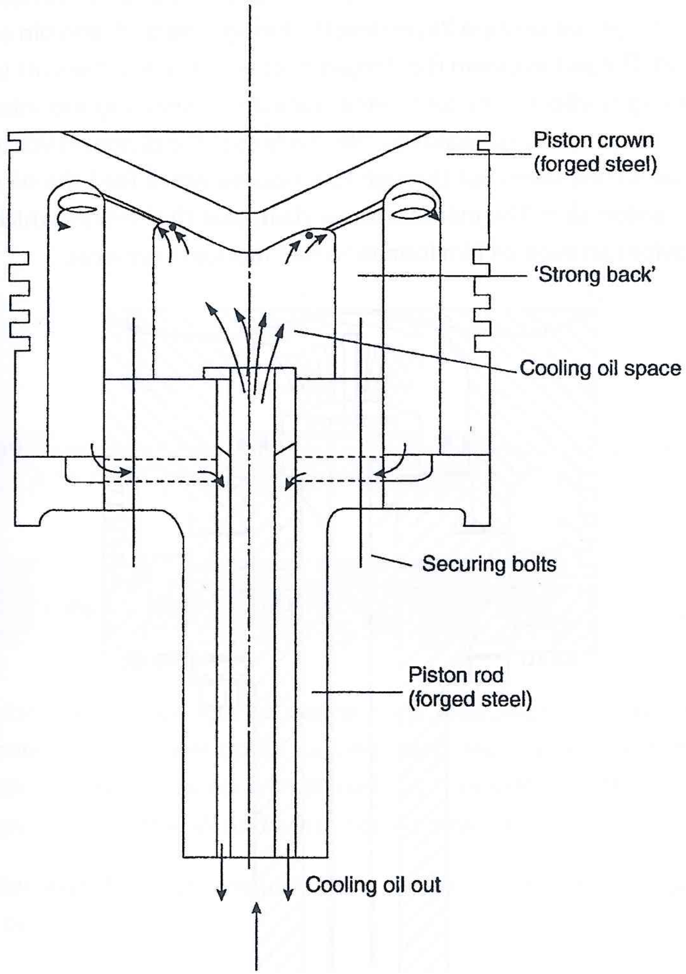
- Piston skirts are made from cast iron which has superior rubbing properties than either cast or forged steel. To reduce weight and reduce inertia loads aluminium is used in some medium-speed four-stroke applications.

The design trend with marine diesels engines has moved away from using water as a cooling medium for the piston. The hazard of introducing water into the scavenge space or the crankcase has led manufacturers away from this practice. Although the designs are still included in flag state examinations because the sea going engineer may find an example still in service and will need to be prepared to look after the system while on an ocean going passage. Figure 2.36 however shows a piston for a large Sulzer (now part of the Wärtsilä Corporation) engine. The crown is of forged steel and combines strength with good heat transfer.



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

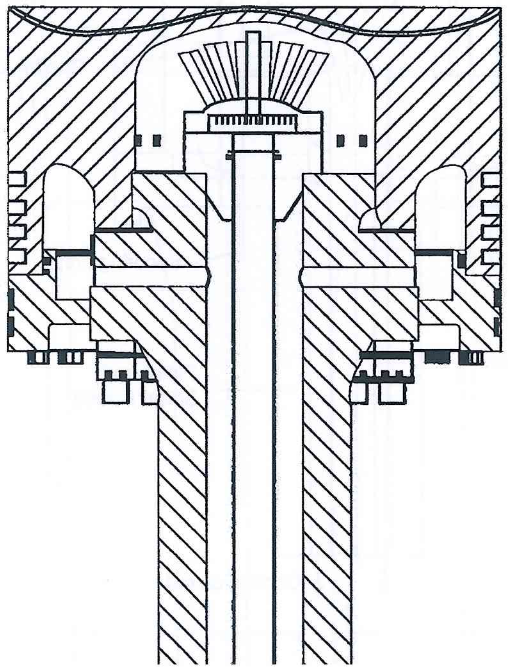
Figure 2.37 shows an oil-cooled piston typical of the design of the large two-stroke engines in service today. The piston crown of this piston is also manufactured from forged steel but in this case the section is relatively fine. Strength being achieved by

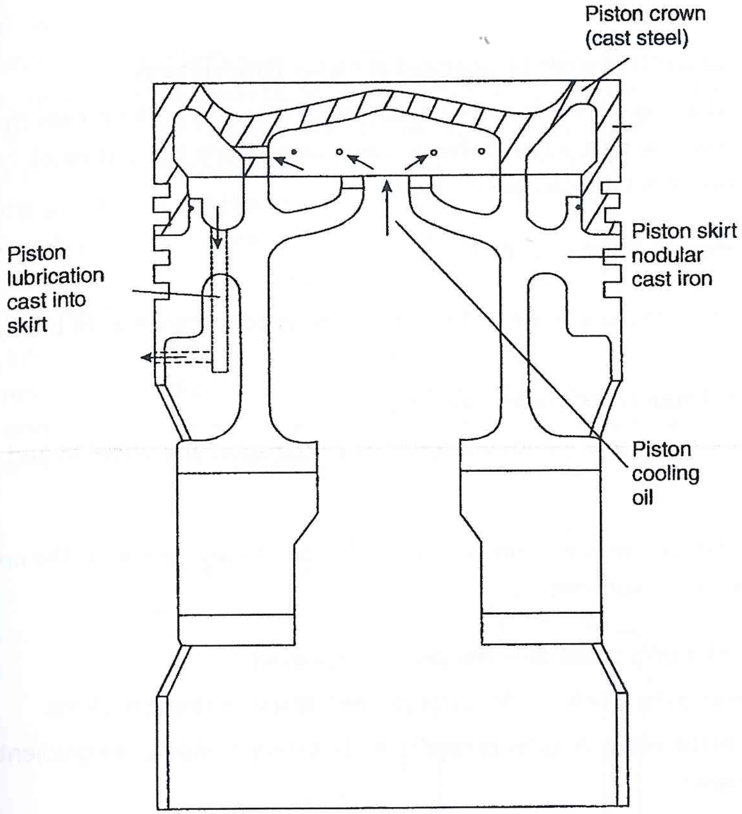


the 'strong back' principle which supports the piston crown from inside. Bore cooling is employed by some designs of large bore two-stroke engines. In this design the bores act as nozzles through which the oil flows radially, spraying onto the underside of the piston crown, before flowing to the drain (similar to the water-cooled design).

The latest large bore two-stroke designs (figure 2.38) have a series of branches extending from the top of the piston rod to direct the oil onto the underneath of the piston crown enhancing the cooling effect of the oil on the piston. The oil is fed up the middle of the piston rod and flows back through channels in the outer part of the piston rod.

Figure 2.39 shows a piston from a medium-speed four-stroke engine. This type of engine design transmits the combustion forces directly through the gudgeon pin and onto the connecting rod. The piston crown is of forged or cast steel while the skirt is of nodular cast iron. Cooling is effected by oil flowing from the connecting rod into the piston crown then flowing radially outwards to effectively cool the piston. In Wärtsilä engines some of this oil is then taken out through four nozzles which feed the oil distribution groove in the piston skirt. The manufacturers claim that this design, which they have patented, provides an even oil film formation that reduces liner wear.





▲ **Figure 2.39** *Composite piston suitable for a high-output medium-speed four-stroke diesel engine*

The choice of water or oil for piston cooling

The choice for the latest designs of marine main propulsion engines is not an issue as all the manufacturers have opted for oil-cooled designs. However, it is expected that the modern marine engineer will still have an appreciation of the advantages and disadvantages between the oil- and water-cooled designs.

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

- Water-cooled pistons can be operated at higher temperatures.
- Water has a specific heat capacity nearly twice that of oil. (This means that, for the same mass flow rate, water is able to carry away nearly twice as much heat as oil, lower mass flow rates can be specified.)

The disadvantages of water are that:

- Leakage into the crankcase will result in serious contamination of the lubricating oil.
- Additional pumps and coolers are required.
- Telescopic pipes and glands are required to transport the water to and from the piston.

Oil is used extensively in modern engines and in all of the new builds. The advantages claimed for this medium are:

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

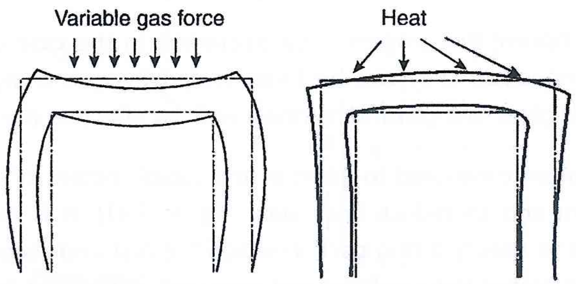
The disadvantages of oil are:

- The temperatures must be kept relatively low in order to limit oxidation of the oil.
- If overheating occurs there is a possibility that carbon deposits could form on internal surfaces and there is a danger that carbon particles could enter the lubricating oil system. This could have an insulating effect and make the cooling process less efficient.

Failure of pistons due to thermal loads

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

At normal working temperatures the piston and cylinder liner surfaces should be parallel. Since there is a temperature gradient from the top to the bottom of the piston, allowance must be made during manufacture for the top cold clearance to be less than the bottom. The temperature gradient is generally non-linear and thermal distortions produce tensile stresses on the inner wall of the piston, gas forces tend to bulge the piston wall out thereby reducing the tensile stress. This variable tensile stress at very high thermal loads could lead to cracks propagating through from the inside of the piston to the piston ring grooves. Modern engine designers use complex mathematical models to evaluate these stresses before the engine is manufactured. However, information from the running test engines is used to update the algorithms in the computer models. For this reason MAN Diesel have their four-cylinder 500-mm bore test engine at their research centre in Copenhagen and Wärtsilä have the RTX5 test engine in Trieste, Italy (figure 2.40).



▲ **Figure 2.40** *Effect of gas and heat*

Piston rings

Properties required of a piston ring are as follows:

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

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

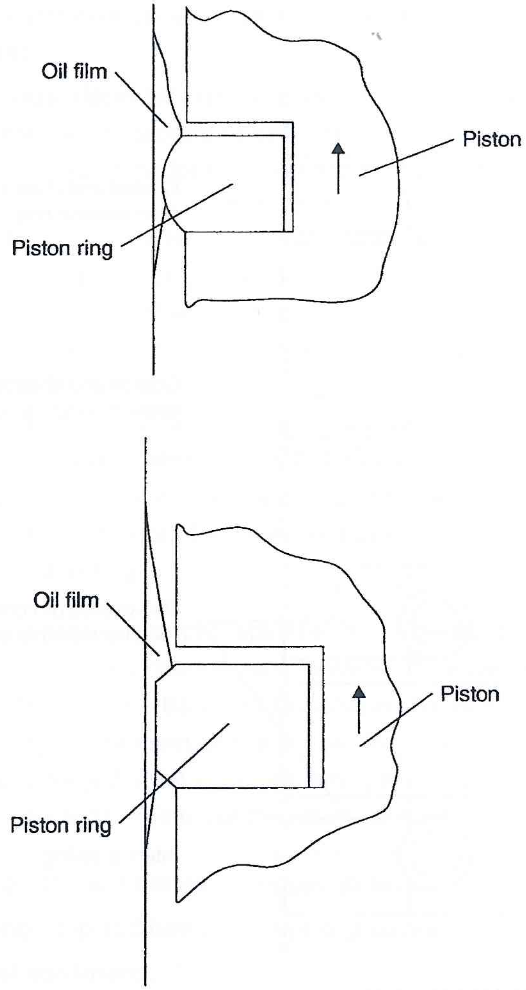
1. Ordinary grey cast iron, in order that it may have good wear resistance and self-lubricating property it must have a large amount of graphite in its structure. This however reduces its strength.
2. Alloyed cast iron, elements and combinations of elements that are alloyed with the iron to give finer grained structure and good graphite formation are: molybdenum, nickel and copper or vanadium and copper.
3. Spheroidal graphitic iron, very good wear resistance, not as self-lubricating as the ordinary grey cast iron. These rings are usually given a protective coating, for example, chromed or aluminised, etc. to improve running-in.

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

Piston rings are often contoured to assist in the establishment of a hydrodynamic lubricating oil film and so reduce liner wear (figure 2.41). It is common practice for manufacturers to specify a ring pack in which the first and second compression rings, subjected to higher temperatures and pressures, differ from the lower rings. It is important when installing new rings that the manufacturer's recommendations are followed since ring failure may result if incorrect rings are fitted.

In addition to compression rings four-stroke medium-speed engines also employ oil control, or oil scraper rings (figure 2.42). Unlike compression rings, which help promote the formation of an oil film, oil scraper rings scrape the oil from the cylinder liner and return it to the sump. Many designs of oil scraper rings can only be 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 four-stroke engines.

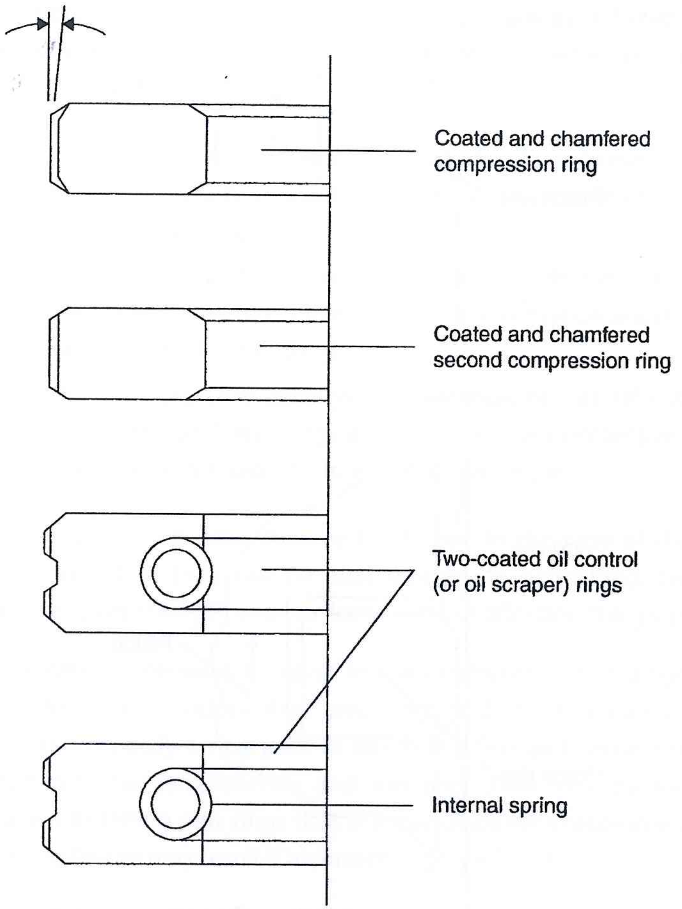
Manufacture



▲ **Figure 2.41** *Two-stroke engine piston ring profile*

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



▲ **Figure 2.42** *Four-stroke piston rings*

be capable of exerting a radial pressure from 2 to 3 bar and have a Brinel hardness from 1,600 to 2,300 (SI 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

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. If the cylinder liner has become worn at the top of the piston travel there could be a ridge from which the piston ring hits at the top of its travel. This ridge can be chamfered during unit overhaul to reduce the damage to the top piston ring.

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 two-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

procedures. Frequency of which depends upon numerous factors, such as: piston size; material and method of cooling; engine speed of rotation; type of engine, two- or four-stroke; fuel and type of cylinder lubricant used. The manufacturers of modern engines have the design aim of extending the time between overhauls. Modern research and development techniques have led to much better understanding of cylinder liner lubrication and modern materials are lasting much better than in the past. Therefore the student must be careful about answering a question about servicing intervals because although the manufacturers might still recommend servicing upon operational hours it will still be the responsibility of the ship's engineering staff to ensure that the engine is running efficiently.

With high-speed four-stroke diesel engines, as a general statement, the running time between piston overhauls used to be greater than that for large slow-speed two-stroke engines. This was 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 across all types of engines can now vary between about 10,000 and 32,000 h. In fact manufacturers look to extend the time between major overhauls as being the same as the time from one drydock to the next. Meaning that the main engines do not have to be opened up in-between docking periods.

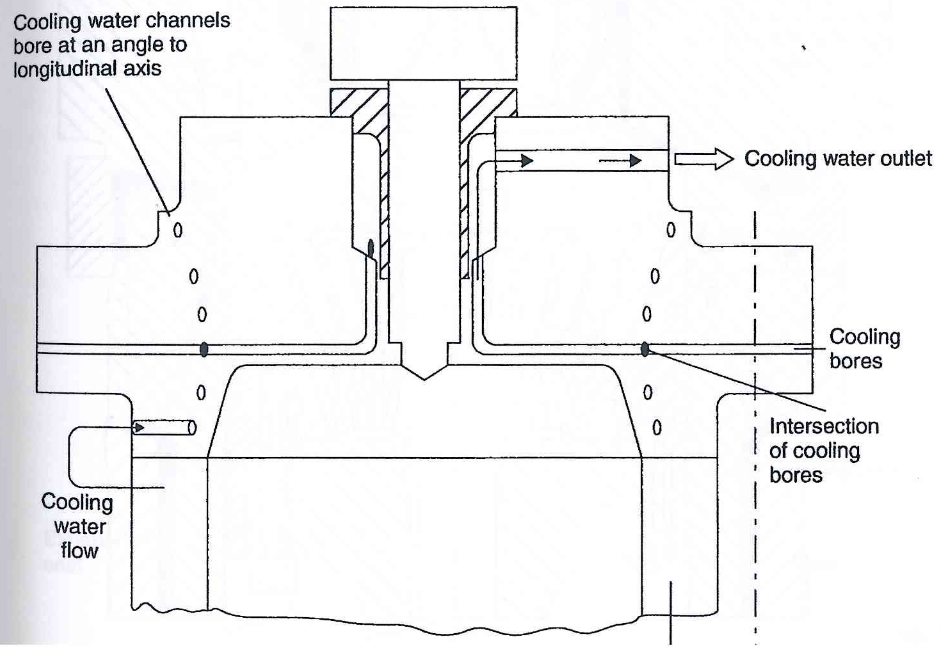
The modern two-stroke engine can have its pistons and cylinder liners inspected without having to remove the piston. After scavenge spaces have been cleaned of inflammable oil sludge and carbon deposits, each piston can in turn be placed at its lowest position. The cylinder liner surfaces can then be 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

Cylinder covers – large two-stroke engines

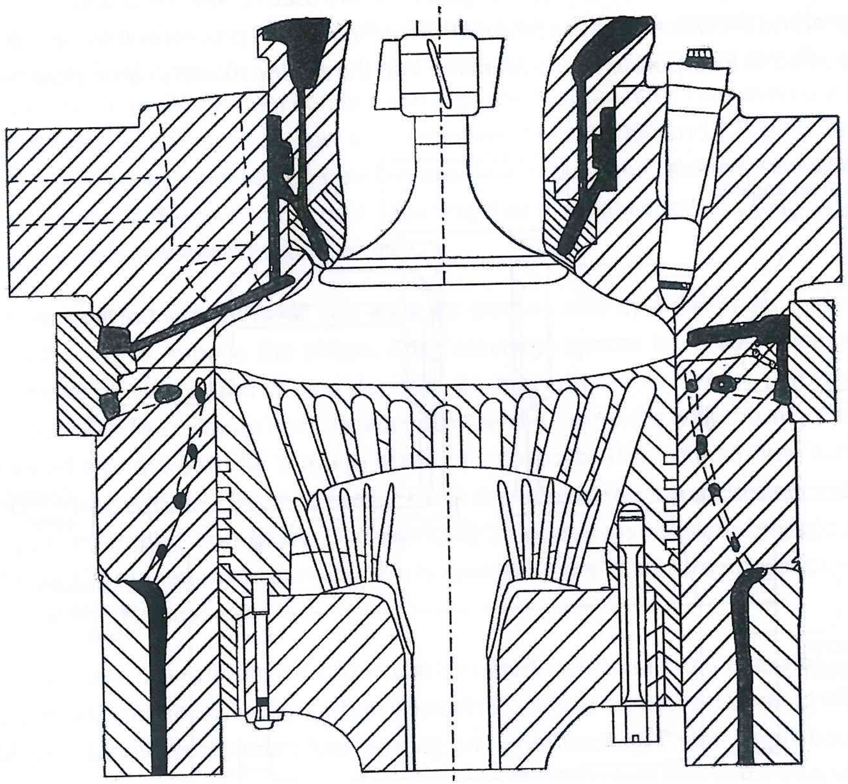
On the older loop scavenged two-stroke engines the cylinder covers tended to be a relatively simple symmetrical design to avoid the problems of differential expansion and the consequent stresses. Early Sulzer (Now Wärtsilä) designs consisted of two pieces with a cast iron main component and a central 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. 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 (figure 2.43).

MAN Diesel & Turbo, Mitsubishi and the Wärtsilä Corporation are currently the only designers of large two-stroke diesel engines that are used for marine propulsion. These designers do allow licensees to carry out the manufacturing process and some licensees are allowed to incorporate their own name into the engine model's name. However, all

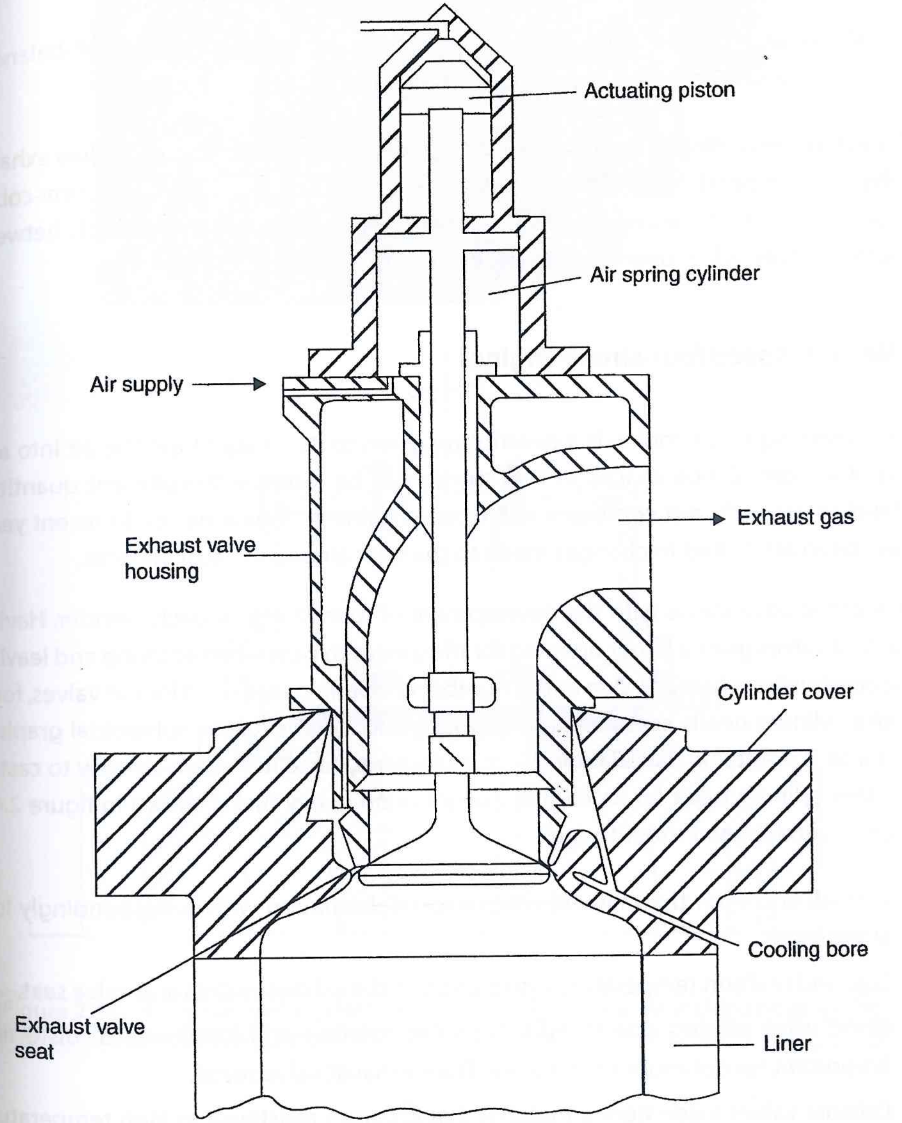


the basic designs have become standardised around the UNI-flow scavenging system. This system started life with opposed piston engines where the exhaust piston was arranged to uncover the exhaust ports. Approximately one third of the engine's power was transmitted through the exhaust piston and eccentrics arranged on the crankshaft on either side of the unit's main journal. Figure 2.44 shows the arrangement of the latest Wärtsilä engine showing the cooling channels and the position of one of the three injectors.

The modern designs have removed the exhaust piston and associated running gear and have arranged for the exhaust gasses to be removed through a poppet valve situated at the centre of the cylinder cover. On the early designs the valve was operated by a traditional system which was a mechanical push rod operated from a cam on the camshaft. The valve was closed and kept in position with a mechanical spring.



The next step in the design was to replace the mechanical push rod with a hydraulic actuator to open the valve which was closed again by using the mechanical spring. Later versions on the current models show that mechanical spring has been replaced with a compressed air return spring (figure 2.45). 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.



The advantages claimed for the latest configuration of hydraulically operated valves are:

- There is no transverse thrust from hydraulic actuators. Thrust is purely axial resulting in less guide wear.
- Controlled landing speed, from the air return spring, ensures minimum stress on valve and seat.
- Valve rotation, caused by impellers fixed to the valve spindle, ensures well-balanced thermal and mechanical stress and uniform valve seating.

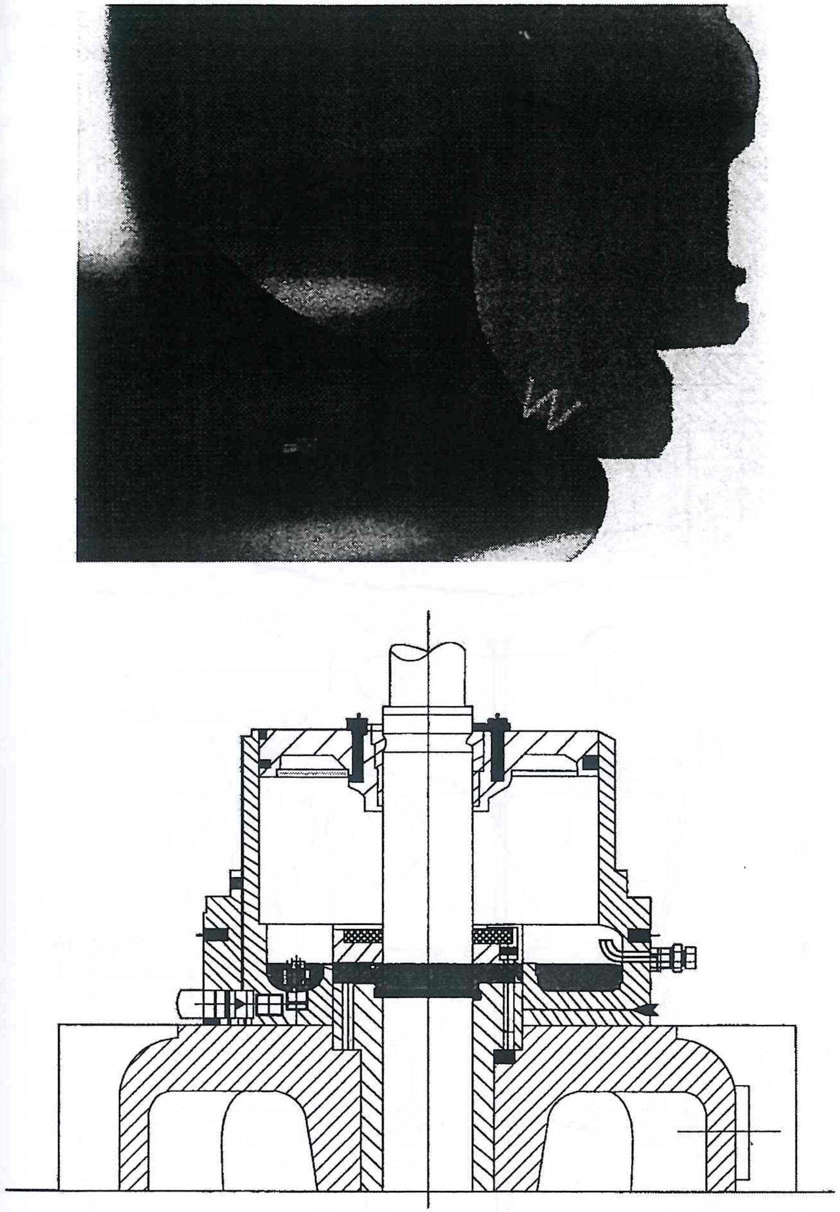
The extensive cooling of the valve cage and seating ring results in relatively low exhaust valve seat temperatures which coupled with the choice of the nicke–chromium–cobalt alloy (Nimonic) for the one-piece valve increases reliability and the intervals between overhauls even when operating on heavy fuel (figure 2.46)

Medium-speed four-stroke engines

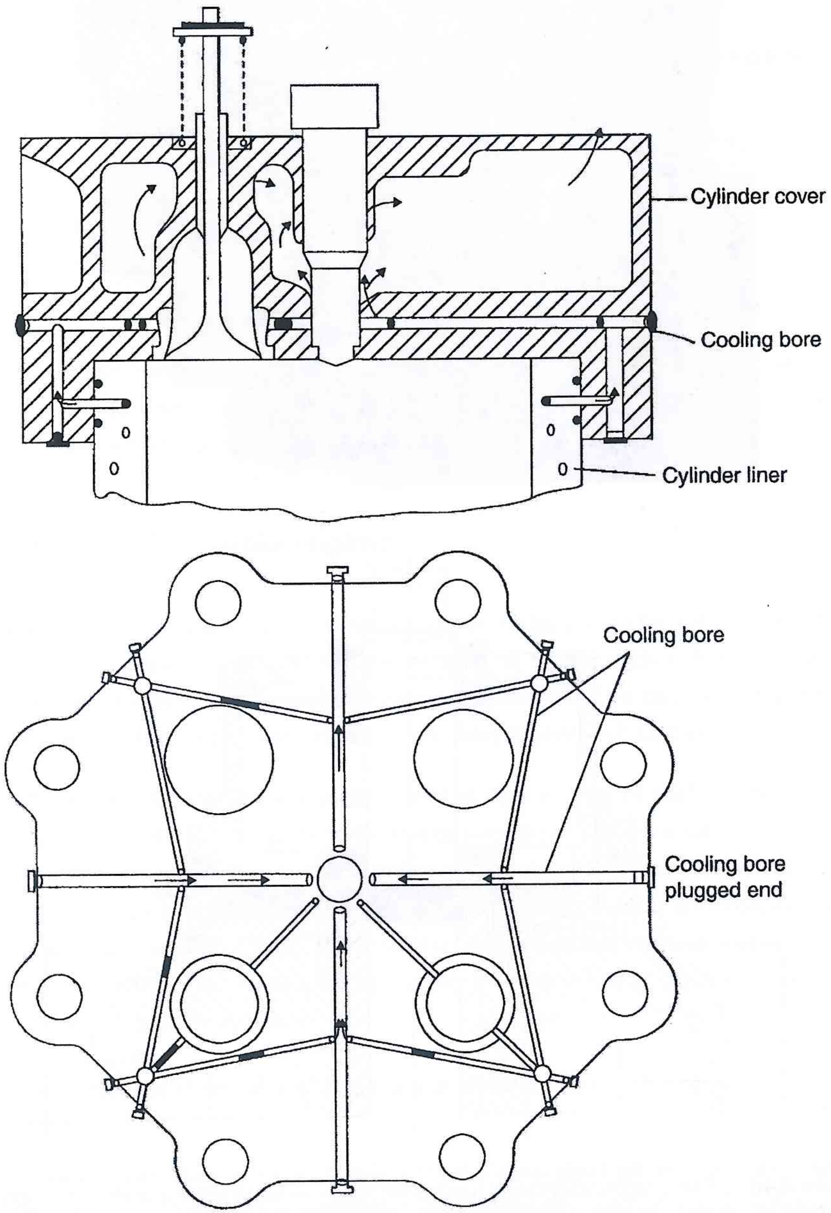
The breathing of an engine is a description given to its ability to get the air into and out of its combustion spaces so that the air can be mixed with sufficient quantities of fuel to give efficient combustion. Efficiency gains in engine design in recent years have been attributed to changes made to the inlet air and exhaust systems.

One of the advances is with the development of four valves for each cylinder. Having the four valves gives a larger opening for the gasses to pass when entering and leaving the combustion chamber. Due to the number of openings required for the valves, four-stroke cylinder heads are of a complex shape and for this reason spheroidal graphite cast iron is a suitable material for the manufacturer, since it is relatively easy to cast. A modern cylinder head for an engine operating on heavy fuel is shown in figure 2.47. Such a cylinder head should have:

- A small and even thermal and mechanical deformation with correspondingly low stress levels.
- Low and uniform temperature distribution at the exhaust valves and valve seat.
- Good valve seating due to effective valve rotation and low levels of distortion (important for optimum heat transfer from exhaust valve seats).
- Exhaust valves made from a material that provides resistance to high temperature

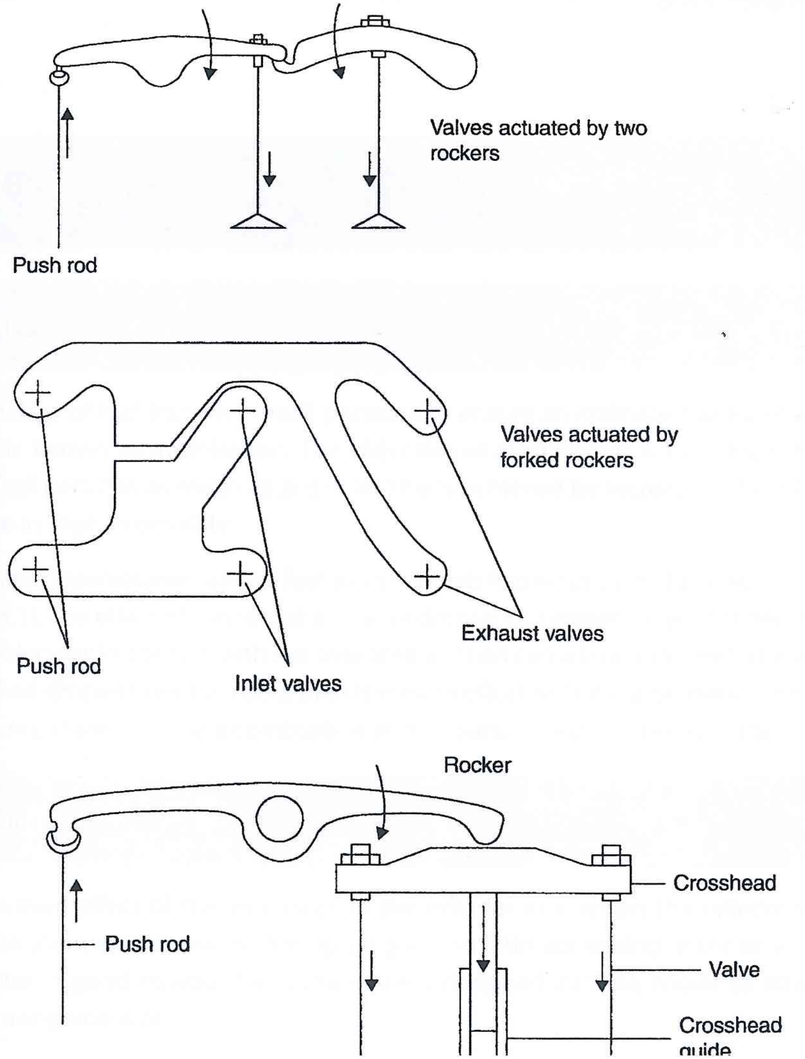


▲ **Figure 2.46** *'W' seat technology (MAN two-stroke engine exhaust valves) and exhaust valve air spring – MAN Diesel*



▲ **Figure 2.47** *Four-stroke diesel engine cylinder cover with bore cooling*

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 figure 2.47. Four valves are usually employed on four-stroke engines. This configuration allows the designer to maximise the Cross Sectional Area (CSA) of the inlet and exhaust ports and so improve the flow through the cylinder. This arrangement results in more complicated valve actuation since the two exhaust and two inlet valves must each be operated by one push rod. Examples of the various ways that the valves are actuated can be seen in figure 2.48. All of the designs shown control the valves together,



and 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.

3 FUEL INJECTION

Definitions and Principles

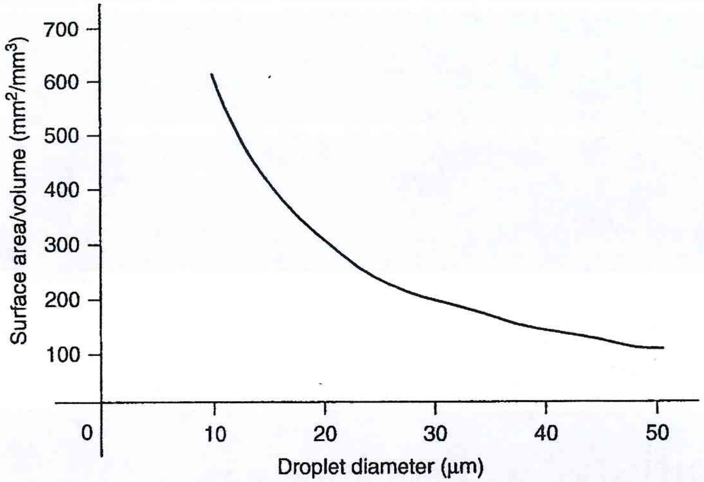
Atomisation

The breakup of fuel into very small particles to ensure an intimate mixing of air and fuel oil is known as atomisation. The objective of atomisation is to reduce the size of the fuel particles as much as possible. This is achieved by increasing the injection pressure as high as possible.

The surface area/volume ratio of fuel-oil in droplets increases as its diameter decreases (figure 3.1). The effect of this is that a smaller droplet can present a greater percentage of its molecules in contact with the available air than can a larger droplet. The smaller the fuel-oil droplets can be made, then the more effective is the atomisation, resulting in more rapid and complete combustion and maximum heat release from the fuel.

Turbulence

This is a swirl effect of the air charge in the cylinder as it enters the cylinder, which in combination with atomised fuel spray gives an intimate mixing of the air and fuel which aids in good combustion. Turbulence is 'designed into' the engine by attention to the arrangement of:



▲ Figure 3.1 *The relationship between fuel droplet size and surface area/volume ratio*

Penetration

This is the ability of the fuel spray droplets to spread across the cylinder combustion space, allowing maximum utilisation of the available volume for combustion.

Ignition delay

The process of atomisation is to achieve as small a fuel droplet size as possible. The reason for this is that at the micro level a drop of pure fuel is too dense to burn instantly. There is a time delay while the outer surface of the fuel droplet absorbs heat, evaporates and mixes with the oxygen to form a flammable mixture. The time interval from the start of injection until the start of ignition is called the ignition delay. Ignition delay will be affected by the:

- level of atomisation achieved
- grade of fuel being burnt
- quality of the fuel being burnt.

Impingement

Excess velocity of fuel spray will result in the fuel droplets making contact with metallic engine parts and resulting in flame burning. Engine manufacturers have found that impingement is responsible for a considerable amount of the smoke produced by an engine.

Diesel knock

The diesel engine is designed to operate on a continuous cycle with every component playing its part and operating at the exact moment in the cycle. The 'timing' of this process is critical with the burning of the fuel at the correct moment probably being the most important of all the processes. When the piston is just into its journey travelling down the cylinder, that is when it needs the energy boost from the pressure built up by the heat released from the combustion process. The piston will then be assisted in its motion. However if the piston is hit by a force, trying to push it down the cylinder, when it is still travelling up the cylinder, then the two will be in opposition and the result will be a 'knocking' sound called diesel knock.

Sprayer nozzle

This is an 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 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 nozzles may be specified for different applications.

For example, vessels with engines that still have mechanical fuel injection control, and are 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. The 'slow steaming' injectors give the engine improved economy at part load operation. The pressure drop ratio about 12:1 and fuel velocity through the hole about 250 m/s. This system is still not ideal for complete combustion control at part load operation. This is one of the primary reasons for the development of electronic control of the combustion process. See the section later in this chapter (p. 134).

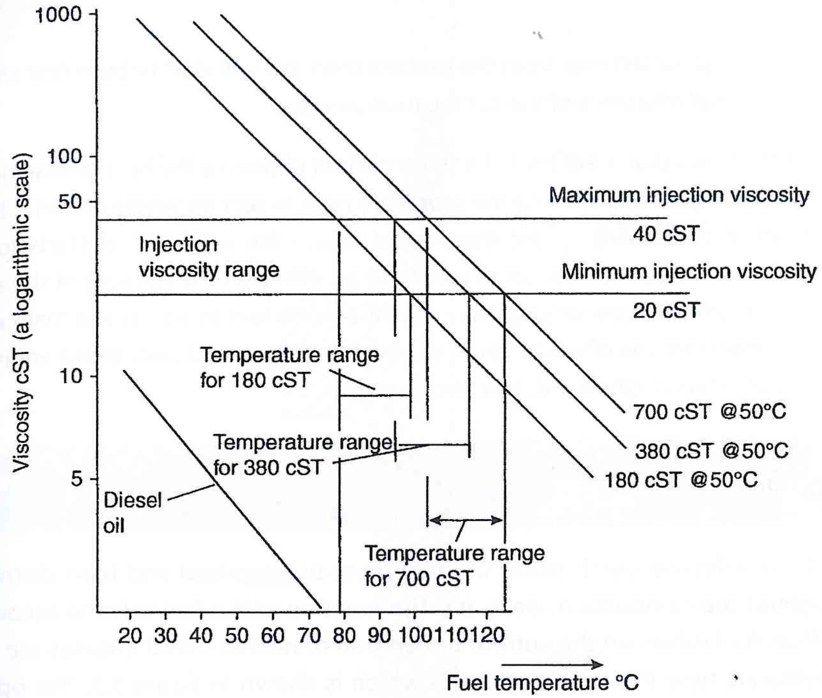
Viscosity

This may be defined as a fluid's resistance to flow due to the molecular friction that is present within its structure. The internal friction can be changed by heating the fluid where the higher temperature reduces internal friction and thus will also reduce the viscosity.

Pre-heating

The correct atomization of fuel will depend upon the viscosity at the point of injection. The demands upon modern engines to burn a variety of fuel types is becoming more important. The ability to burn residual fuel in marine diesel engines is still important and therefore to ensure optimum fuel injection it is important that the 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, resulting in ignition delay, excessive penetration and possible impingement on internal surfaces.

If the fuel temperature is too high then the viscosity will be low thus reducing penetration and causing deposits to be left on nozzle tip affecting atomisation. The relationship between viscosity and temperature is shown in figure 3.2. 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 engine manufacturers. It is modern practice to utilise viscosity controllers that ensure correct fuel viscosity by the careful control of fuel temperature. Despite the manufacturers of the viscosity controller specifying that it is fitted in close proximity to the engine, 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



▲ **Figure 3.2** *Viscosity temperature chart for marine fuels*

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 for 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.

Heat release of residual fuel

The quality of fuel is a problem associated with the use of residual fuels and recently with LSF, which will be explained in more detail later in this chapter. However the general rate of heat release (ROHR) for residual fuel is affected by the quality of the if there is any gasoil left over from the process then this will start to burn first and may affect the overall efficiency of the combustion process.

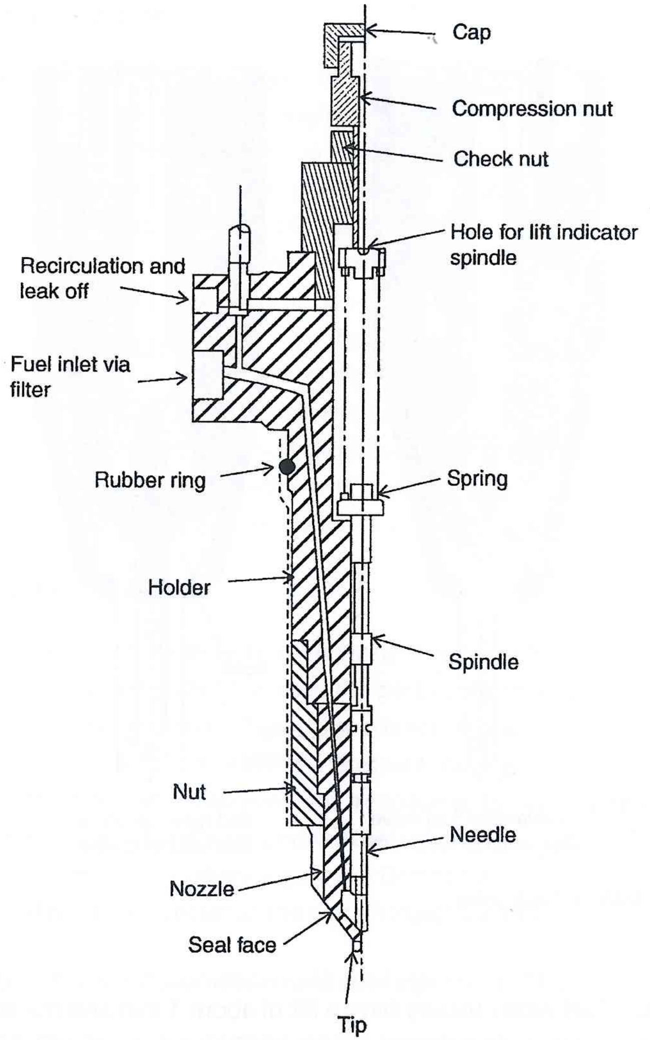
Figure 3.5b shows that there could be three distinct phases to the heat release process. The naphtha ingites first starting the process, a middle section where gas oil is ignited is next which is followed by the main event where the residual fuel starts to burn. The quality of the bunker fuel could well have an effect on the running of the engine especially as ships are becoming more reliant on one fuel to power the main engine and the generators one of which could be variable, slow-speed, two-stroke engine and the other constant, high-speed, four-stroke engine.

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 the current generation of marine diesel engines are still of the hydraulic type the cross-section of which is shown in figure 3.3. The opening and closing of this type of valve is controlled by the fuel pressure delivered by the fuel pump. The fuel pressure acts on the needle in the lower chamber and when the force is sufficient to overcome the force of the spring keeping the valve shut, the needle lifts.

Full lift occurs quickly as an extra area of the needle is exposed to the fuel pressure, after initial lift, placing additional force against the closing force of the spring. The full action of lift is limited by the needle shoulder which halts against a thrust face on the injector's body (see figure 3.4). The injector lift pressure varies with the different designs but may be about 140 bar on average with some designs reaching as much as 250 bar.

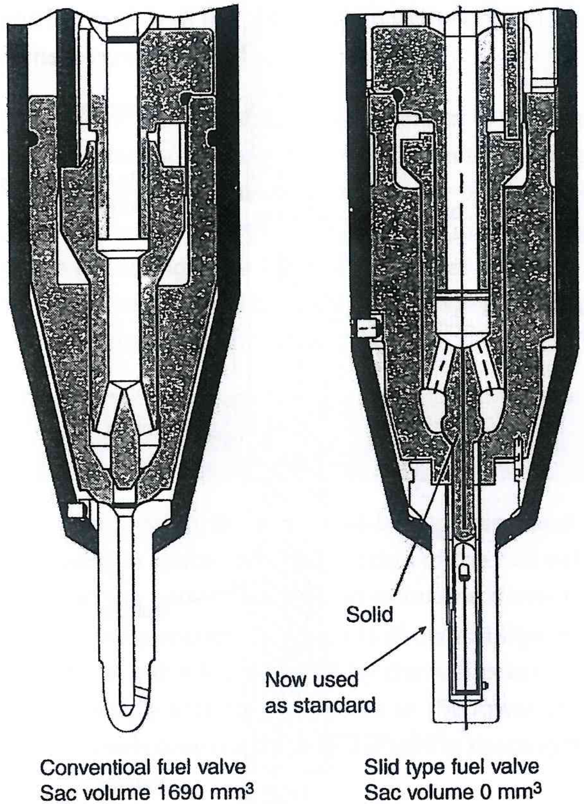
A fuel valve lift diagram for such an injector is given in figure 1.12 in Chapter 1. Figure 3.3 shows that by removal of the spring cap the valve lift indicator needle can be removed, reassembled or adjusted. The particular design as sketched is not cooled by itself but is enclosed in an injector holder in the cylinder head that will be kept at the necessary design temperature by the engine's cooling systems. The fuel valve will require a seal which in the case of the one shown in figure 3.3 is intended to be a face-to-face seal with the cylinder head's pocket. Some injectors are designed to have a copper ring between the fuel valve nut on the lower end and the bottom of the cylinder head fuel



▲ **Figure 3.3** *Fuel valve injector (hydraulic)*

to the rest of the combustion space which will have a dramatic effect on the engine's performance due to poor combustion.

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



▲ **Figure 3.4** *MAN fuel slide valve*

200°C. Hydraulic fuel valves usually have a lift of about 1 mm and the action is almost instantaneous.

Fuel injection systems

- Mechanical fuel injection
- Mechanical common rail
- Electronically controlled (common rail) fuel injection (EFI).

Mechanical fuel injection

This is the traditional system employed in modern marine diesel engines. It is the most commonly used one in existing engines but it is very quickly being replaced by the newer electronic fuel injection systems. Fuel is supplied to the high-pressure fuel pump from a low-pressure fuel delivery pump and associated pipework. The fuel flows into the high-pressure pump where it fills the internal spaces in the delivery chamber. As the pump is operated by a can on the camshaft the delivery chamber becomes closed off and the fuel pressure starts to rise dramatically following a few degrees of rotation of the cam operating the plunger. Fuel is delivered directly to spring-loaded injectors via the pump's delivery valve and high-pressure pipework. The fuel injectors are opened up by the hydraulic action of the fuel after the high-pressure fuel pump plunger movement has generated sufficient fuel pressure to overcome the spring pressure keeping the fuel valve closed. The action is designed to be rapid as an aid to the timing and atomisation of the fuel ready for combustion.

Mechanical common rail

Although this is now an obsolete system it is worth a mention here to show the student that good systems that fell out of favour in the past could make a return to efficiency especially with the advancement of material science and electronics. This very early system had fuel pumps to deliver fuel to a pressure main and various cylinder valves were opened to the main which allowed fuel injection to the appropriate cylinder. The system required either mechanically operated fuel valves (e.g. older Doxford engines) or mechanically operated timing valves (e.g. newer Doxford engines) allowing connection between rail and hydraulic injector at the correct injection timing.

Electronically controlled (common rail) fuel injection (EFI)

Modern engine designers are coming under increasingly more pressure to build engines that are considerably more fuel efficient than they were just a few years ago. The most important strategy to achieve this is to use sophisticated and very close combustion control which cannot be achieved by the use of mechanical systems such as described in the first section here – mechanical fuel injection systems.

The fuel is supplied – via a low-pressure delivery pump and associated pipework – to a high-pressure pump and piping system running the entire length of the engine. The electrically operated fuel valves (injectors) are connected to the high-pressure 'rail' pipework. When combustion within a given cylinder is required an electrical signal

The important aspect about the EFI system is that the start, duration, quantity and end of fuel delivery are all completely controllable to a very fine degree and they are parameters that can be modified independently of each other and independent of any other consideration such as engine speed or ambient temperature. The more complex systems can also provide a pre-injection and a post-injection sequence which give an added advantage as described over the next few pages.

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.

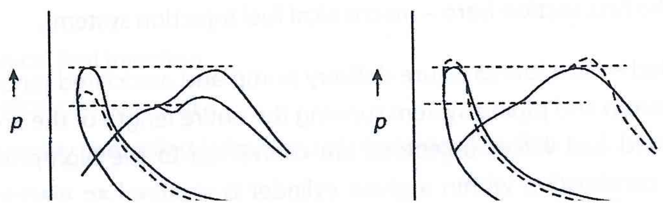
Indicator diagrams

The development of EFI and the associated equipment means that some of the same control mechanisms can be used to give detailed feedback about the equipment's performance, to the engineers. This improved level of information will enable the knowledgeable engineers to carry out performance analysis and fault finding at a level not achievable in the past. The heart of this diagnostic process will be the combustion indicator diagram.

Details have been given of some typical indicator diagrams showing engine faults in Chapter 1 and some fuel injection faults have been outlined including late and early injection, shown on the draw card, fuel valve lift diagrams, as well as related details such as compression cards. Two further typical faults are as illustrated in figure 3.5.

Afterburning is generally associated with poor quality fuels and will be characterised as shown in figure 3.5b by:

- an increased peak pressure
- loss of power



- increased cylinder exhaust temperature
- possible discolouration of exhaust gases.

The effects of fuel restriction on the combustion process can be seen in figure 3.5a. This may be due to:

- blocked fuel filters
- injectors
- incorrect viscosity
- poor fuel quality.

These will all result in a loss of power and reduced maximum pressure.

Fuel Pumps

General

The physical energy demand of fuel injection is substantial. Typical requirements include delivery of about 100 ml of fuel in 1/30 s at 750 bar so as to atomise an area of 40 m. A peak energy input can reach 230 kW. A short injection period at high pressure, arranged to give the desired firing pressure, at the right time and in the right direction is very important. Pilot injection and phased injection of charge is now not a problem with modern engines that have electronic combustion control systems.

Quantity control

Traditionally, diesel engine fuel delivery, injection and atomisation has been carried out by using totally mechanical systems with the majority operating by varying the amount of fuel injected per stroke which is controlled by varying the effective plunger stroke of the high-pressure fuel pump (see figure 3.10). Mechanical systems achieve this by:

1. Varying the beginning of delivery

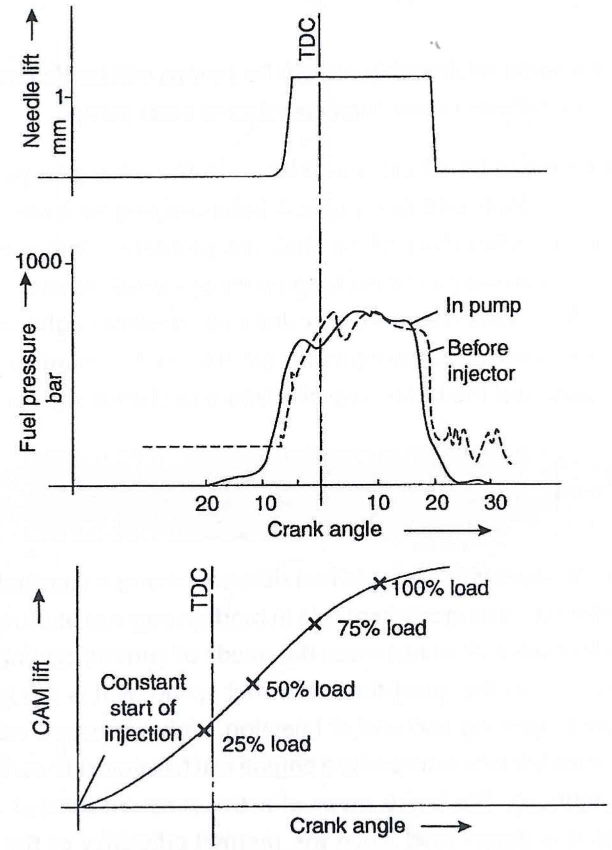
Currently the most popular method of mechanical fuel injection on larger marine engines is by using a single high-pressure fuel pump for each cylinder. Regulation of the quantity of fuel is matched to the load of the engine by a governor operating various linkages that control the output from the pump.

The pump is a single piston moving inside a barrel or cylinder. As the piston is at the bottom of the stroke the space above fills with fuel. As the plunger rises the inlet port is closed and the fuel is delivered from the delivery valve at the top of the pump. The control method is to change the effective length of the pump's delivery stroke by altering the end of delivery. This is still an important method of diesel engine control and students will need to be familiar with the constant stroke and helical groove system that has a constant beginning of injection which is described later. This method is more suited to constant speed engines which would require a fixed start of injection and the amount of fuel required would increase as the load increased.

They are regularly fitted to auxiliary engines and give fuel injection early in the cycle at light load which not only gives higher efficiency but also leads to higher firing pressures. They have also been used with large direct drive engines such as the older Burmeister & Wain (now MAN Diesel & Turbo). However, the large two-stroke engine is a variable speed engine and therefore requires a variable time for the start of injection.

Therefore, the valve-type pump was favoured for large engines which had a constant end of delivery and regulation of the start of injection accomplished by varying the suction valve closure. Part load performance of these engines with later injection is always 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 part loads due to reduced turbo-charger delivery and pressure.

Owing to the limitations of varying the fuel quantity delivered by only varying the beginning of delivery Sulzer (now owned by the Wärtsilä corporation) redesigned their fuel pumps to include a suction valve and spill valve. Initially the spill valve was the only controlled valve resulting in constant beginning with variable end of delivery. Latter, however, in the interests of fuel economy and in common with other manufacturers, both valves were controlled to give a controllable beginning and end of delivery.



▲ **Figure 3.6** *Injection characteristics*

injector lift diagram with a total lift of about 1.3 mm and injection period at full load approximately 6° before to 22° after TDC. High firing pressures achieved at full load in the high powered turbo-charged range of engines can be reduced when using the constant beginning of injection method.

The ideal injection profile (for a fuel with a constant ignition characteristic) corresponds to a rectangle with almost constant fuel pressure before and during injection. The practical curves illustrated show almost constant pressure at a reasonable maximum (750 bar). However, fuel injection pumps can be a problem especially when running on residual fuel oil and also when running at the same load over a prolonged period. Impurities in the fuel cause wear on the barrels and plunger of individual fuel pumps. This may happen to pumps in different amounts. Therefore, one pump might wear are operated in the same relationship not all the pumps will be delivering fuel in the same way due to the different wear characteristics of each pump.

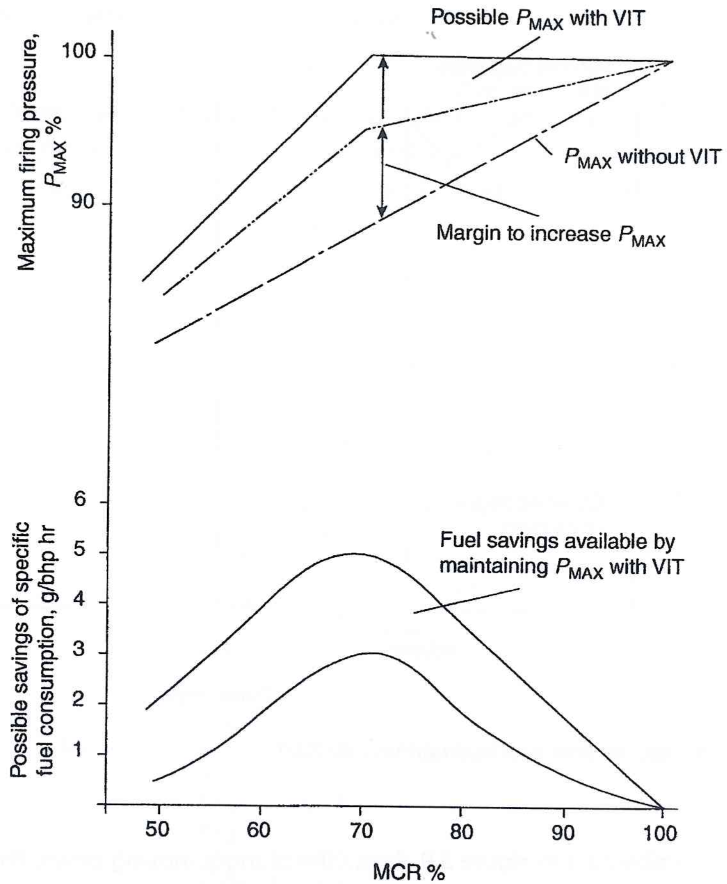
Individual fuel pumps can be adjusted in relation to the other pumps on the engine however this must be done with great care. A balanced engine is one where each of the cylinders is taking its fair share of the load. The problem is that one of the pumps could be pumping low at one part of the range but just a small movement further it will be pumping the full amount. Therefore, any slight adjustment might cause a pump to move from under pumping to pumping more than it should. In many cases only a small adjustment is possible and the best cause of action is to change the pump.

Variable injection timing

The previous section dealt with variable fuel delivery having a constant beginning of injection. This system is no longer acceptable in modern engines because the part load and low load performance does not meet the needs of current legislation relating to engine emissions. Also in the quest for better fuel economy it is modern practice to now vary both the beginning and end of injection. With a constant start of injection system the maximum firing pressure of the engine will fall almost linearly as the power of the engine is reduced. The brake mean effective pressure (bmep) of the engine, however, reduces at a slower rate. 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, and the start of combustion, as the engine load is reduced (see figure 3.7). The advancement of the injection timing continues until about 65–70%, thereafter the injection is retarded (figure 3.8).

The ability to retard the fuel injection is extremely important, especially when used to control the emissions from the engine. Retarding the fuel injection delay the heat release controls the highest peak temperatures in the combustion process. The ability to advance and retard the fuel injection process will reduce the 'diesel knock' at low engine loads that is sometimes experienced in engines without VIT.

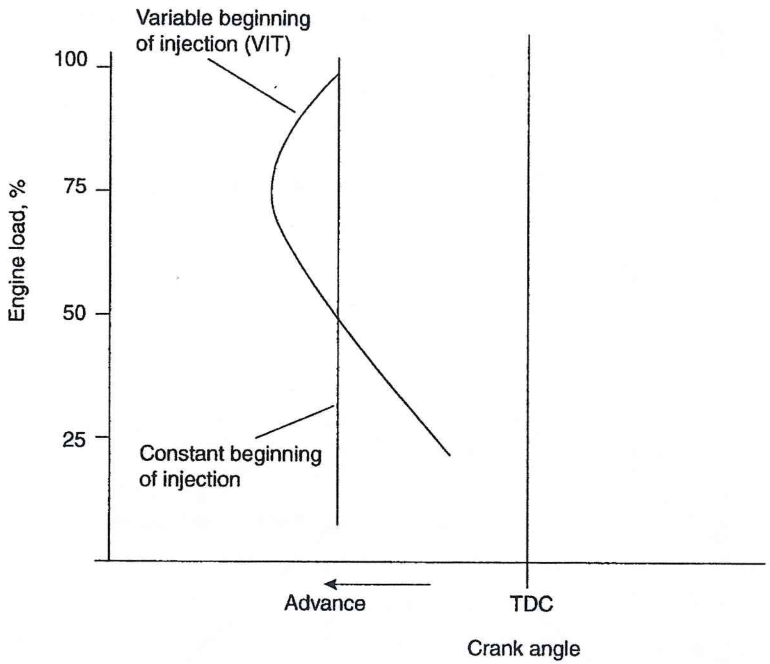


▲ **Figure 3.7** *Fuel savings available by utilising variable injection timing (VIT)*

As we shall see later variable valve control will allow the designers to further change the combustion process to the requirements of local regulation, fuel type and operational conditions.

Scroll-type high-pressure fuel pump

Despite the move towards common rail fuel injection the scroll-type high-pressure fuel pump is still very important to the industry. Therefore, students should study the principle of operation described here as the examiners will be very interested to find out all you know about the operation of this type of pump. Figure 3.9 shows a scroll-



▲ **Figure 3.8** *Injection timing variation with engine load*

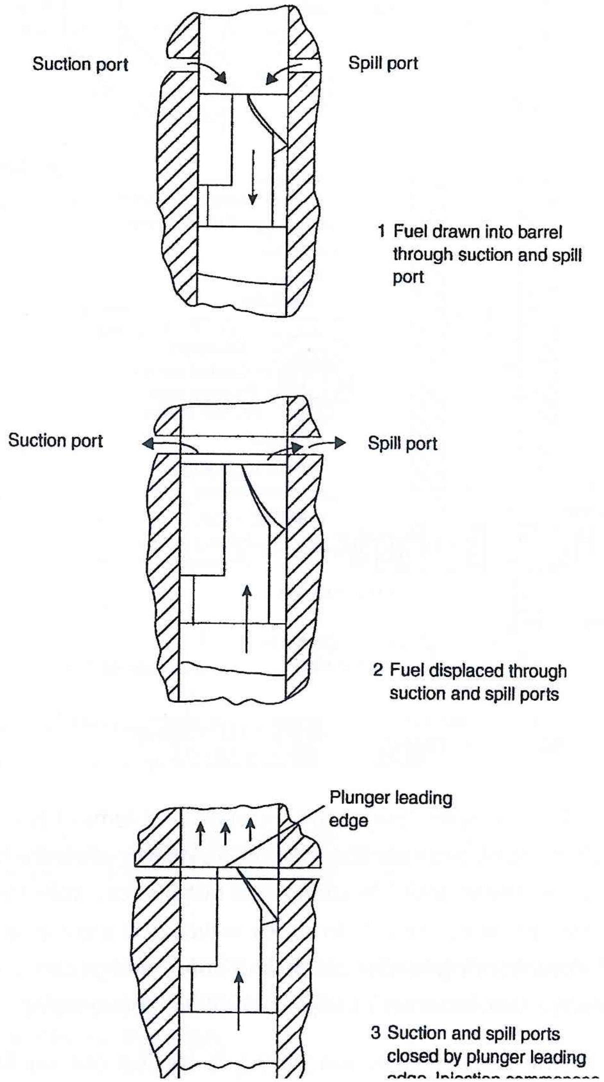
The sketch numbered 1 in Figure 3.9 shows the plunger moving down. The pressure in the barrel falls and as the suction and spill ports open to the fuel rail the fuel flows into the barrel. In sketch 2 the plunger is moving upwards. The fuel is displaced from the barrel through the spill and suction ports. This displacement will continue until the plunger completely covers both ports.

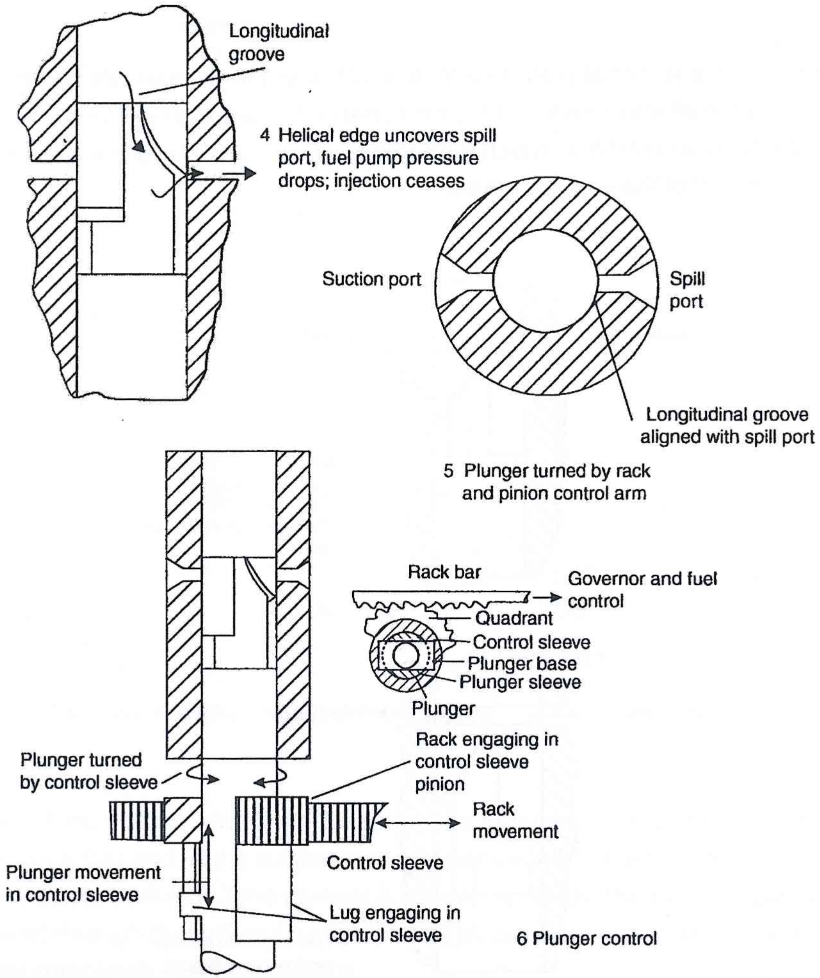
Sketch 3 shows the plunger continuing to move upwards and just covering the spill and suction ports. This is the effective beginning of delivery and any further upward movement of the plunger will pressurise the fuel that has already filled the high-pressure pipe between the pump and the fuel valve, and it will open the fuel valve injecting fuel to the engine.

In sketch 4 the plunger continues to move upwards. Injection continues until the point when the lower 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

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 in order to vary the effective height of the helix relative to the spill port. The plunger base is slotted into a control sleeve which is rotated by quadrant and rack bar that are both under the control of the engine's governor.

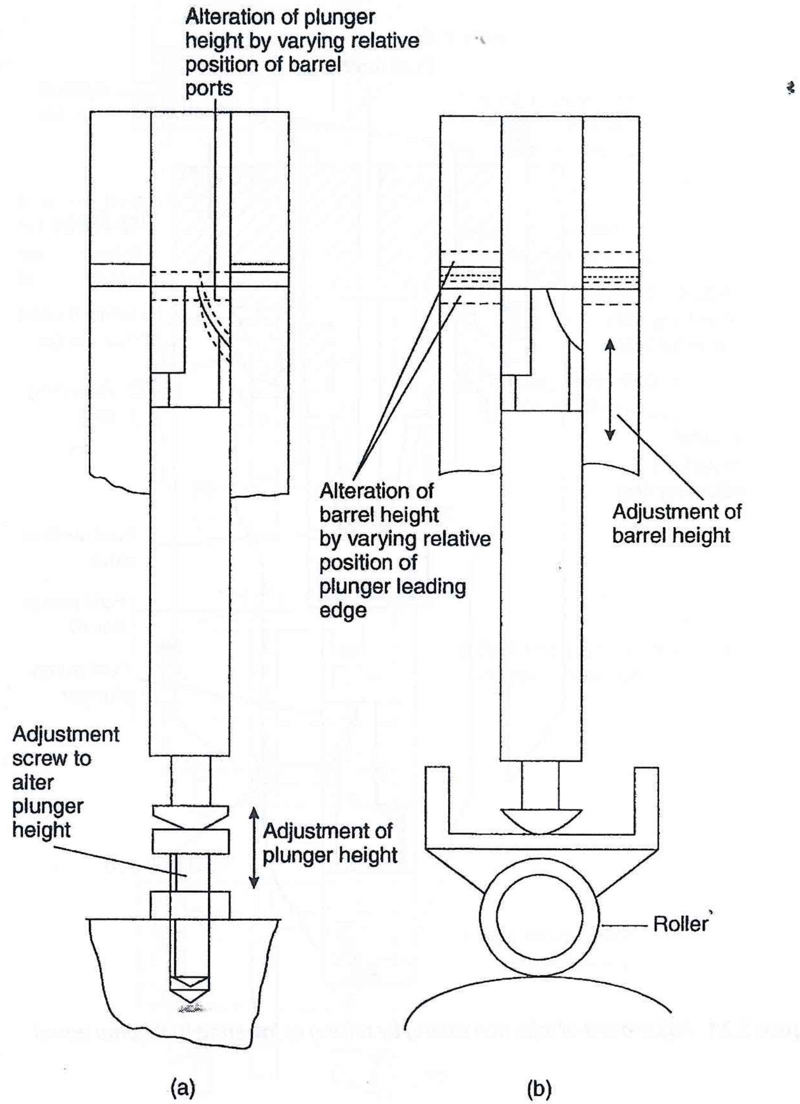




▲ **Figure 3.9** *continued*

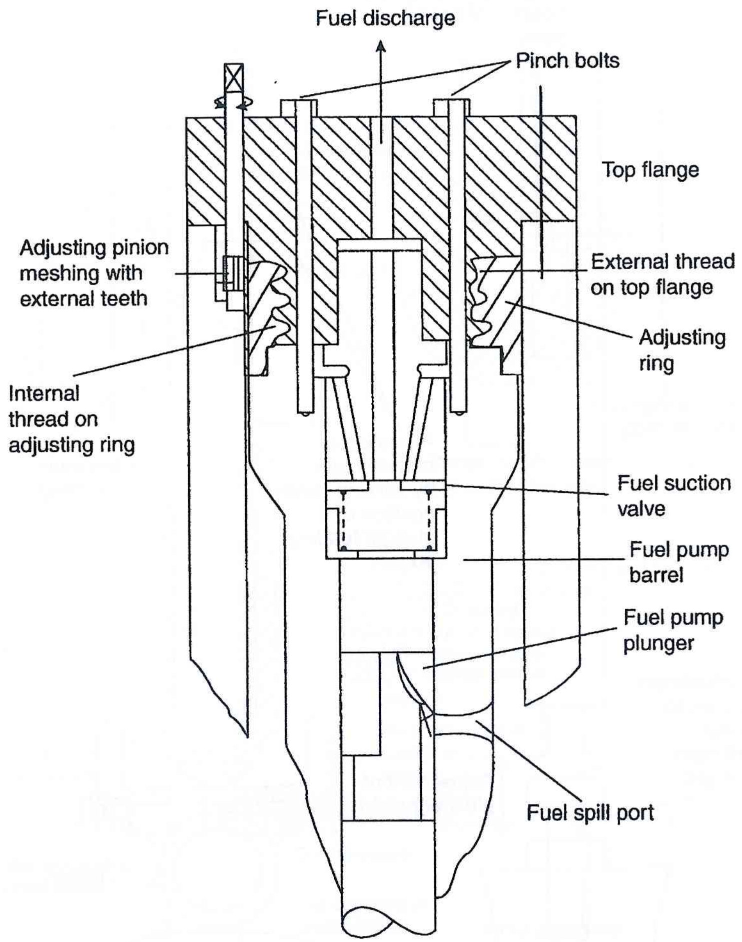
With both valve and helical spill designs there can be problems of fuel cavitation due to very high velocities. Velocities near 200 m/s can create low-pressure vapour bubbles if pressure drops below the vapour pressure. These bubbles can subsequently collapse during pressure changes which results in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

The adjustment of the start of injection timing is carried out on Bosch-type fuel



▲ **Figure 3.10** *Adjustment of fuel injection timing by varying relative position of plunger and barrel*

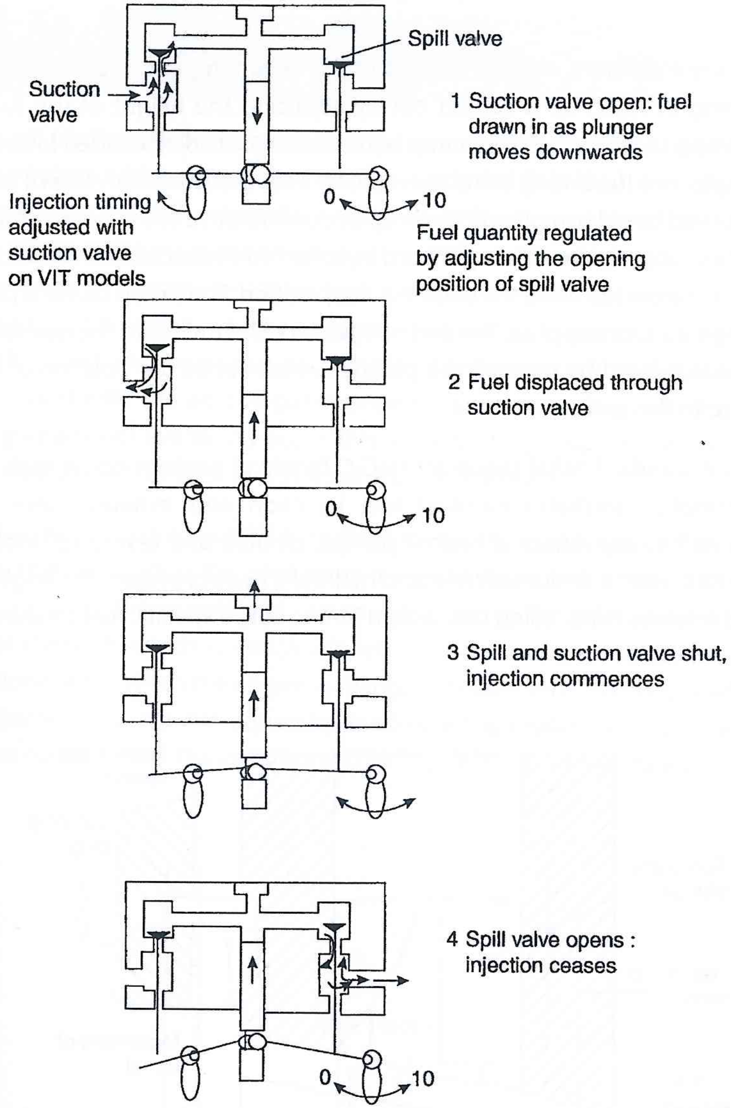
Adjustment of plunger height can be accomplished in some installations by adjusting the cam follower. Lowering the plunger has the effect of retarding the injection. Raising the plunger advances the injection.



▲ **Figure 3.11** *Adjustment of injection timing by raising or lowering fuel pump barrel*

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 is 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.

Fuel pump (valve-type fuel pump) detail

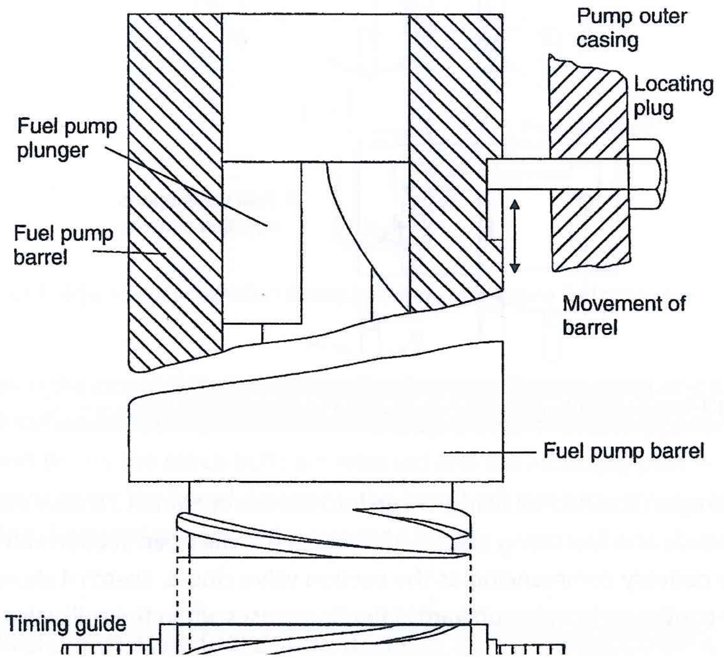


▲ **Figure 3.12** *Wärtsilä-Sulzer valve-type fuel pump*

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 shows that as the plunger continues to move upwards injection ceases when the spill valve opens.

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 figure 3.13 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. The moment the plunger covers the spill port, injection commences. The duration of this process can be adjusted while the engine is in operation. The pump barrel is prevented from turning by a locating 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.

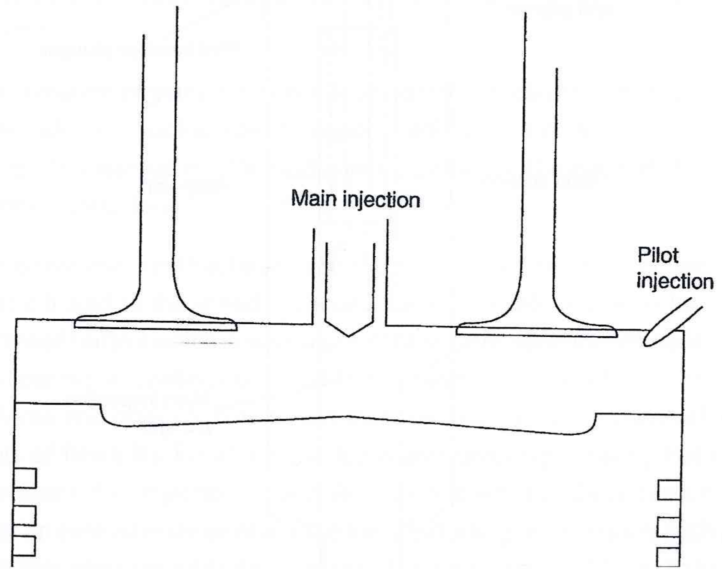
Currently the standard MAN Diesel MC/MC-C family of engines come with a chain-driven camshaft, camshaft-controlled fuel injection and exhaust valve opening systems as well as conventional fuel oil pumps, all tried and tested technology. The engine is fitted with a pneumatic/electric/hydraulic control system for engine speed control and manoeuvring. Using this system MAN quote a specific fuel consumption of 167 g/kWh ± 5%.



Two-stage fuel injection

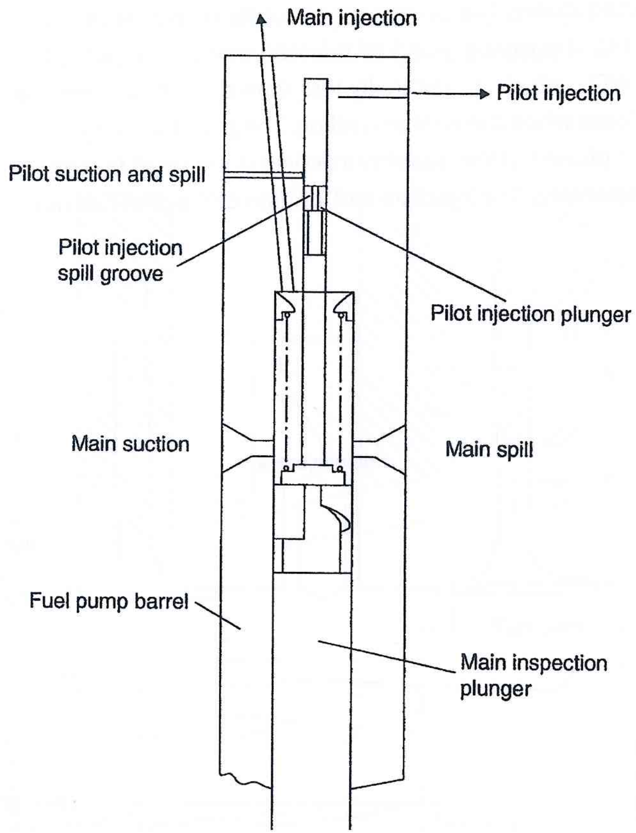
Efforts are continually being made to improve the reliability and economy of medium-speed engines which operate on heavier grades of fuel. One way of achieving this is by having a carefully controlled, reliable combustion process. This requires good atomisation with short injection periods but this results 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. As 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 developed a two-stage injection process in which the fuel injected during the pilot stage is constant and independent of engine load; see figure 3.14). The quantity of fuel injected during the pilot phase is set at about 2–6% of the MCR, which 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. 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 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 (figure 3.15).



This arrangement has the advantage that the emission of NOx are reduced and allows the use of low Cetane fuels.

EFI (some with common rail)

Now that the student has studied the basics of fuel injection/combustion and seen the limitations and constraints that the older mechanical systems imposed upon engine designers, let's now have a look at the next steps in diesel engine design.

Recent advances in material science, linked to the continual quest to reduce costs, have led to the development of reliable and accurate measuring and sensing technology. It is now possible to measure the fuel rail pressure and the combustion pressure in real time and feed this information back to a central processing unit that will be able to continually adjust the engine's settings to give the best combustion conditions possible at all times.

Riding on the crest of this wave of development is the four-stroke medium- and high-speed diesel engine. Not only does it have an infinitely variable, common rail, fuel injection system but it also has variable opening and closing of the inlet and exhaust valves giving a very sophisticated and powerful system. However, such a system does not come easy and there has been considerable investment in development time and resources been spent by the engine manufacturers to design reliable systems.

The changes in marine engines have been lagging behind advancement in the engines used by other industries such as road transport. Legislation is now driving the need for change and lessons learned in other industries means that the pace of change in the marine engines is considerable.

The problem is that engines that have a camshaft-controlled combustion process have a system that is linked to the speed of the engine and therefore there is little room to design a system of control which is load dependent and not speed dependent. Common rail systems permit a continuous, load-independent control of injection timing, pressure, volume and phasing. This means that common rail technology achieves the highest levels of flexibility for all engine loads and gives significantly better results than a conventional fuel injection system. Reliable and efficient CR systems have been developed for an extensive range of marine fuels, including residual fuels such as heavy fuel oil (HFO). This gives the added advantage of using a single fuel for both two-stroke

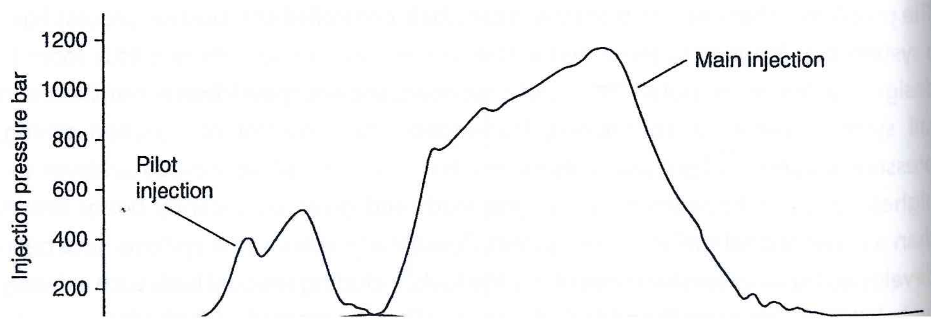
Basic system design

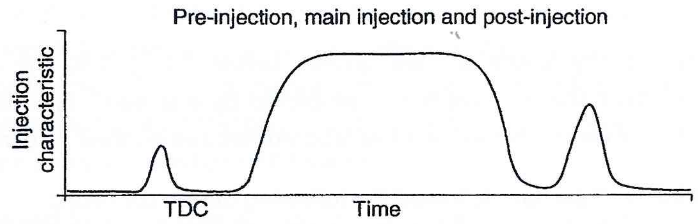
The basic idea of the common rail system is quite simple. High-pressure fuel – as much as 2,000 bar – is circulated around the engine close to the fuel injectors. There will be a short length of pipe from the 'common rail' to the fuel injector. When the fuel injector is opened the high-pressure fuel flows into the combustion space through the small holes in the injector.

The fuel will atomise well due to the very high pressure of the fuel in the fuel rail and due to the very small angled holes in the injector outlet. The major advance with this system comes from the ability to open and close the fuel injector so quickly. This enables very close control over the timing, duration and sequencing of the fuel injection process.

Therefore, so much more can be done to influence the combustion process. A two-stage phased injection is shown in figure 3.16 and has the effect of lowering the peak temperature of the main injection. The peak temperature is partly responsible for the production of NOx which are harmful to the environment. Therefore, ability to modify the injection has an immediate pay back in an engine with a cleaner exhaust.

Figure 3.17 shows a further development where a third phase is introduced. This follows the main injection phase, assists the mixing of fuel and air and raises the temperature towards the end of combustion. The temperature increase promotes soot oxidation and reduces the amount of particulate matter being emitted with the flue gasses.





▲ **Figure 3.17** *Phased injection timing*

Safety considerations

The flag state examiner will need to be sure that any engineer gaining a certificate of competency from his/her administration has a comprehensive understanding of the safety required with engines operating the common rail fuel injection system.

The first and very obvious statement to make is that we have high-pressure fuel inside pipework that stretches over longer distances than has been the case in the past. Due to the high pressure if any fuel does escape then it also has the potential to spray over a larger area. Given the fact that over the years fuel spraying onto hot surfaces has started a good number of serious engine room fires. Then the examiner will quite rightly concentrate his/her questioning of the candidate to ensure an understanding of this point.

Yes all the high-pressure piping will be shielded in a double casing and yes care will be taken at the design stage to ensure that the fuel lines run in as safe a place as possible. However, it will be the candidate's responsibility to point out that this is a potential source of danger and that particular attention should be paid to the integrity of the pipework and associated fixings. Good maintenance practices must also be followed. It will be vitally important that the correct 'high-pressure' fittings, couplings and seals are used – unapproved alternatives are not acceptable. The pipework must be supported correctly after any maintenance work and it is important that any supports, which were removed to ease the dismantling of the pipework, are replaced.

Sub-standard workmanship could lead to failure in the pipework or could allow leaks of fuel out of the system or air into the system. Either of these faults will affect the operation of the engine and if the engine stopped it could endanger the vessel as well.

consequences of any failure of their system, before fitting it to the ME engine. Verification of their studies have been completed by a research and development (R&D) programme carried out in their test laboratories and on their research engines.

MAN Diesel for example have followed the following design concepts:

- The fuel injectors are only pressurized during injection.
  - Meaning that there is no danger of uncontrolled injection, even if a control valve or injection valve leaks.
- All high-pressure components are double-walled.
  - Any fuel from leaking or broken pipes is contained within the double-walled pipework. The leaking fuel can then be led to an accumulator that is fitted with an alarm.
- Flow-limiting valves are fitted to the fuel pipe for each cylinder.
  - The valves will limit the quantity of fuel to be injected, even in case of leaking or broken components.
- Non-return valves fitted in the line for each cylinder.
  - These valves prevent backflow from the low-pressure system into the cylinder, for example, if there is a fuel valve seizure or breakage.
- There are between two and four high-pressure pumps fitted to the systems.
  - Therefore, there is redundancy in the system and should one pump fail, emergency operation is possible.
- Pressure-limiting valves are fitted. These have the additional pressure control function and act as a safety valve.
  - This design characteristic means that emergency operation is possible, even in case of any failure in fuel rail pressure.
- Emergency stop valve/flushing valve included in the design.
  - The valve, which is actuated by compressed air, stops the engine in case of emergency.
- Surplus rail-pressure sensors and TDC speed pick-ups are fitted.
  - No interruption of engine operation will occur due to pick-up or sensor error.

All this hard work has gone into the carful design of the new system. However, it must be understood by the engineering staff on-board ship that it is now their responsibility

The Wärtsilä approach with their RT Flex engine is to control the fuel injection process with a 'volumetric fuel injection control unit' which controls the timing and volume of the fuel to be injected. Wärtsilä do have a common rail which is fed with heated fuel oil at high pressure (nominally 1,000 bar) ready for injection. Fuel supply is via a number of high-pressure pumps driven by multi-lobe cams.

Fuel is delivered from the common rail through the injection control unit, which is placed next to each cylinder, to the standard fuel injection valves. The fuel injectors are hydraulically operated by the high-pressure fuel oil. The control unit uses quick-acting valves to regulate the timing of fuel injection, control the volume of fuel injected and set the shape and phase of the injection. The three fuel injection valves in each cylinder cover are separately controlled so that, although they normally act in unison, they can also be programmed to operate separately as necessary.

Fuel Systems

In recent years the quality of the fuel available to the marine industry has deteriorated. This had not only led to problems with the combustion, but also to problems with the storage of the fuel. To minimise the effects from some of these problems careful consideration should be given to the design of the overall fuel system and this should include the bunkering system.

Modern residual fuels tend to have a high viscosity and may also have a high pour point so it is important that upon the completion of the bunkering process, it is checked that that the fuel has drained freely into the bunker tanks.

If the vessel is loading bunkers in cold climates it may be necessary to include insulation on the exposed bunker lines. An indication of a fuel with a high pour point may be a high loading temperature. Here the supplier is trying to ensure that the fuel is easily pumped on-board. If a waxy fuel is suspected then a pour point test should be carried out.

Due to the problems associated with incompatibility, fuels from different sources should not be mixed. The importance of segregation of fuels from different sources cannot be overstated and should be practiced, wherever possible, by transferring remaining fuel into smaller tanks prior to bunkering in order that the total quantity of

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 pumping in the next, possibly incompatible, fuel.

With the introduction of the new emission regulations and the requirements for vessel to operate on 'LSF', fuel suppliers are using more blending techniques to comply with the regulations and for this reason the risk of incompatibility will only increase. Fuels supplied to a ship must be treated before use. In fact comment has been made that this is one of very few products which is purchased but is not 'fit for purpose' and must have additional treatment by the purchaser before it can be used.

Technically bunker fuel is any grade of fuel that is used by the ship but the term main bunker fuel has come to mean the fuel used for powering the main engine. The term comes from the days when the ships were powered by coal and this was loaded into a 'bunker'. Main engine fuel must be supplied to a specification which is set out in the ISO standard 8217. The two latest revisions from 2005 and 2010 are the most important.

The standard sets out the specification for the fuel characteristics including viscosity, density, flash point, pour point, sulphur, carbon residue, water and ash. The new (ISO8217:2010) specification addresses some of the residual fuel quality problems that has been experienced by the industry, with the inclusion of acid number limits as well as a limit on hydrogen sulphide. The distillate grades have had the inclusion of oxidation stability and a lubricity requirement introduced and the residual marine fuels have a calculated carbon aromaticity index added as an indicator of ignition delay. There is also a limit on sodium content as well as stricter limits for ash and vanadium and there has been a significant reduction in limits for aluminium and silicon, which are also known as cat fines (see Chapter 2, Volume 8 of the Reeds series).

However, the new specification fuel does come at a price and DNV Petroleum Services (DNVPS) reported in 2011 that there was a resistance to using the new ISO8217:2010 fuel specification. Some of this was due to charter party agreements but another reason was problems with availability of products meeting the new specification but according to the DNVPS survey, only 10% of the total respondents said they would not eventually switch to ISO8217:2010.

The knock on effect from taking on 'off spec' fuel was that filters started clogging due

involved. It is the job of the engineering officers to ensure that they have evidence to support any claim on insurance, or against a third party, that the ship owner might wish to make.

There are some basic precautions that the ship's staff must take during the bunkering stage. These include:

- *Communication*
  - The engineering officers must make sure that the rest of the ship company know at which port bunkering is likely to be taking place.
  - Engineering officers must work with other relevant officers and crew during the actual operation so that they can all keep a watch on the process while carrying out their own tasks such as loading cargo or taking on stores and spare gear.
  - Communication between the ship and the bunkering vehicle (barge or road tanker) is vital to completing a safe operation.
  - Efficient lines of internal ship communication are also extremely important.
- *Resources*
  - The bunker station – which is the point where the vessel's fixed pipework is connected to the flexible bunkering hose usually provided by the bunker suppliers – should be 'manned' at all times during the operation.
  - The engineroom valve operating station of valve chest should be manned at all times and the officer stationed here should be in constant communication with the officer at the bunker station.
  - Adequate 'drip trays' should be placed under the final flange where the ship's pipework meets the flexible bunkering pipe.
  - Appropriate 'oil spill' dispersant and absorbent material should be placed close to the bunkering station.
  - Any water freeing holes in the ship's bulwark around the site of the bunker station should be temporally blocked so that if there was a spill the oil would be retained on-board where it can be cleaned up without incurring a financial penalty.
- *Other actions*
  - The chief engineer should check and agree the order quantity and quality with the manager in charge of the bunker barge or tanker.
  - Samples should be taken ideally at the start and at the end of the operation but

- The quantity being delivered needs to be checked – the traditional way has been to check the bunker fuel in the ship's tanks before and after delivery of the fuel. Alternatively, an engineer could go to the bunker barge or road tanker and check the quantities there before and after delivery (see below for the more modern approach).
- Chief engineer needs to record where the bunkers from that load are stored and that the records are understood by all the ship's senior management team.
- Chief engineer needs to update his/her standing orders so that the engineers know the sequence for drawing the fuel during the next voyage.

MARPOL Annex VI gives minimum values for the emissions from the flue of ships that come under the juristriction of countries that are signatories to IMO. Regulation 18 states that fuel of the correct standard should be available. However, it also recognises that there will be bunker ports in countries that are not party to the MARPOL agreement. When purchasing fuel from such ports IMO recommend that ship managers have a clause inserted in their agreement detailing the specification of the fuel, ideally to meet the IMO requirements. A Bunker Receipt Note with specific contents must be issued for each delivery together with a sample that is fully representative of the fuel delivered. These must be retained, not necessarily on-board, for 3 years in the case of the documentation and at least 12 months in the case of the fuel sample, in case they are required as proof of compliance. Furthermore, the regulation gives steps that must be taken in the event of non-compliance.

Flag states issue guidance for ship operators about the requirements for meeting the MARPOL Annex VI regulations. Classification societies also issue assistance to their members about the necessary steps needed to comply with good practice.

Fuel management – on-board systems

Ideally if ships were designed with two service tanks and two settling tanks then fuel change-overs could be accomplished with the minimum amount of mixing. This complexity of design would however have to be considered at the design stage and shipowners may wish to consider this when drawing up a new build specification.

The temperature of the stored fuel must be monitored to ensure that it does not

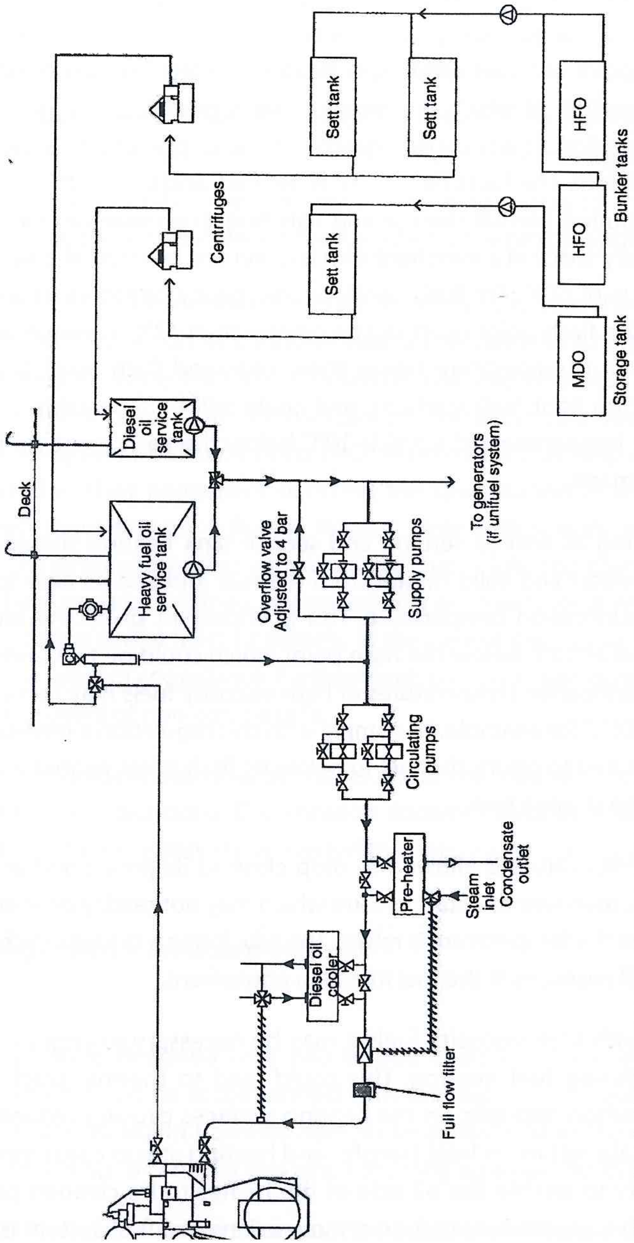
the vessel is in climates that can be considered temperate. The heating capacity of the fuel system, including the tank heating, trace heating and main system heating should be able to deal with the viscosity of any fuel the vessel is likely to encounter. Tank heating must be able to maintain temperatures above the maximum likely pour point.

Keeping the flash point of a fuel within specification is a legal requirement. The flash point is the temperature at which any vapour that is given off will ignite when an external flame is applied. This is usually quoted as the temperature measured under standardised conditions. The fuel's flash point is defined, and kept within tolerance, to minimise fire risk during normal storage and handling. The minimum flash point for fuel in the machinery space of a merchant ship is governed by international legislation and set at the value of 60°C. For fuels used for emergency purposes, external to the machinery space the flash point must still be greater than 43°C. However even when residual fuels are at a temperature below their measured flash point they are still capable of producing light hydrocarbons, and could still be flammable. The normal maximum storage temperature of a fuel is 10°C below the flash point, unless special arrangements are made.

Storage tank heating as well as settling and service tank heating should maximise the separation of water and solid matter from the fuel and still be able to maintain the correct post-purification temperature. This is important given the requirement for fuel to be stored at 10°C below the flash point which could be 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 to return the fuel to below its flash point-related value, as it is returned back to the storage tank.

If a fuel storage temperature is allowed to drop close to its pour point at any stage during the storage, then wax can start to form which may not readily be absorbed into the fuel again when the temperature is raised. The wax forms a sludge which can block filters and the small passages in the fuel injection equipment.

When operating with high viscosity fuels it may be necessary to employ high rates of heat transfer during fuel heating. This could lead to thermal cracking of the fuel resulting in carbon deposits on the heating surfaces causing 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



gure 3.18 *General arrangement of a modern fuel system*

Many fuel-related problems will not arise if an effective 'on-board fuel management policy' is adopted and followed through closely by each of the crew who serve on-board. To recap, such a policy would include the following:

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 to 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 and 10°C below the flash point.
5. Sending a representative fuel sample for analysis and taking the appropriate action upon receipt of the results.
6. Fitting a certified mass flow meter to accurately measure the quantity of bunkers delivered free of contamination.
7. Draining water from tanks at regular intervals.
8. Monitoring fuel consumption against fuel remaining on-board. This should be achieved by daily measurement of all fuel tanks: on older ships dipping the fuel tanks will probably provide the most accurate results. However, on a vessel fitted with accurate depth gauges the remote readings will provide the data required.
9. Regular checks of the fuel purification plant to see if excess water or solid impurities are being removed by the purifiers.
10. Temperature and viscosity control of the fuel from the storage tanks to the point of use by the machinery is very important. If the fuel temperature drops close to its pour point that the fuel filters could become clogged.
11. If the bunker tanks are filled to the very top and the temperature is slightly low, then as the temperature rises so will the volume. It will not take much for the fuel to rise up the vent pipe and cover the vessel with fuel which could spill into the water causing the vessel to be fined or the master and chief engineer arrested.

Analysis of the fuel

As already stated fuel should now be supplied according to the specification set out operators as well as engine manufacturers. Analysis of fuel oil is now recommended. Diligent operators are encouraged to seek the help of one of the oil majors and have their fuel oil samples collected and analysed at short notice. Some problems identified could be as follows:

1. *Low flash point:* Regulations require a flash point above 60°C. If the flash point is found to be below this level then 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:* Sulphur is present in crude oil and the specific amount depends on the original source of crude oil used and the type of refining process. During combustion, sulphur is converted into sulphur oxides which become corrosive upon contact with water and if left unchecked will damage engine pistons and cylinder liners. The acids produced must be neutralized by the cylinder lubricant and marine engine lubricants are especially developed with a high BN to cope with this acidity. If the correct lubricant is used, the sulphur content of a marine fuel is technically not important but the increasing environmental implications is now of great concern to the legislators.

Annex VI of MARPOL 73/78 sets out the sulphur content of any fuel oil used on-board ships which originally were not to exceed 4.50% m/m max. After 2010 both Annex VI and the EU directive 2005/33/EC restricted the SOx emissions of ships sailing in the Baltic Sea SECA to 6 g/kWh which corresponds to a fuel oil sulphur content of maximum 1.5% m/m. In addition, the EU directive extended the 1.5% m/m sulphur limit to ferries operating to and from any EU port. The North Sea and English Channel have now become a SECA area where the 1.5% m/m sulphur limit applied. The EU directive has further set a limit of 0.1% m/m max on the sulphur content of marine fuels used by ships at berth (and by inland waterways), which was effective from 1 January 2010 which also becomes a SECA area requirement from 2015.

The knock on effect is that from 2015 the current generation of marine engines will not achieve these low levels of emission without additional 'after engine' technology such as selective catalytic reduction (SCR) (see p. 313).

3. *High water content:* This may separate when heated, however water could also form a stable emulsion which is difficult to separate without the addition of emulsion breaking chemicals. If the water contamination is salt water, not uncommon in the marine environment, serious problems ...

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 by far 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. However, vanadium is bound chemically within the fuel and as a consequence cannot be removed. The vanadium deposits are very hard and can cause extensive damage to turbo-chargers.

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. Therefore fuels from different sources should not be mixed on-board.

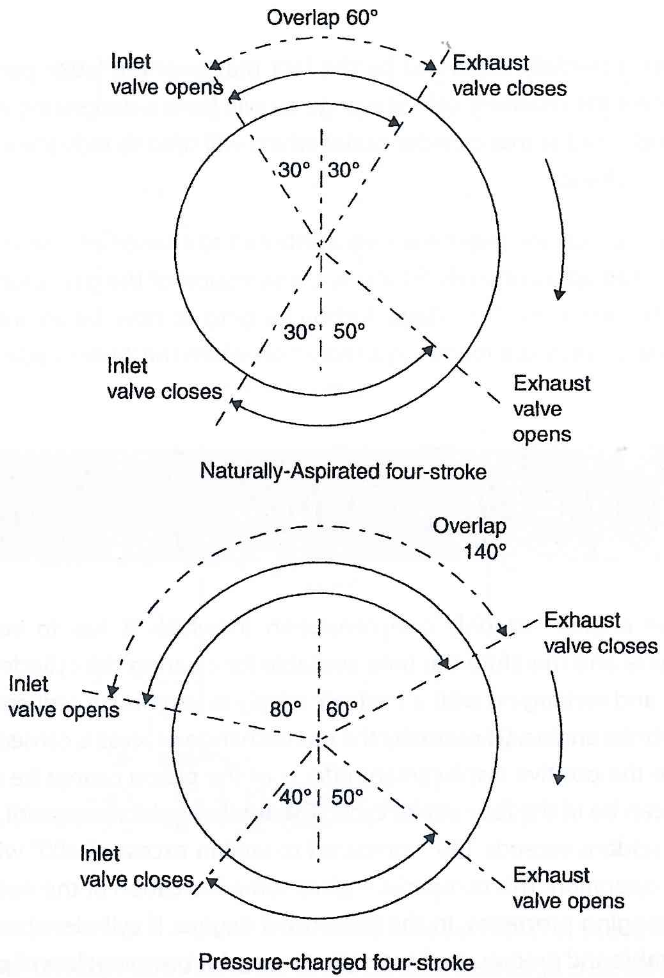
6. *High aluminium content:* This contamination is a result of carry-over of 'catalytic fines' from the refining process of the initial oil. These 'fines' are an aluminium compound ranging in size from 5 mm to 50 mm and are extremely abrasive. Very low levels of aluminium indicate the presence of catalytic 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. As a result of the small size of these compounds they are difficult to be removed 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 in attaining a safe level of 10 ppm after purification. Due to the problem of 'cat fines' the 2010 version of the ISO 8217 specification for fuel oil was introduced.

4

SCAVENGING AND SUPERCHARGING

It used to be very simple. For maximum performance and economy to be maintained it was essential that during the gas exchange process the cylinder was completely purged of residual gases at the completion of the exhaust phase of the cycle and a fresh charge of air introduced into the cylinder ready for the following compression stroke and this is still the ultimate aim for the current generation of engines. However, even the most efficient systems still leave behind unburnt hydrocarbons from the previous cycle and as we shall see later the most fuel efficient engines are using techniques such as exhaust gas recirculation (EGR) which is an attempt to burn some of the unburnt gasses and reduce harmful emissions from the engine.

In the case of four-stroke engines, purging the cylinder of the gasses from the previous cycle is relatively easy and 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 figure 4.1 for both normally aspirated and pressure-charged four-stroke engine types. Crank angle available for exhaust and induction with normally aspirated engines is seen to be of the order of 420°–450° with a valve overlap of 40°–60° depending upon precise timing – with more modern pressure-charged engines this



▲ **Figure 4.1** *Typical timing diagrams*

(b) 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 (b) it becomes clear that thermal stressing of engine parts is relieved and with exhaust gas turbo-charger operation prolonged running at excessively high temperatures is avoided. This latter process would have an adverse effect on materials used in turbo-charger construction and could also contribute towards increased contamination.

90° higher. This is 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 that cylinder outlet which will tend to indicate a mean value over the cyclic exchange.

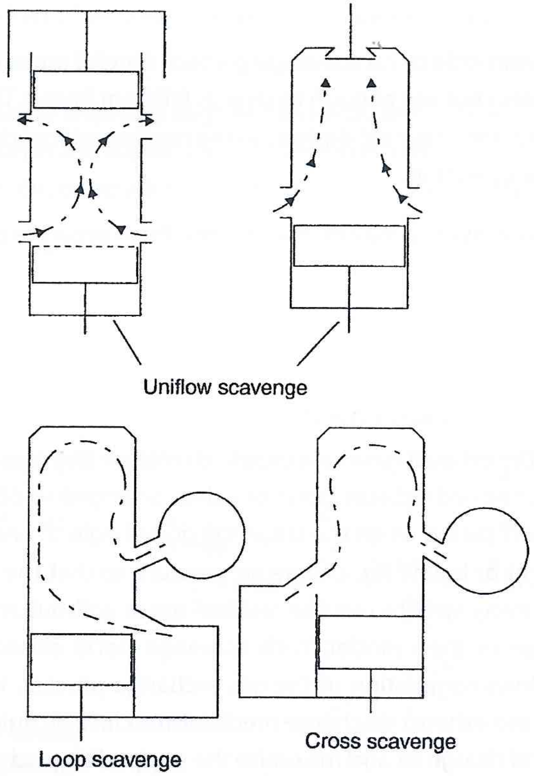
More probably the increase may be largely attributed to change of kinetic energy into heat energy and an approximately adiabatic compression of the gas column between cylinder and turbine inlet. Two-stage turbo-charging is now being introduced to increase the charge pressure following a reduction when the 'Miller' cycle is used (see Chapter 1).

Two-Stroke Cycle Engines

The two-stroke engine has only one revolution in which it has to complete the combustion cycle and therefore the time available for clearing the cylinder of residual exhaust gases and recharging with a fresh air supply is very much reduced compared with the four-stroke engine. Of necessity the gas exchange process is carried out around the BDC where the positive displacement effects of the piston cannot be exploited as much as they can be in the four-stroke cycle. The total angular movement, of the two-stroke piston, seldom exceeds 140° compared to well in excess of 400° with the four-stroke piston's operation. This comparison gives some indication of the need for a high efficiency scavenging processes, in the two-stroke engine, if cylinder charge is not to suffer considerable and progressive contamination and subsequent loss of performance as well as increased temperature and thermal loading. Prior to the introduction of turbo-charging to two-stroke machinery there was a need for a low degree of pressure charging of 1.1–1.2 bar to ensure adequacy of the gas exchange process. This was carried out by the use of a scavenge pump or in smaller engines by the use of under piston pressurisation of the inlet air that was fed through a transfer port to the mail cylinder.

Modern two-stroke engines are all turbo-charged making the scavenging process much easier. The scavenging arrangement of two-stroke engines is generally described as: (a) uniflow or longitudinal scavenge and (b) loop and cross scavenge (figure 4.2).

All the latest large two-stroke marine engines currently under construction have been designed to work with the uniflow system that employs a poppet-style exhaust valve



▲ **Figure 4.2** *Scavenging of two-stroke engines*

In the uniflow system the charge air is admitted through ports at the lower end of the cylinder and as it sweeps upwards towards the exhaust discharge area, 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 a degree of turbulence within the charge which is required for good combustion when fuel injection takes place.

Both cross and loop scavenge systems have 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 or upper exhaust ports and the associated operating gear. This simplifies the engine construction considerably and, in the past, it might also have led to a reduction of maintenance. Due to a simplified cylinder head construction straight to exhaust with little or no scavenging effect. Careful attention to port design did reduce this problem but not enough to stop its fall from favour. There will of course still be engines working to these old designs so the marine engineering student should be familiar with their operation.

The gas exchange process itself may be divided into three separate phases:

- blowdown
- scavenge
- post-scavenge.

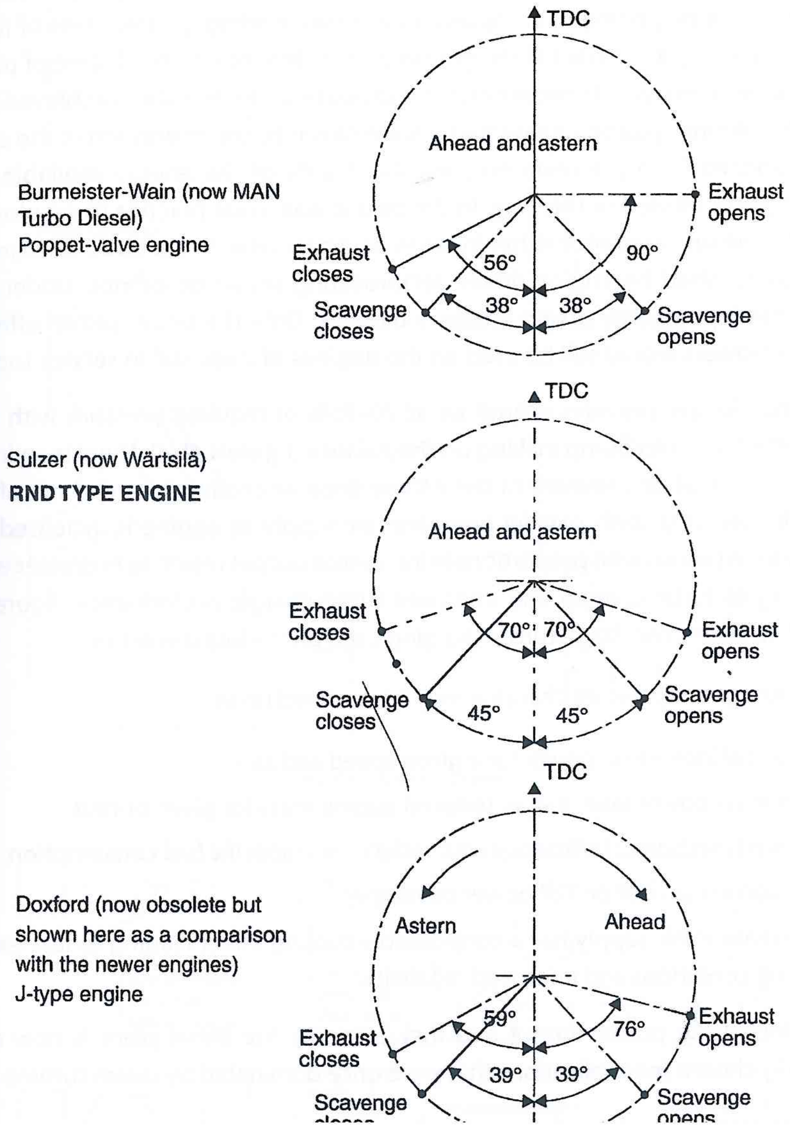
During 'blowdown' the exhaust gases are expelled rapidly – the process being assisted by generously dimensioned exhaust ports or valves arranged to open rapidly. At the end of this 'blowdown' period when the scavenge ports begin to uncover, the cylinder pressure should be at or below the charge air pressure so that the scavenge process which follows effectively sweeps out the residual gases without any resistance from a pressurised charge in the cylinder. With scavenge ports closed again the post-scavenge period allows completion of the gas exchange process. The engine design should ensure that the exhaust discharge mechanisms close as quickly as possible to prevent undue loss of charge air and maximise the trapped air ready for the beginning of compression, giving the highest possible density of charge ready for combustion.

Although some loss of charge air is unavoidable it should be borne in mind that the air supply is 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 production of NOx during combustion, happens at the peak temperatures during the process. Therefore, if these peak temperatures are reduced then so is the volume of environmentally harmful NOx gasses. In the latest engines this is accomplished by using the 'Miller' cycle (see Chapter 1), which modifies the timing of the inlet and exhaust valves to ensure that there are no peak temperatures produced during the combustion process. This can only be done with an engine that has full control over the inlet, exhaust and the start and stop of the fuel injection. Also with this system the pressure charging is increased with the use of two-stage turbo-charging.

The increased cylinder pressures encountered with modern turbo-charged machinery may result in exhaust opening being advanced so that sufficient time is given for

opening, since the loss of expansive working is more than offset by the gain in turbo-charger output.

Obviously in the case of reversing engines there may be some slight penalty incurred if prolonged operation in the astern direction is considered. Figure 4.3 shows the timing for some of the present generation of direct drive slow-speed diesels.



Pressure Charging

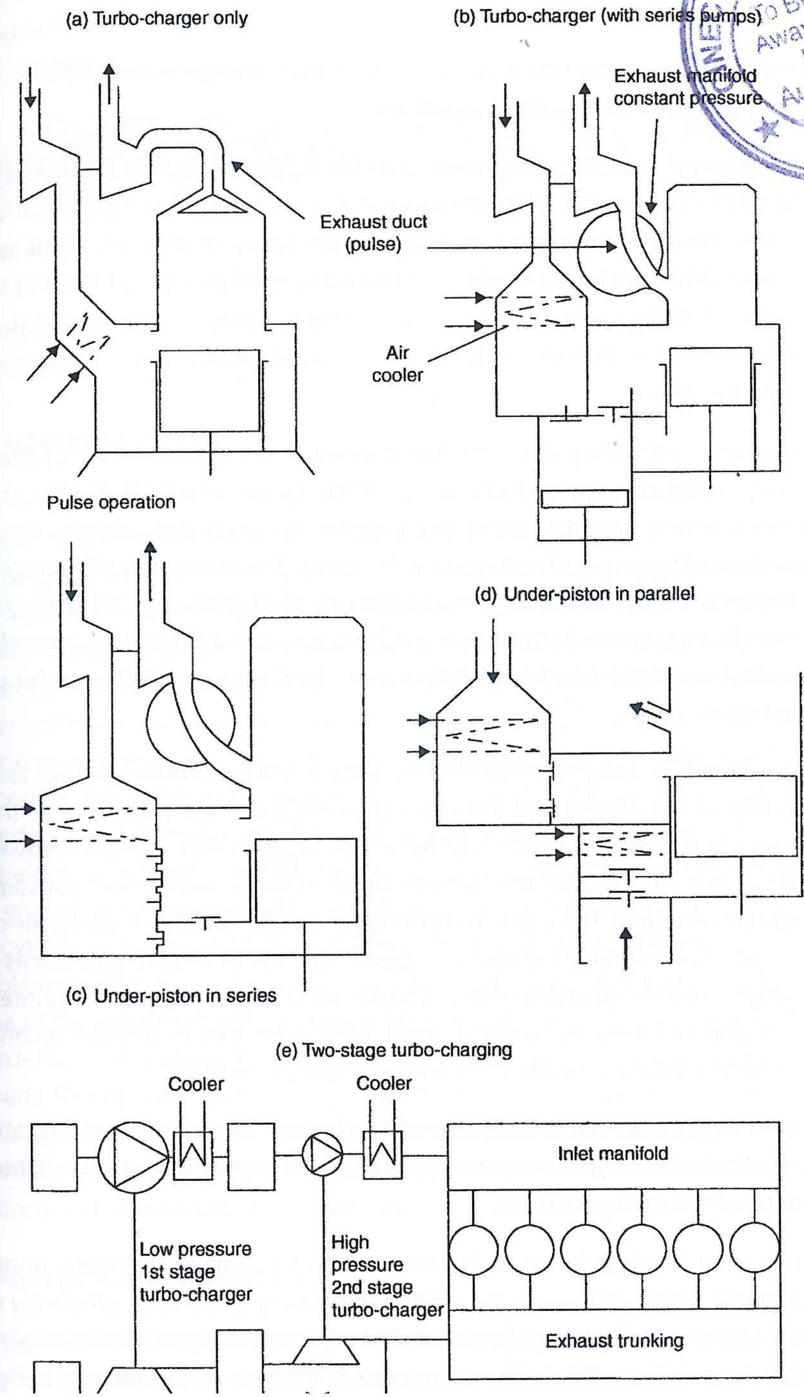
By increasing the density of the air, and therefore the mass of oxygen present in the cylinder at the beginning of compression a corresponding greater mass of fuel can be burned giving a substantial increase in power developed. The degree of pressure charging required, which determines the increase in air density, is achieved by the use of free-running turbo-chargers which are driven by the energy left in the exhaust gases expelled from 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 in parallel with the turbo-chargers. This was accomplished by engine-driven reciprocating scavenge pumps, under piston effect or independently driven auxiliary blowers. Only the under-piston effect and auxiliary blowers would still be used on the engines of ships still in service today.

The turbo-charger provides charge air at 70–95% of required pressure with under-piston effect or series pump making up the balance (figure 4.4b,c). There is an increase in temperature of air delivered to the engine since air cooling is carried out after the turbo-charger only. 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 turbo-charger and improved turbo-charger performance (figure 4.4d). Figure 4.4e shows two-stage turbo-charging used on the latest engines.

The advantages of pressure charging may be summed up as:

- substantial increase in power for a given speed and size
- better mass power ratio, that is, reduced engine mass for given output
- improved mechanical efficiency with reduction in specific fuel consumption
- reduction in cost per unit of power developed
- an increase in air supply has a considerable cooling effect leading to less exacting working conditions and improved reliability.

Due to increasing power output and fuel economy the diesel plant is now almost universally chosen for applications that were once dominated by steam turbine plant.



driving the quest for a reduction in emissions from marine engines and the importance of the turbo-charger in this quest is significant.

The turbo-charger manufacturers have invested in new research using the latest computation fluid dynamics (CFD) techniques and as the knowledge base and advances in material science move forward, so does the efficiency of the new generation of turbo-charger. While further work still has to be completed in this area to meet the full requirements of IMO regulations, the industry now has other challenges as the total energy efficiency of ships starts to focus the minds of naval architects and ship and engineering designers.

However, energy efficiency does not just depend upon the efficiency of the main engine; increasing the thermal efficiency of the power plant takes into account waste heat recovery systems (WHR), see Chapter 9, which also contributes to the vessel's overall efficiency and reduction in life costs. The increasing efficiency of the turbo-charger is the key in allowing the development of sophisticated WHR systems. The more efficient turbo-chargers use less heat energy to drive the turbo-charger for the operation of the engine, therefore more heat energy is left over for use by additional WHR.

The improvements in design include the ability to control the turbine's output to match the operation of the engine and this is set to provide owners with engines that are flexible and can therefore be optimised for vessel requirements. These improvements are resulting in a high overall efficiency of up to 70%, impacting fuel consumption and firing temperatures. The latest turbo-charger design features include: air-cooled operation, which will reduce the cost, complexity and installation requirements of the turbo-charger; a cartridge-style construction aimed at improving on-site maintenance procedures and reducing operational down time; advances in bearing technology, contributing to a reduction in life costs and extending service life.

Alternative compressor options are also enabling better turbo-charger optimisation for specific applications. Compressors are now made from aluminium which is lighter and therefore takes less energy to drive.

Variable vane technology is set to increase the turbo-charge's flexibility, improving the operational range and giving the engine that ability to perform efficiently over a wider operational envelope. High pressure ratio capability of up to 5:1 can be achieved in a single stage using an aluminium compressor, without compromising design life.

carrying out work on these advanced machines, by non-service engineers, is no longer recommended.

Constant pressure and pulse operation

In general the manner in which the energy contained within the exhaust gases is utilised to drive the turbo-charger may be described in two ways:

1. The pulse system of operation
2. Constant pressure operation.

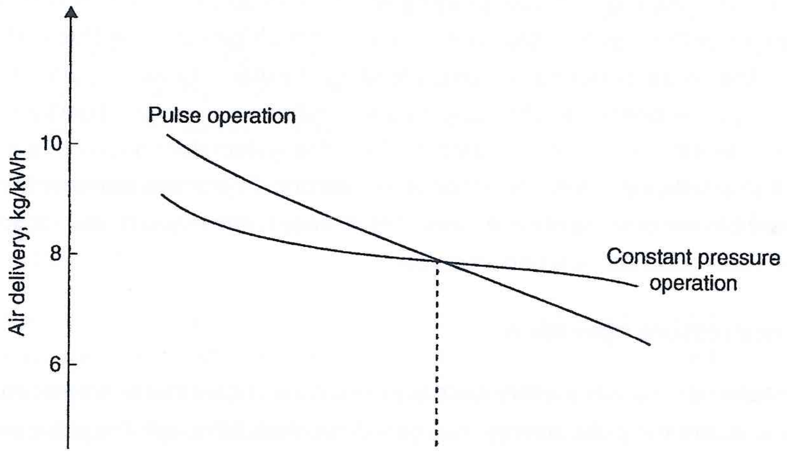
Pulse operation

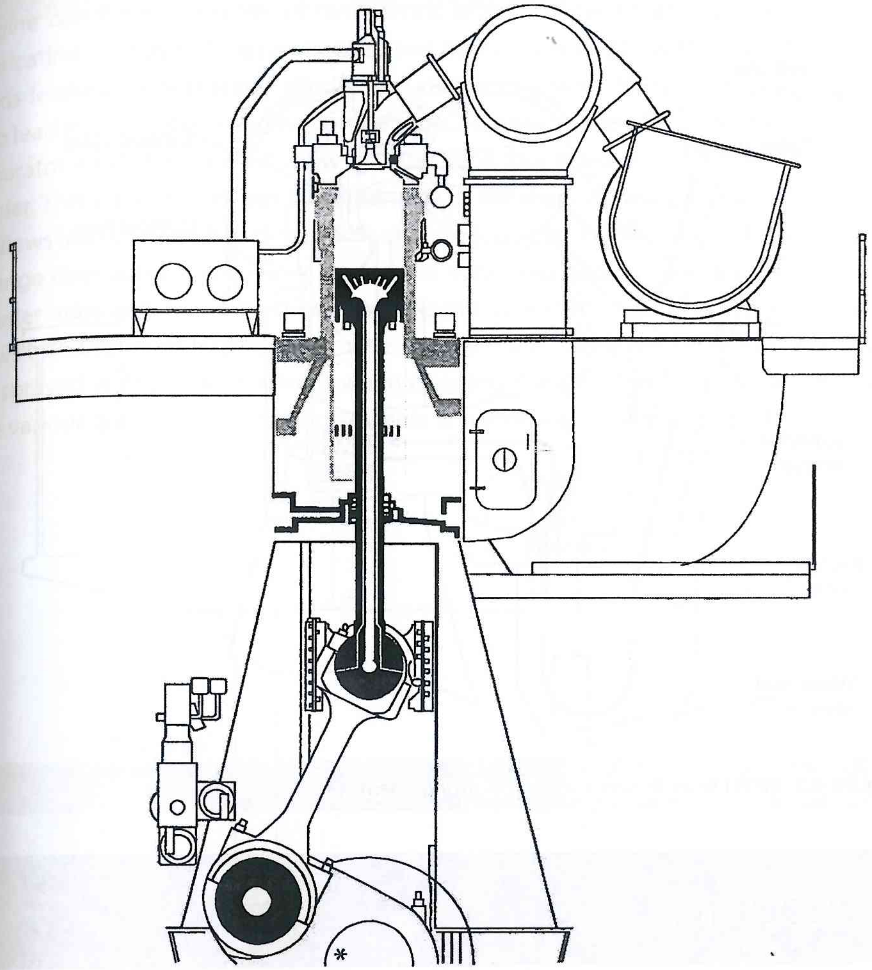
This makes full use of the higher pressures and temperatures of the exhaust gas during the blow-down period and with rapidly opening exhaust valves or ports the gases leave the cylinder at high velocity as pressure energy is converted into kinetic energy to create a pressure wave or pulse in the exhaust leading to the turbo-charger. For pulse operation it is essential that the exhaust leading from the cylinder to the 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 turbo-charger do not interfere with each other to cause pressure disturbances that would affect purging and recharging with an adverse effect upon engine performance. With two-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, turbo-charging 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 contained in the exhaust gas becomes increasingly dominant so that sacrifice of pulse energy in a large volume receiver is of less consequence. Figure 4.5 shows the results of tests carried out on a Wärtsilä two-stroke engine which indicates that up to a BMEP of around 7 bar the advantage lies with the pulse system but as the BMEP increases beyond this figure the constant pressure system becomes more efficient giving greater air throughout and some slight reduction in the fuel rate.

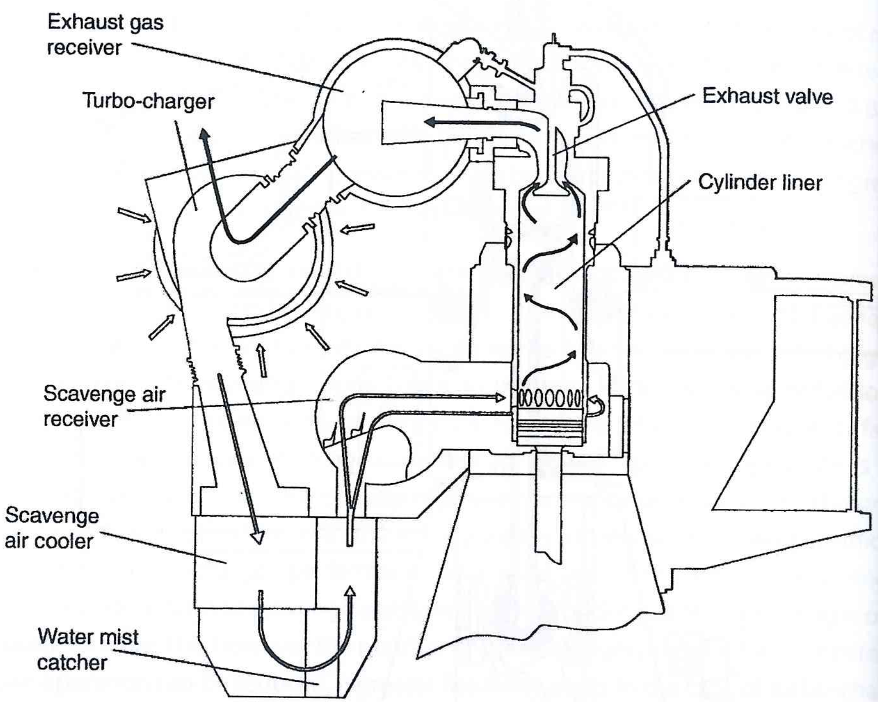
Due to the much larger volume of the exhaust system associated with constant pressure operation the release of exhaust gas is rapid and earlier opening to exhaust is generally only necessary to ensure that cylinder pressure has fallen to or below the charge air pressure when the scavenge ports begin 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 turbo-charger to changing engine conditions with the energy supply at slow speeds being insufficient to maintain turbo-charger 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 turbo-chargers required as compared to pulse operation can be reduced, a greater flexibility exists in the case of turbo-charger location and exhaust arrangement and no de-rating of engine need to be considered for cylinder groupings other than multiples of three. For this reason most large slow-speed two-stroke engines tend to be of the constant pressure configuration.





▲ **Figure 4.6** *Wärtsilä RTA scavenge arrangement*

Figure 4.6 shows the diagrammatic arrangement of the Wärtsilä RTA scavenge 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 space 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 preset value. Air is drawn from space A and delivered into space F and this together

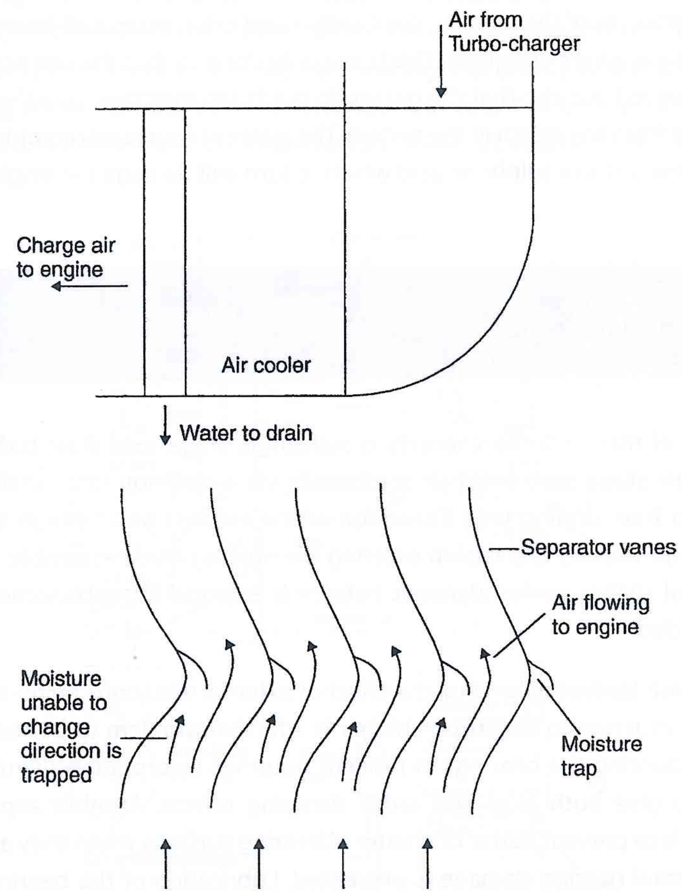


▲ **Figure 4.7** *MAN Diesel 'M' series scavenge arrangement*

Air Cooling

During compression of the air at the turbo-blower, which is fundamentally adiabatic, the temperature may increase by about 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 that 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.

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 carryover of water a water separator is fitted. Figure 4.8 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. Due to 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.

Some engine manufacturers are introducing water injection into the combustion process. This is different from the water contamination described above because it is a carefully designed system that has been developed following an extensive research and development programme where the effect on all the engine components and fluids will have been considered and any adverse effect will have been removed as part of the engine's design. The reason for the water injection is to reduce the peak temperatures of combustion thus reducing the harmful NOx.

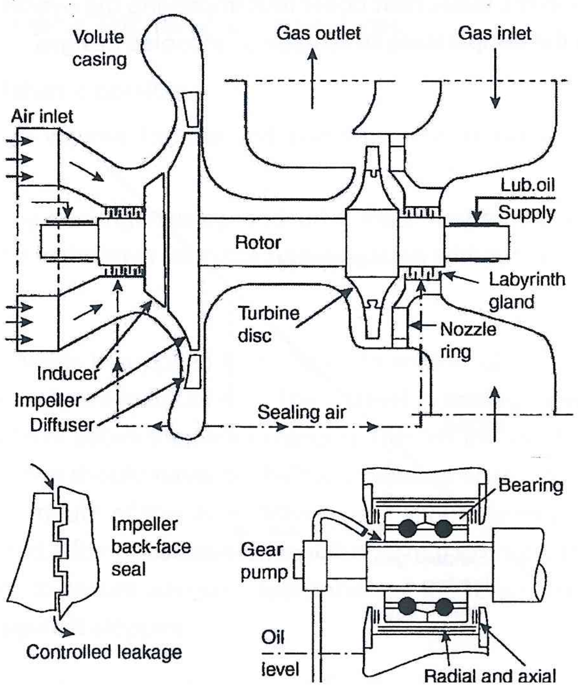
Another very important point which needs to be understood by the students presenting themselves for examination is the reason for not overcooling the charge air. The temperatures of the inlet air, combustion and exhaust have all been calculated carefully by the engine's designers. This is not only done so that the correct density of air can be achieved but also that the gasses do not fall to their 'dew point' where water will be formed from any steam in the system. The water of course will combine with any oxide of sulphur to form sulphuric acid which in turn will damage the engine or other components.

Turbo-chargers

The majority of marine turbo-chargers is still single-stage axial flow turbine wheel 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. Due to the high rotational speeds perfect dynamic balance is essential if troublesome vibrations are to be avoided.

Despite the high level of balancing, the effect of external vibrations being transmitted via the ship's structure to the turbo-charger 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 effects. Another aspect of this arrangement is to prevent flutter or chatter at bearing surfaces when they are stopped so that incidental bearing damage is prevented. Lubrication of the bearings may be

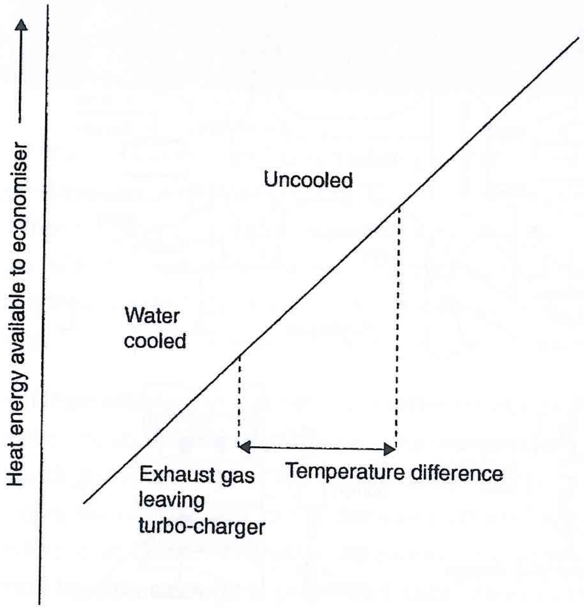
The various claims of superiority as to the effectiveness of the different types of bearings centre around the mechanical efficiency of the bearing configuration. The manufacturers of turbo-chargers equipped with rolling element bearings claim a distinct mechanical efficiency advantage across the whole operating range. On the other hand manufacturers of turbo-chargers equipped with sleeve-type bearings claim comparable efficiency under full-load conditions but admit 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 be 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, 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. Plain bearings should however have a life equal to that of the blower provided that normal operating conditions are not exceeded. Ball bearings can end up with tiny indentations in the rolling surface caused by vibrations from the vessel when the turbo-charger is at rest for longer lengths of time. There has also been a trend towards 'inboard' plane



bearings which enable the rotor to be supported without any bending as is the case when the bearings are at either end of the rotor.

Referring to figure 4.9 it can be seen that the blower end of the turbo-charger 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 turbo-charger consist of casings which house the nozzle-ring turbine wheel and blading, etc. In older designs casings were water cooled but in turbo-chargers for modern large slow-speed two-stroke engines, with relatively low exhaust gas temperatures the casings are uncooled. Uncooled designs retain more heat energy in the exhaust gas in the waste heat boiler thus improving the overall plant efficiency. Figure 4.10 shows the temperature advantage of uncooled designs.



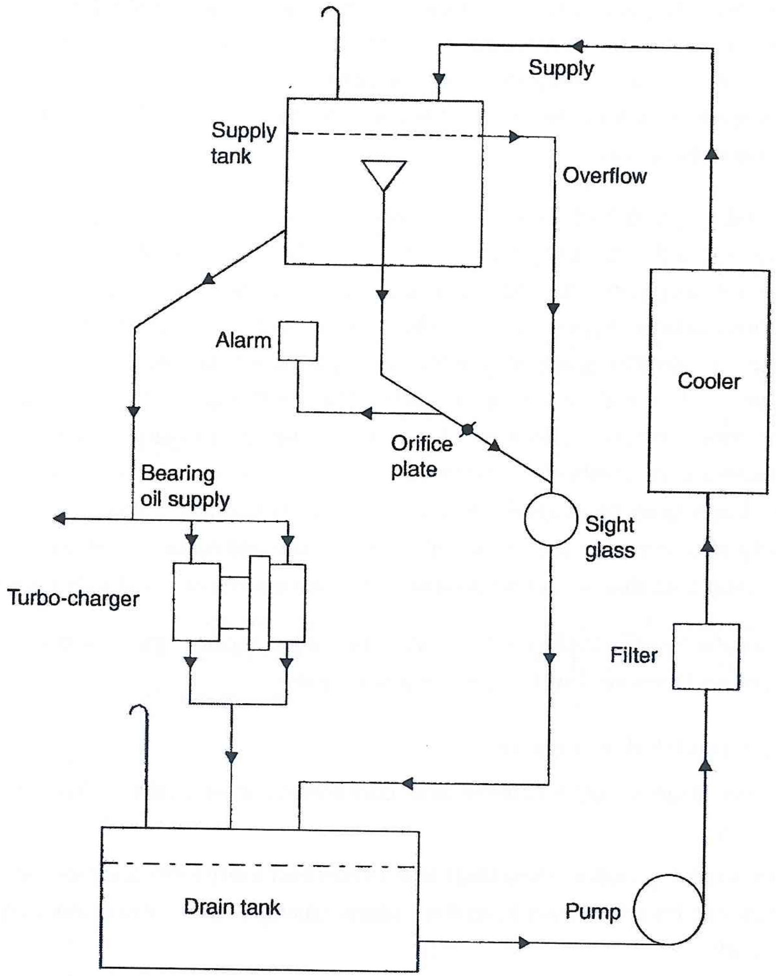
The components in the high-temperature gas stream, that is, the nozzle ring, turbine wheel, blades and rotor shaft are manufactured from heat resisting nickel-chrome alloy steel to withstand continuous operation at temperatures in excess of 450°C. Some degree of cooling may be given by controlled air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

Cooling media for cooled exhaust gas casings is generally from the engine jacket water cooling system although in some cases seawater 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:

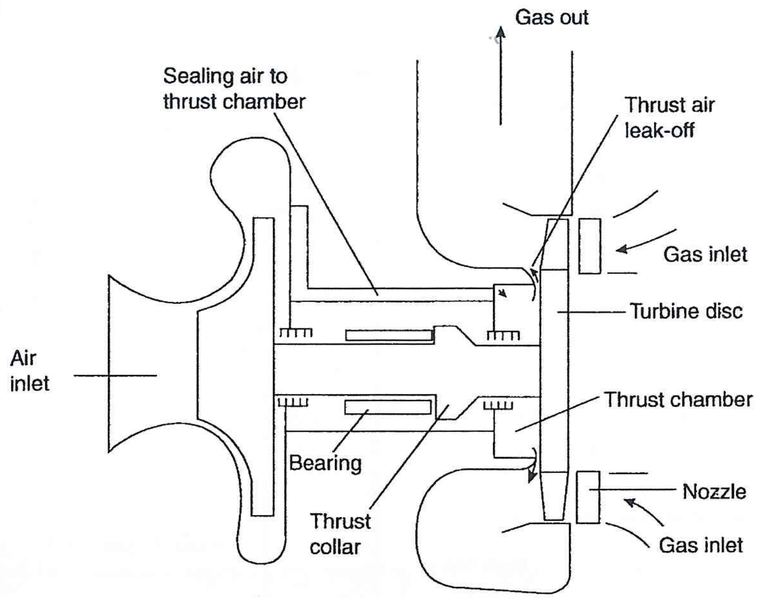
1. A short, rigid shaft is possible.
2. It allows large volume turbine and compressor inlet casings, free of bearing housings.
3. The main casing, bearing housings and turbo-machinery form one module allowing the rotor to be withdrawn from the turbine casing without disconnecting engine ductwork.

The oil for the bearings is supplied from the main engine lubricating oil system or a separate oil feed as shown in figure 4.11. The oil level in the high-level tank should be maintained about 6 m above the turbo-chargers. 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 is initiated. After an alarm it takes about 10 min to empty the high-level tank which is sufficient to ensure adequate lubrication of the turbo-chargers as they run down after the engine is stopped.



▲ **Figure 4.11** *Turbo-charger lubrication system*

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. Figure 4.12 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 turbo-charger 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

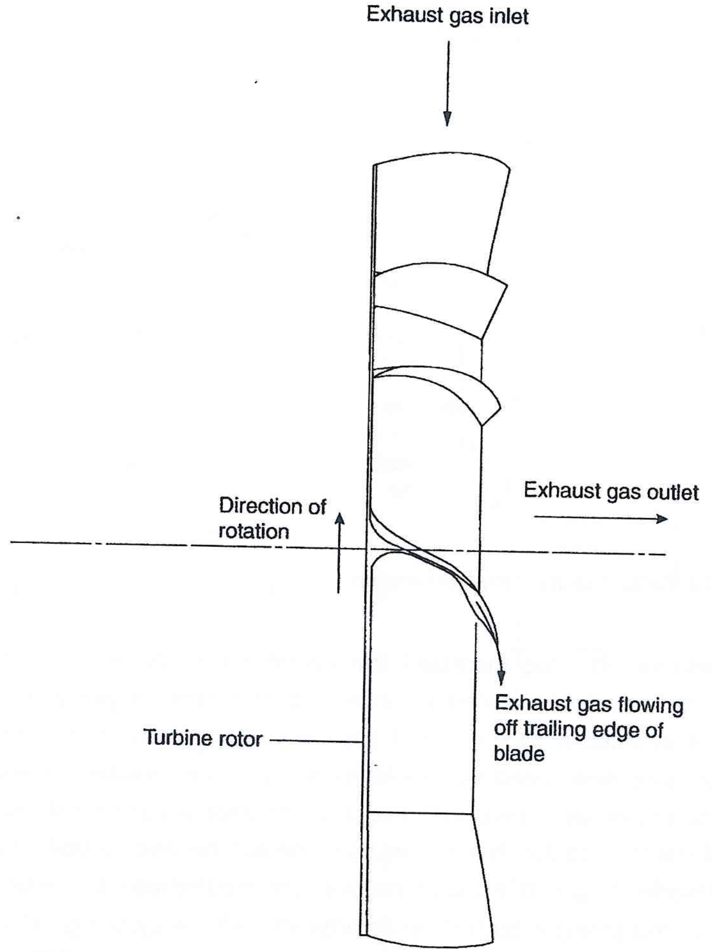


▲ **Figure 4.12** *Turbo-charger with plain bearings*

Recent developments have increased the overall efficiency of turbo-chargers 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 two-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

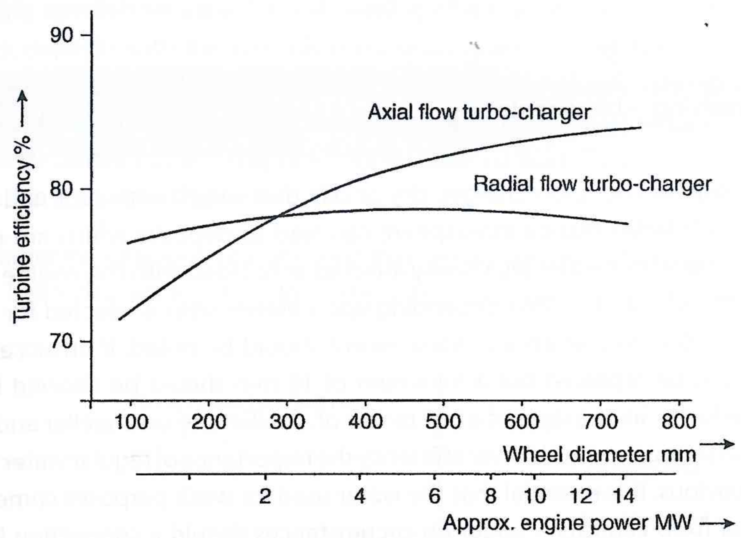
For smaller higher-speed diesel applications (690–6,700 kW range) the use of radial flow turbo-chargers is common (figure 4.13). The casings are uncooled but require insulation. Bearings are of the sleeve type and are lubricated from the engine lubricating oil system. The turbine wheel is a one-piece 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 giving stable operating characteristics. At high



▲ **Figure 4.13** *Radial flow turbine*

MAN Diesel & Turbo state the following characteristics for their NR series of radial flow turbines:

- For engine outputs from 450 to 5,400 kW per turbo-charger
- Maximum pressure ratio 4.5
- Maximum permissible temperature 650–720°C
- Suitable for heavy fuel, diesel or, biofuel and gas operation



▲ **Figure 4.14** *Comparison of radial and axial flow turbo-charger efficiency*

- Lubricated by the engines lube oil system
- Easy maintenance.

Turbo-charger Fouling and Cleaning

Turbo-charger fouling

Turbo-chargers that have contaminated turbines and compressors will have less efficiency and lower performance than their design specification, which results in higher exhaust temperatures. In four-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 turbo-charger efficiency it is important to ensure that all operating parameters are maintained according to the manufacturer's recommendations. 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

Water washing – blower side

On the air side of the turbo-charger, dry or oily dust mixed with soot and possibly salt from a salt-laden marine 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 (1–2½ l depending upon blower size) is injected for a period of from 4 to 10 s after which an improvement should be noted. If unsuccessful the treatment can be repeated but a minimum of 10 min 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. A cup full of water could be mixed with a general cleaning agent and carefully poured into the air filter as the turbo-charger is running.

Water wash – turbine side

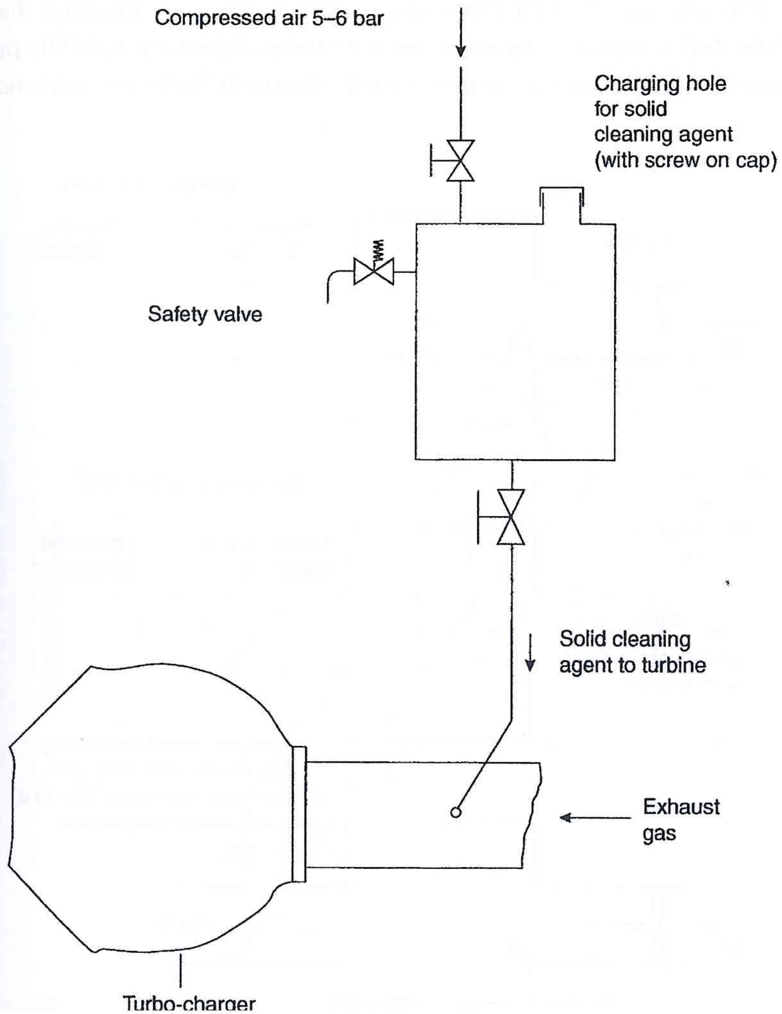
This must be carried out at reduced speed by rigging a portable connection to the domestic fresh water system and injecting water, via a spray orifice before the protective grating at turbine inlet, for a period of 15–20 min with drains open to discharge excessive moisture and or deposits which do not evaporate off. Since water washing may not completely remove all deposits, and can interact with sulphur, causing a 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 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

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 24–50 h. 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 figure 4.15.

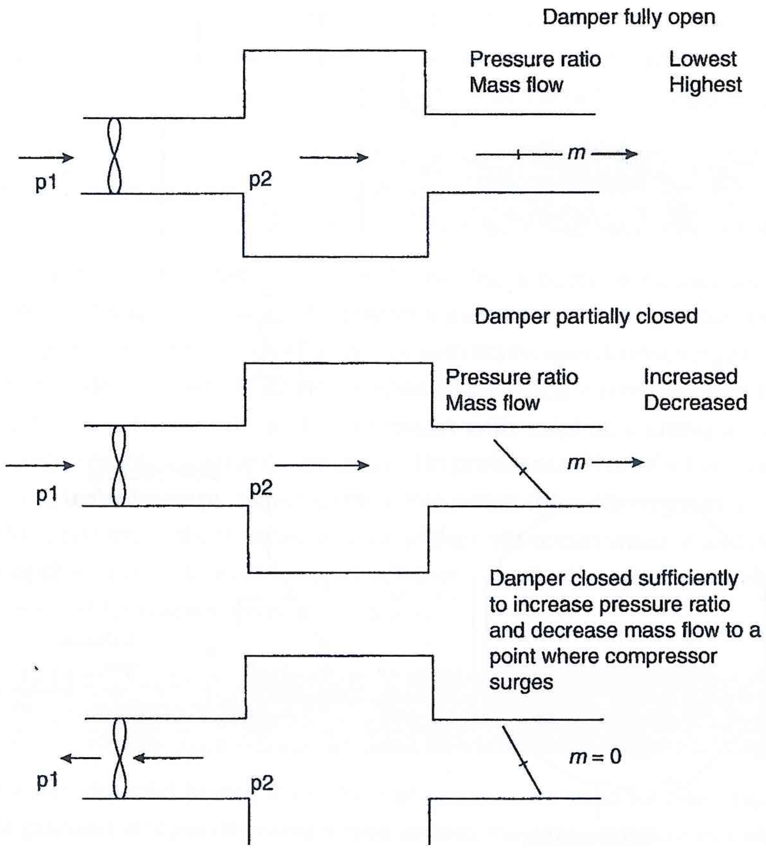
Turbo-charger manufacturers recommend that heavily contaminated machines, which have not been cleaned regularly from the very beginning or after overhaul, should

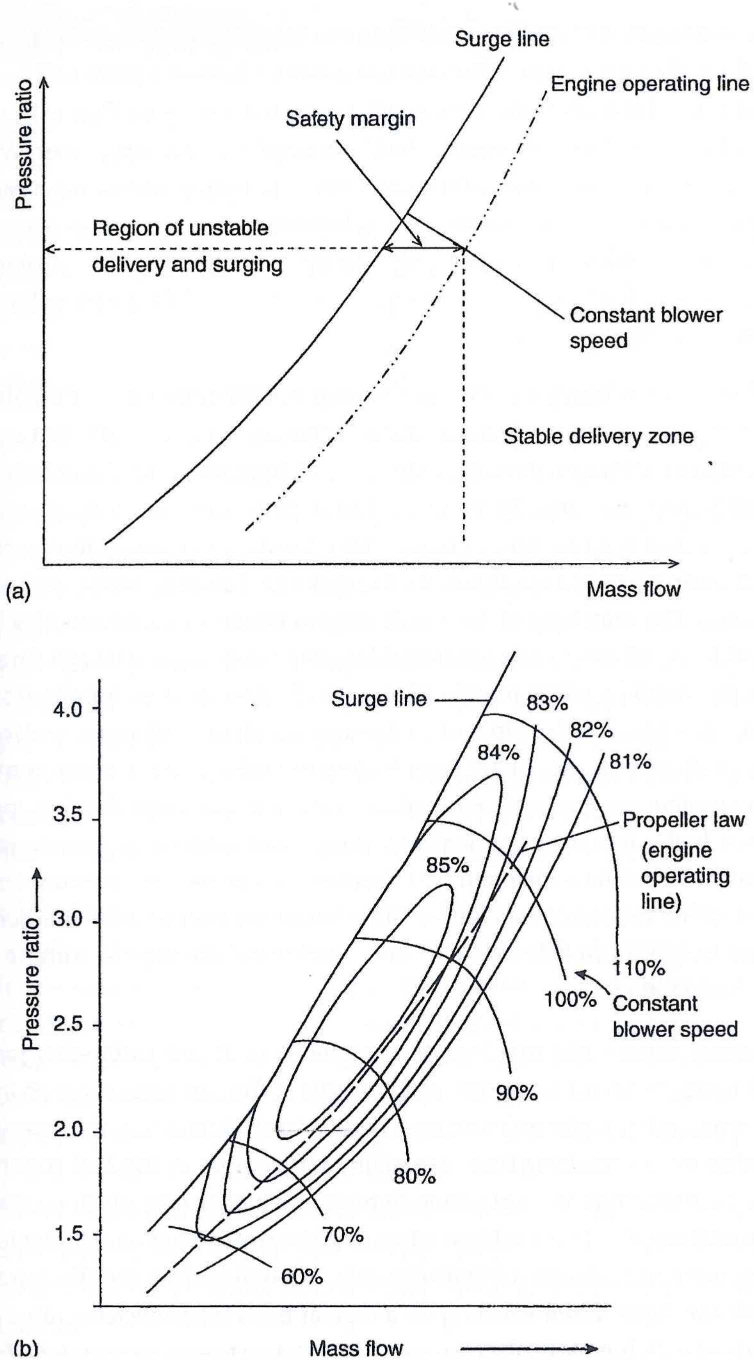


not be cleaned by water washing or granulate injection. This is because the dangers of incomplete removal of deposits may cause rotor imbalance. These turbo-chargers need careful dismantling and cleaning with the machine apart is recommended.

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 figure 4.16 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 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 (figure 4.17b).

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 in figure 4.17a 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 15–20% to allow for deterioration of service conditions such as fouling and contamination of turbo-chargers and increasing resistance of ship's hull, etc. Apart from fouling of turbo-charger 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 turbo-chargers due to unbalanced output from cylinder units and mechanical damage to turbine blading, nozzles or bearings, etc.

During normal service the build-up of contaminants at the turbo-charger can be attributed to deposition of air-borne contaminants at the compressor which in general are easily removed by water washing 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 from 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

Turbo-charger breakdown

Correct operating and shut down procedures, depending upon engine type, will be found in the engine builders and/or turbo-charger manufacturers' recommended practice. As a general rule however, in the event of damage to the turbo-chargers, the engine should be stopped immediately so that the damage is limited and broken fragments do not cause further damage elsewhere within the engine. Under conditions where the engine cannot be stopped, without endangering the ship, engine speed must be reduced to a point where the turbo-charger speed has dropped to a level where any vibration that is 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.

Where only a single turbo-charger has been affected, out of a number associated with the engine, 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 turbo-chargers becoming defective it is possible to remove blank covers from the scavenge air receiver so that natural aspiration supplemented by any under-piston effect, 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 forces, due to variation in output from affected units, do not bring about any undue engine vibration.

5

STARTING AND REVERSING

Starting Air Overlap

There must be some overlap between the operation of starting air valves to the different cylinders of an engine, so that as one cylinder valve is closing another one is opening just at the correct moment to ensure a continued rotation of the engine before the fuel is introduced. This is essential to ensure a positive angular motion of the engine crankshaft with sufficient momentum 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.*

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

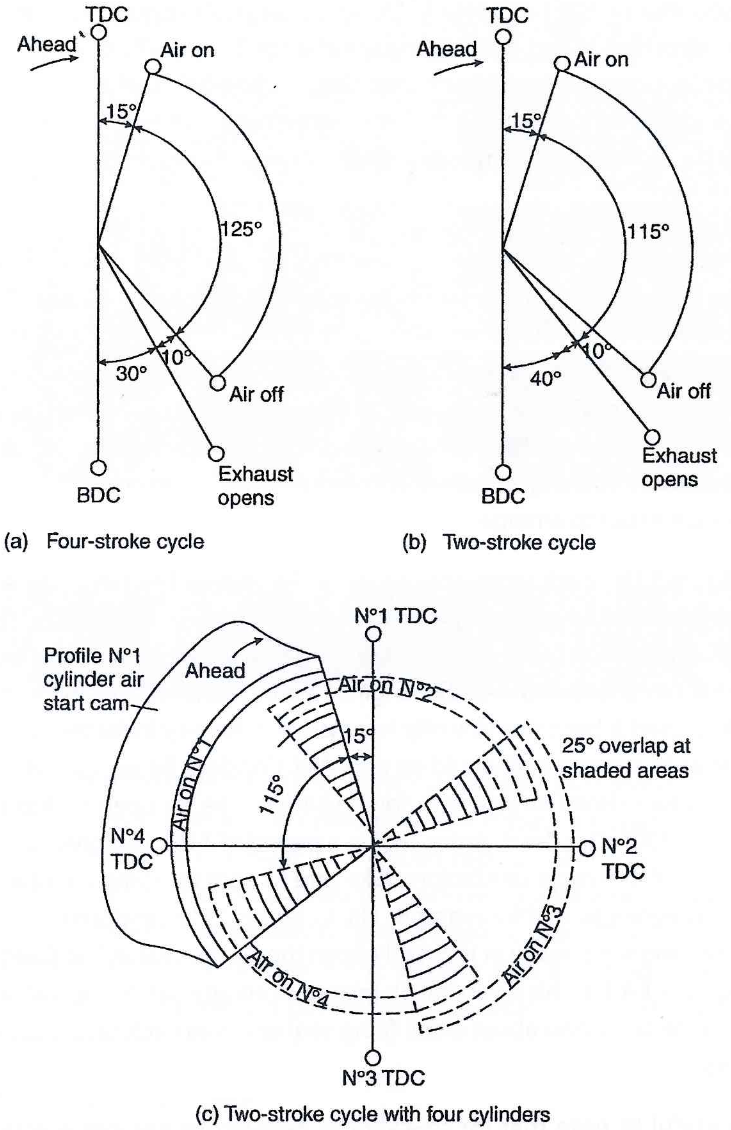
for example, with a four-cylinder two-stroke engine the firing interval is 90°, that is, 360/4 and if each cylinder valve covered 90° of the cycle then the engine would not

3. The cylinder starting air valve should allow the air to enter the cylinder after its associated piston has passed TDC 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 TDC because the engine is past this position before the valve is effectively open and the compressed air is having an effect. Any reverse turning effect is negligible as the turning moment exerted on a crank very near dead centre is small indeed.

Consider figure 5.1a for a four-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 firing interval for a six-cylinder four-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 turbo-charged engines. A seven-cylinder four-stroke engine is much easier to arrange.

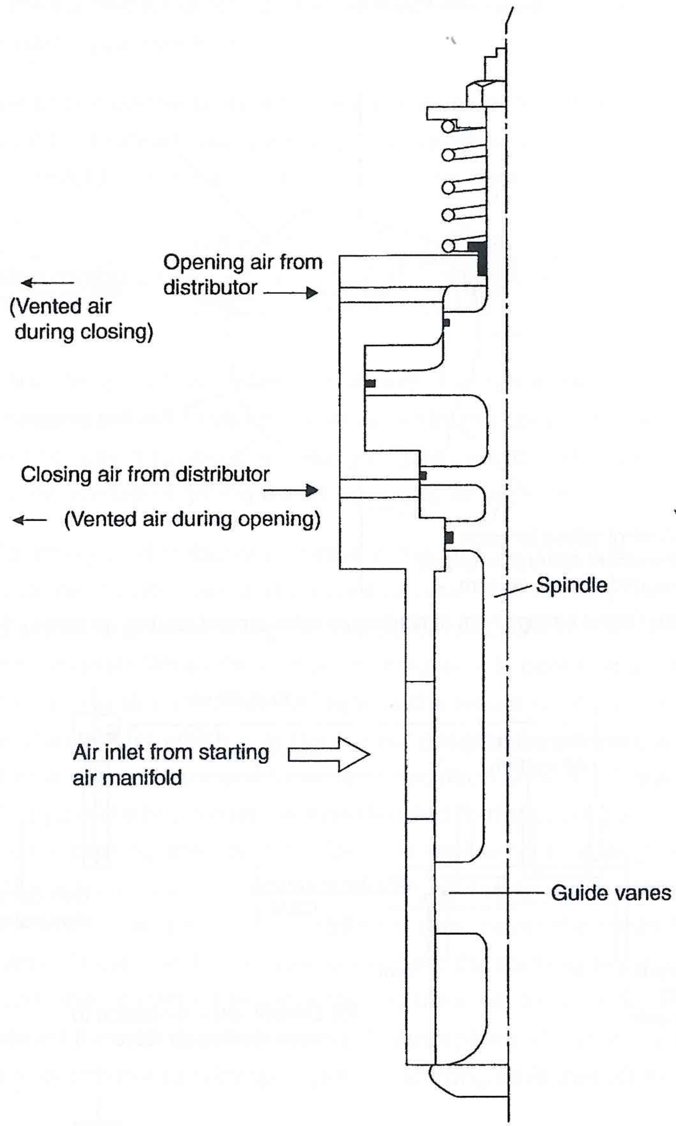
Consider figure 5.1b which represents a two-stroke engine. This has an air start period of 115°. Firing interval for a three-cylinder two-stroke engine = 360/3 = 120°. This means no overlap. Modification can arrange to give satisfactory starting with this example but for modern turbo-charged two-stroke engines having exhaust opening as early as 75° before BDC (outer) it becomes virtually impossible. A four-cylinder two-stroke engine is much easier to arrange and would be adopted. Consider figure 5.1c which is a cam diagram for a two-stroke engine with four cylinders. The air open period is 15° after dead centre to 130° after dead centre, that is, 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 six-cylinder, two-stroke engines a very common firing sequence is 1 5 3 6 2 4 and similarly for seven and eight 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 cylinder numbers 1, 4, 3 and 2 are shown respectively



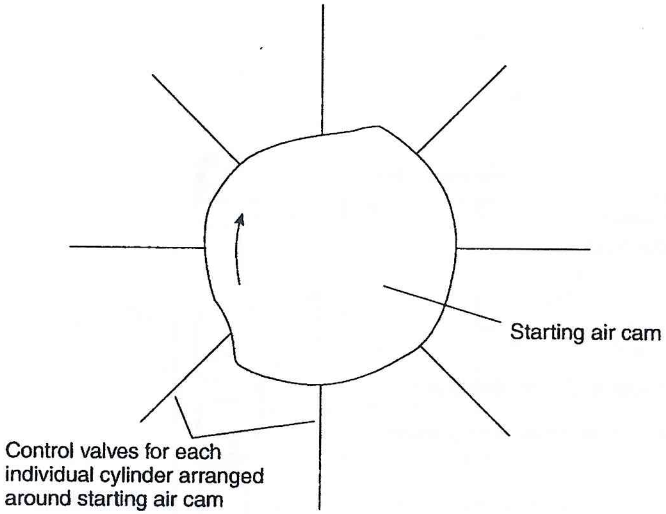
▲ **Figure 5.1** *Air start cam and crank timing diagrams*

Starting air valves

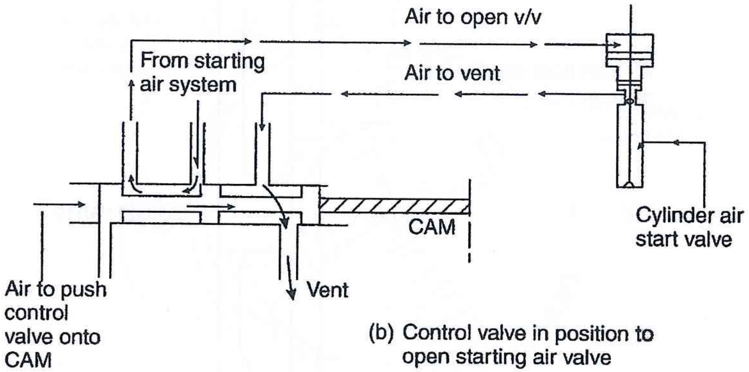


▲ **Figure 5.2** *Starting air valve*

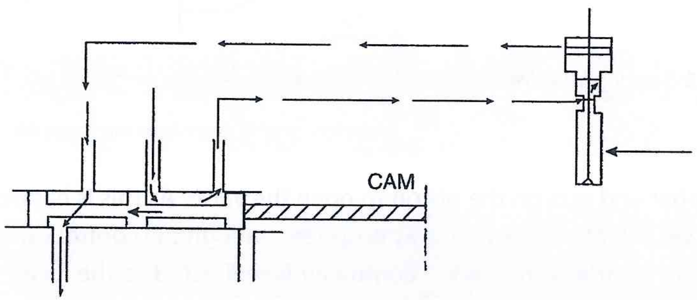
upper chamber and acts on the piston to open the valve. As this is happening the air from the lower chamber is vented to atmosphere through the control valve. At the end of the starting air admission period control air is redirected to the lower chamber to



(a) Radial arrangement of starting air valve around starting air cam



(b) Control valve in position to open starting air valve



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.

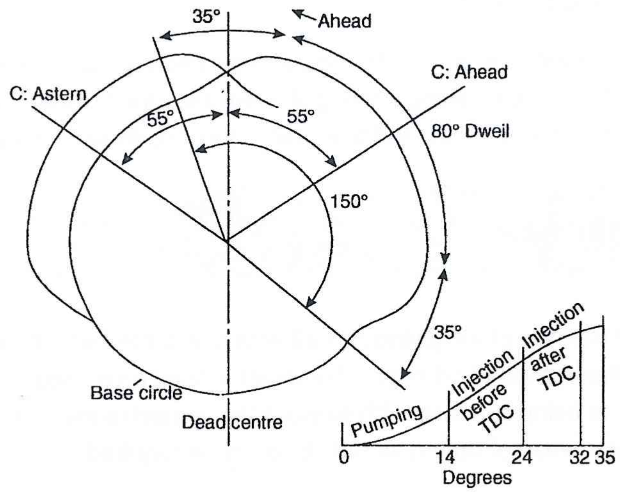
Starting air distributor

There are many designs of air distributor all with the same basic principles, that is, 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 figure 5.3a. This is based on a design where 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, 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 figure 5.3b. 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 figure 5.3c. 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 direct drive two-stroke engines. The general trend in four-stroke practice is to utilise a unidirectional engine, coupled, via a reduction gearbox, to a controllable pitch propeller. The need for reversing mechanisms on the four-stroke engines is therefore no longer required. For this reason the two-stroke reversing



▲ **Figure 5.4** *Lost motion cam diagram*

Two-stroke reversing gear

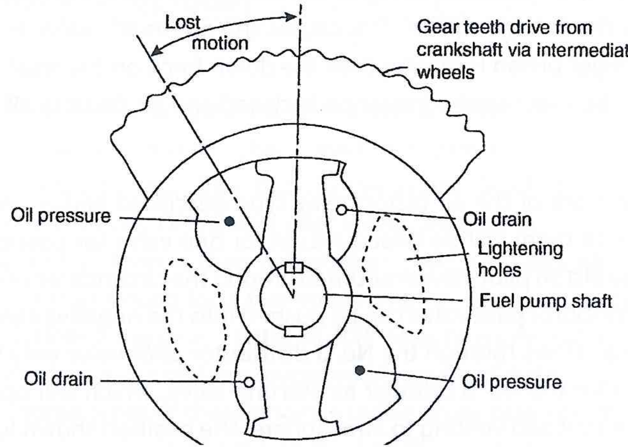
One of the easiest ways of changing the settings on a two-stroke engine is to reposition the fuel, exhaust valve and starting air cams on the camshaft, with their associated equipment, so that the engine operating in reverse can utilise one cam. This avoids the complication of moving the camshaft axially but it also means that it is necessary to provide a 'lost motion clutch' on the camshaft and the arrangement of such a clutch is described in the next few paragraphs.

Referring to figure 5.4, consider the engine piston to be positioned at TDC in the ahead mode with the fuel cam peak centre line at 55° after this position for correct injection timing ahead and assuming anti-clockwise ahead rotation. If now the 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 that the symmetrical cam 75° of each side of the cam peak centre line is made up of 35° rising flank and 40° of dwell.

The flank of the cam is shown on an enlarged scale in figure 5.4. 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

1. Older MAN Diesel & Turbo two-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. However, the latest engines are designed to move the position of the follower so that when the direction is reversed the closing cam profile becomes the opening profile and vice versa.
2. The dwell period is not normally necessary from the fuel injection aspect alone, that is, 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 Wärtsilä 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 figure 5.5. The design in this figure which is based on older Wärtsilä 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, that is, anti-clockwise as shown in figure 5.5. 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. It is worth pausing for a moment and reflecting upon the limitations of the mechanical designs. For example, think how difficult it would be to arrange a fuel cam with a profile that gave a pre-ignition and post-ignition phase to the injection process.

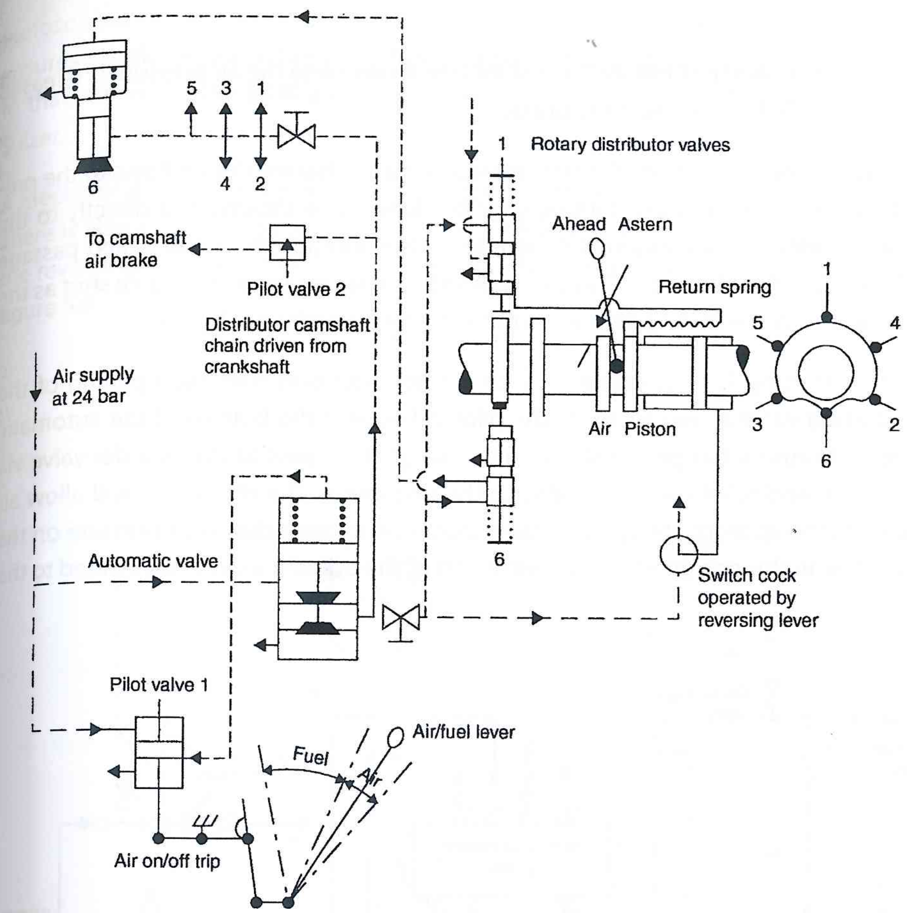
Practical Systems

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

Starting air system

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 figure 5.6. 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 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, that is, 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

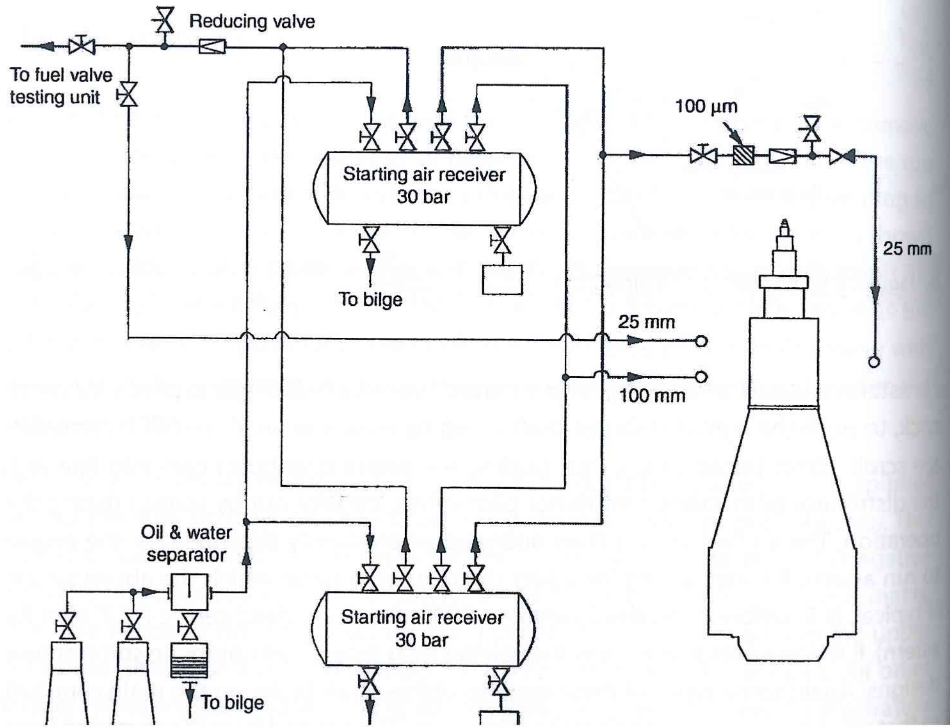


▲ **Figure 5.6** *Typical starting air system*

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.), thus 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 two-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 kept stationary and just before the lost motion is complete the air pressure is released to atmosphere thus releasing the brake.

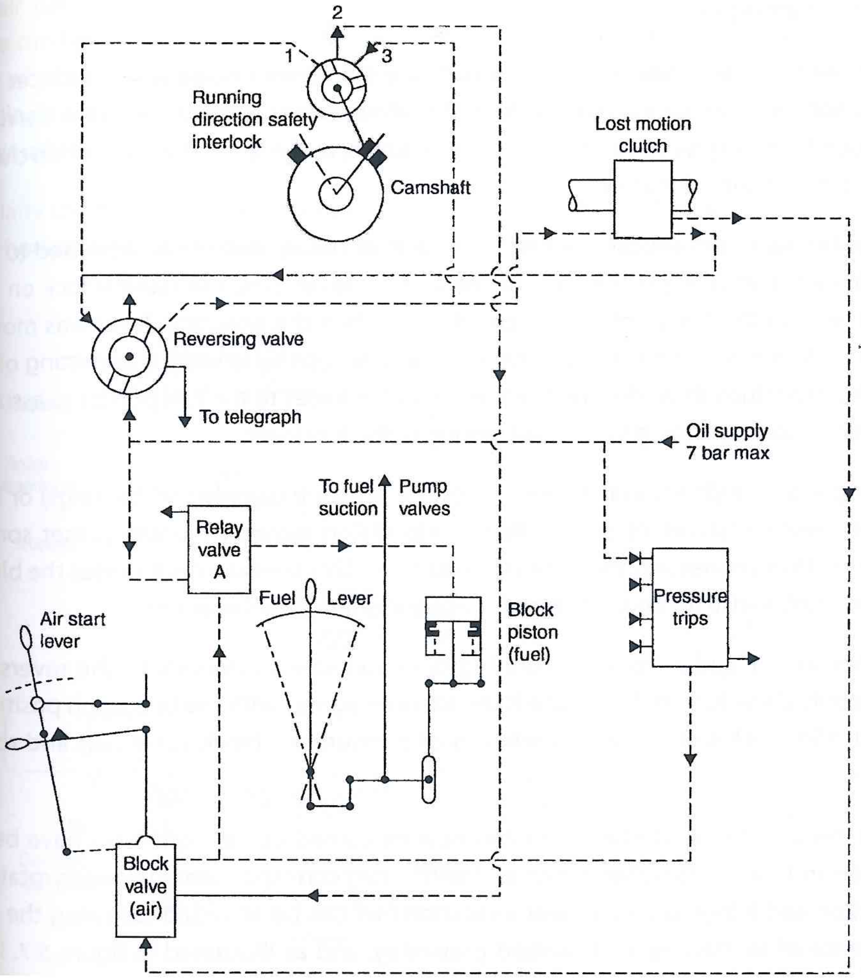
Consider figure 5.7. Air from the starting receiver at 30 bar maximum flows to the pre-starting valve (via the open turning gear blocking valve shown), and directly to the automatic valve. At the automatic valve air passes through the small drilled passage to the back of the piston and this together 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 and 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 in figure 5.8.



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 31/2, the starting lever is mechanically blocked by the linkage shown in figure 5.7.

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) thus 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 in figure 5.8. 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) thus 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 thus connecting the oil pressure 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 relieved of pressure via block valve (air) and relay valve A.

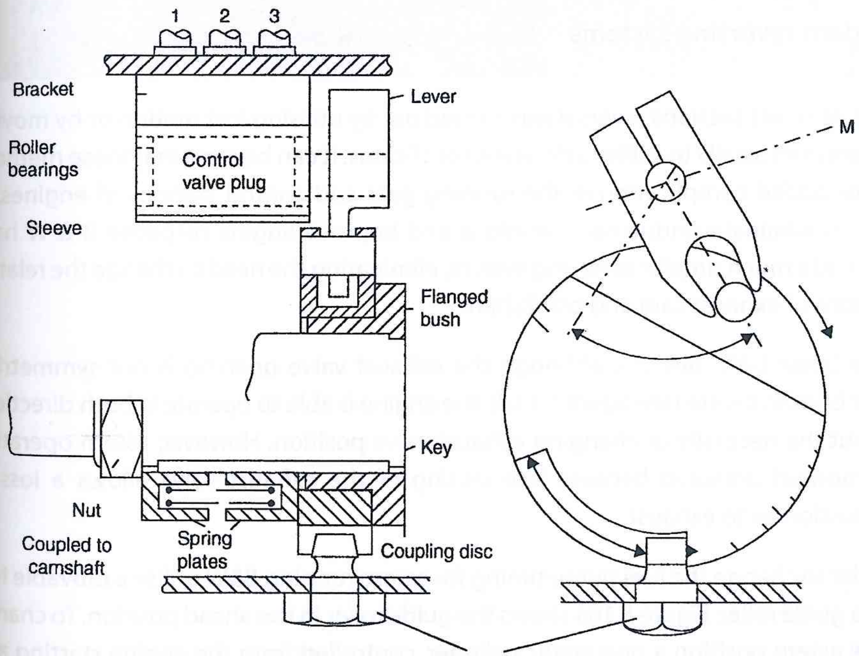
Movement of the air starting lever can now be carried out as both locks have been cleared and subject to no trip action and satisfactory correspondence between rotation direction and telegraph reply lever indication fuel can be admitted following the full sequence of air starting as described previously, and as illustrated in figure 5.7. It is

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 figures 5.8 and 5.9. 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 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 figure 5.9.

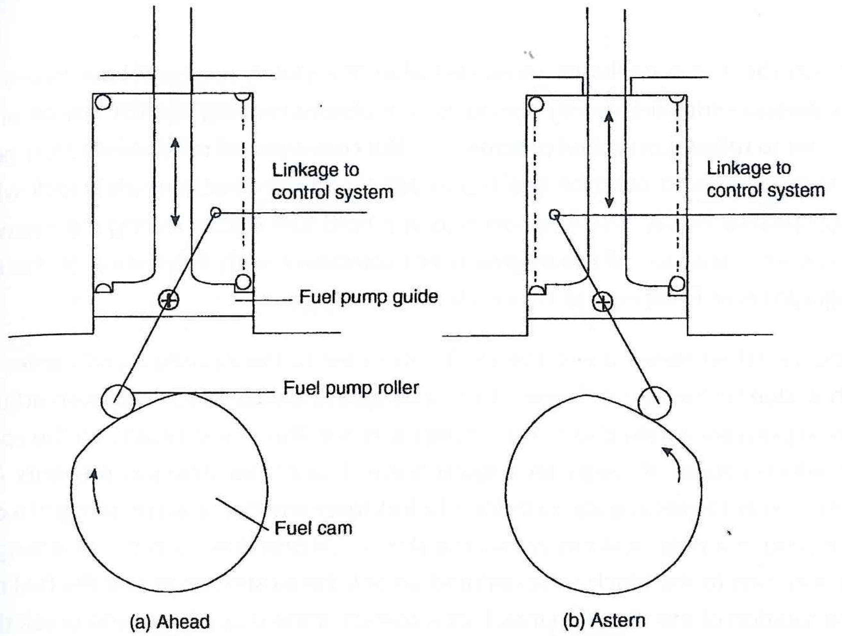
At the camshaft forward end the shaft is coupled to the camshaft and carries round with it, due to the key, a flanged bush and spring plates which cause an adjustable friction pressure axially due to the springs and nut. This pressure acts on the coupling disc which rotates through an angular travel 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 figure 1.13d), 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. Figure 5.10a shows the guide roller in the ahead position. To change to the astern position a pneumatic cylinder, controlled from the engine starting and



▲ **Figure 5.10** *Reversing mechanism of modern MAN Turbo Diesel B&W engines*