CONTROL

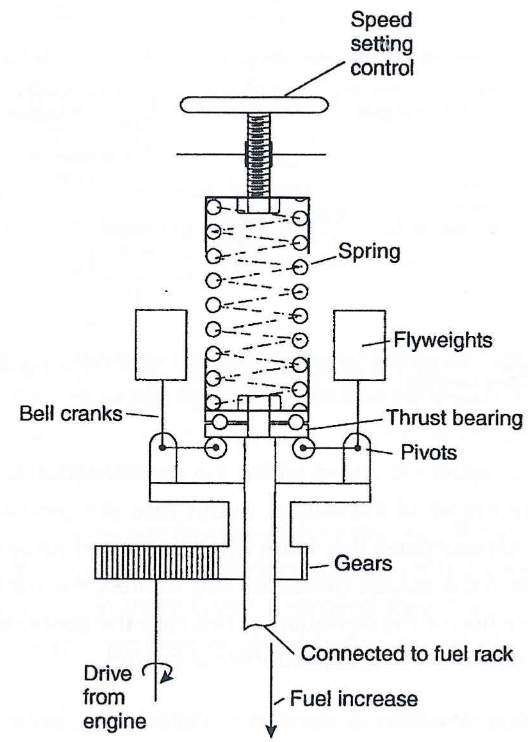
The study and application of instrument and control devices has developed from the beginning of engineering itself. It is however in recent years that this branch of engineering has assumed greater importance and with the advent of programmable logic controllers (PLC) the science is about to take the industry by storm as we will see.

Automatic control in a simple sense has always been utilised to try and ensure the safety of personnel, for example, cylinder relief valves, speed governors, overspeed trips, etc. It is intended in this chapter to examine the control of the modern diesel engine and its associated equipment and to apply control terminology, with explanations, where required. The subject as a whole is covered in chapter 11 of Volume 8 and in greater depth still in Volume 10.

Governing of Marine Diesel Engines

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

In the past diesels driving electrical generators invariably utilised a mechanical



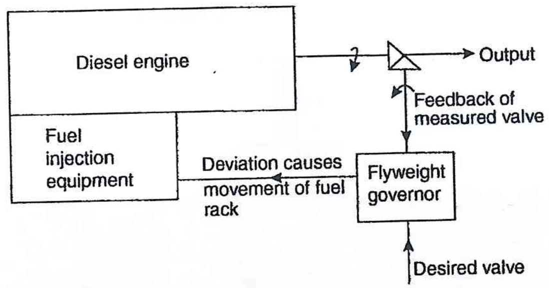
▲ **Figure 6.1** *Mechanical governor*

they invariably were fitted with an overspeed trip, usually of the Aspinal inertia type. This trip was arranged to allow full energy supply under normal operating conditions but in the event of revolutions rising about 5% above normal the energy was totally shut off until revolutions dropped to normal again. At about 15% above normal revolutions the trip would stay locked, with energy shut off, and this would continue until reset by hand.

A plain flyweight governor – so-called because the weights fly out under centrifugal force caused by their rotation – has to perform two separate functions which are to

1. act as a speed measuring device and
2. supply the necessary power to move the fuel-controlled system.

Figure 6.2 shows, in block diagram form, the arrangement of such a centrifugal or flyweight mechanical governor. An open loop system is one where there is no signal, representing the value of the output condition, being fed back to act as a comparison

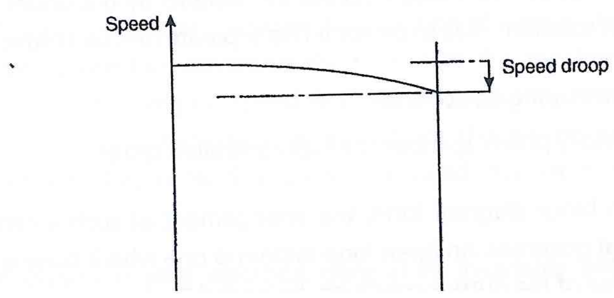


▲ **Figure 6.2** *Closed loop control*

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

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

When considering governors on diesel engines we have to think about the use of the term speed droop (figure 6.3) or just droop. It is used to define the change in speed between no load and full load conditions. If speed droop did not happen and the



governor was continually trying to keep the engine at one speed then due to the delay and mechanical inertia the governor would over-compensate for a given change in load. In this case the diesel would hunt while the governor constantly made changes to keep the steady running condition.

A set of circumstances without droop is known as an isochronous condition and an engine fitted with an isochronous governor will hunt usually over-compensating more and more until the over-speed trip activates. However, the term isochronous has taken a new meaning as we will find out later.

The forces involved in determining the mechanical governor's movement are, inertia, friction and spring. Considerable effort may therefore be required to cause movement one way or the other and this would result in a change of speed without any alteration of the governor's control position.

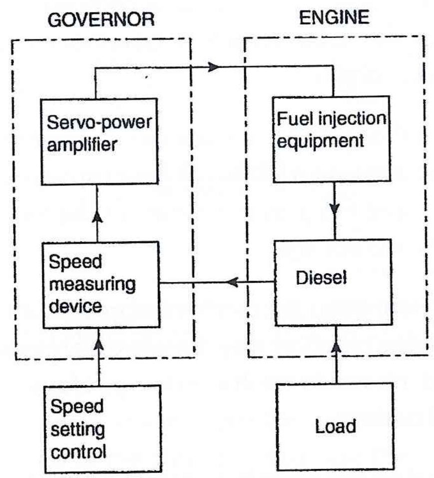
The control is at best very slow and insensitive as various equilibrium speeds are possible. For simple systems these various equilibrium speeds are not an embarrassment, but if we require a system to be more finely controlled then the two functions that the mechanical governor has to perform would be better separated into

1. speed measurement and
2. servo-power amplification.

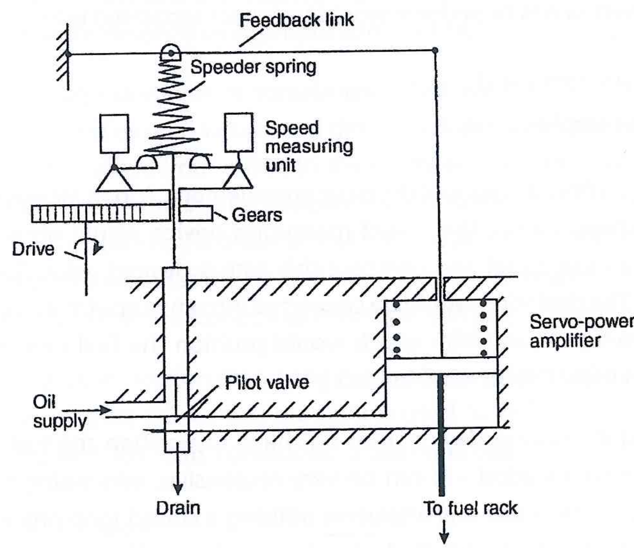
Figure 6.4 shows a flow diagram of the basic arrangement. A load increase would cause a momentary speed droop. The speed measuring device would obtain a measured value signal from the diesel and compare this with a desired value from the speed-setting control. The deviation would be converted into an output that would bring into action the servo-power amplifier which would position the fuel rack, increasing the supply of fuel to meet the increase in load.

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

A proportional action governor is diagrammatically shown in figure 6.5. The centrifugal speed measuring unit is fitted with a conically shaped spring, unlike that shown in figure 6.1; this gives a spring rate which varies as the square of the speed (figure 6.6) and gives a linear movement to the speed measuring system, that is, the response is directly proportional to the change in speed. If we consider an increase in load on

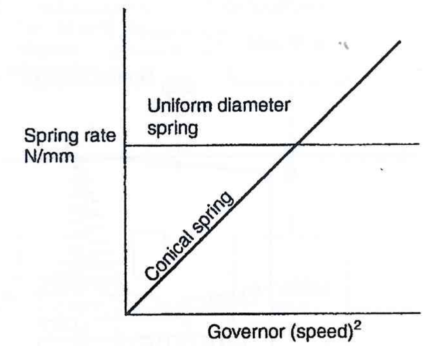


▲ **Figure 6.4** *Basic arrangement*



▲ **Figure 6.5** *Proportional action governor*

servo-amplifier will move up and increase the fuel supply to the engine. The feedback link reduces the force in the speeder spring so that the flyweights can move outwards to a new position, thus raising the pilot valve and closing off the oil supply. If for some reason the oil supply system should fail then the spring-loaded piston in the servo-



▲ **Figure 6.6** *Governor (speed)*

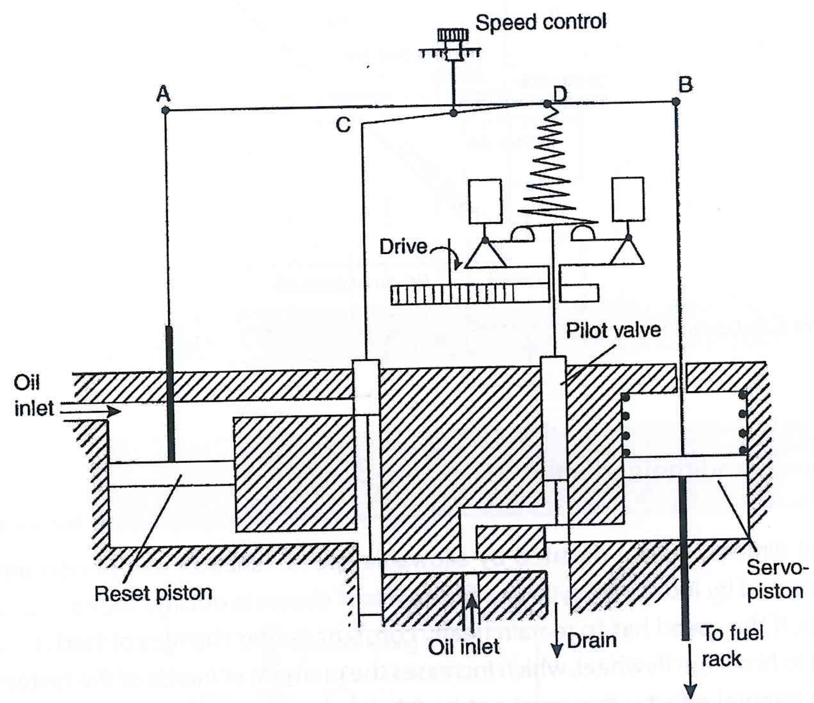
Flywheels and their effect

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

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

Governor with proportional and reset action

Figure 6.7 shows diagrammatically the type of governor that will, after an alteration in engine load, return the speed of the engine back to the value it was operating at before the alteration. If an increase in engine load is considered, the flyweights will move radially inwards and the pilot valve will open to admit oil to the servo-piston. The servo-piston will move up the cylinder compressing the spring and at the same time it will cause (a) the fuel rack to be repositioned to increase fuel supply to the engine, (b) rotate the feedback link 'A–B' anti-clockwise about the pivot point 'A' (this point 'A'

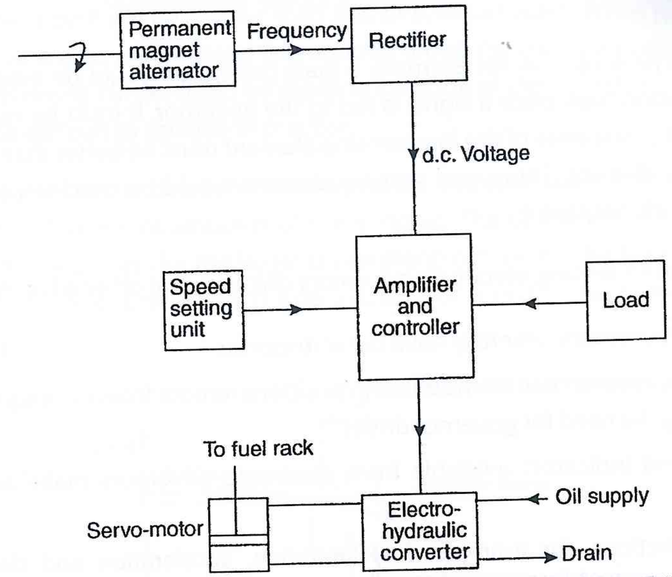


▲ **Figure 6.7** *Governor with proportional and reset action*

(c) rotate link 'C–D' will move the reset piston control valve down and some oil will drain from the reset piston cylinder. As the reset piston moves down to a new equilibrium position the feedback link 'A–B' will pivot about 'B' and the link 'C–D' will be rotated clockwise, closing the drain from the reset piston cylinder (and thus locking the reset piston in a new position), returning the point 'D' to its original position. This means that the engine is now running at its original speed but with increased fuel supply. Speed droop that took place during the change of the relative positions of the two pistons was transient. This type of governor, that has proportional and reset action, is called in governor parlance as an 'isochronous governor'.

Electric governor (figure 6.8)

Here we are still describing an old style governor that happens to be electric,



▲ **Figure 6.8** *Electric governor*

that there will be no slip rings or brushes with their attendant wear. The speed signal taken from the frequency of the generated a.c. voltage is converted into a d.c. voltage which is proportional to the speed. A reference d.c. voltage of opposite polarity, which is representative of the desired operating speed, is fed into the controller from the speed setting unit. These two voltages are connected to the input of an electric amplifier. If the two voltages are equal and opposite, they cancel each other out and there will be no change in amplifier voltage output. If they are different, then the amplifier will send a signal through the controller to the electro-hydraulic converter which will in turn, via the servo-motor, reposition the fuel rack. In order that the system is isochronous the amplifier controller has internal feedback.

Load sensing

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

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

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

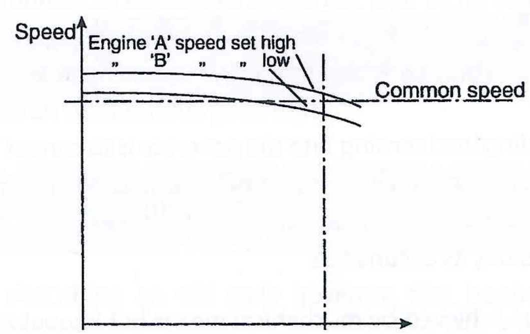
1. Electronic governors generally have faster response.
2. Electronic governors can be mounted in positions remote from the engine thereby eliminating the need for governor drives.
3. Controls and indicators available from electronic governors make automation easier.
4. Control functions, for example,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

The move towards PLC and full electronic control is covered more fully under chapter 11 of Volume 8.

Geared diesels

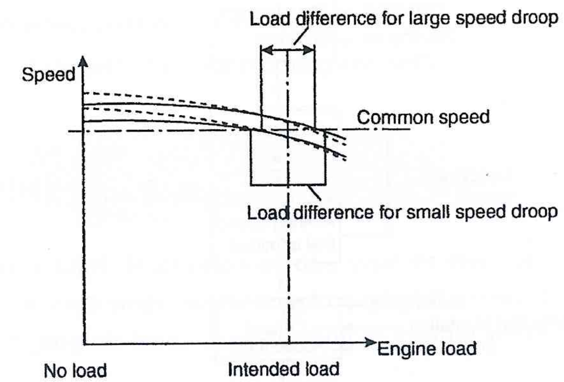
Two diesels geared together must run at the same speed, but if the governors of the two are not set equally then they will not carry equal shares of the load.

Figure 6.9 shows the governor droop curves for two diesels A and B. Governor A has a higher speed setting than that of B, but since they must both run at a common speed



the load carried by A will be greater than that of B. Actual load carried is given by the intersection of the common speed line and the droop curves. By adjusting the speed settings both droop curves could be made to coincide at the intended load, although this would be difficult to achieve in practice.

Shown in figure 6.10 are two sets of droop curves with the same difference in speed settings but with different amounts of speed droop. The difference in load sharing at the common speed is less for the larger speed droop curves than for the smaller. Hence speed droop and fine control over the desired level of speed are necessary for effective load sharing.

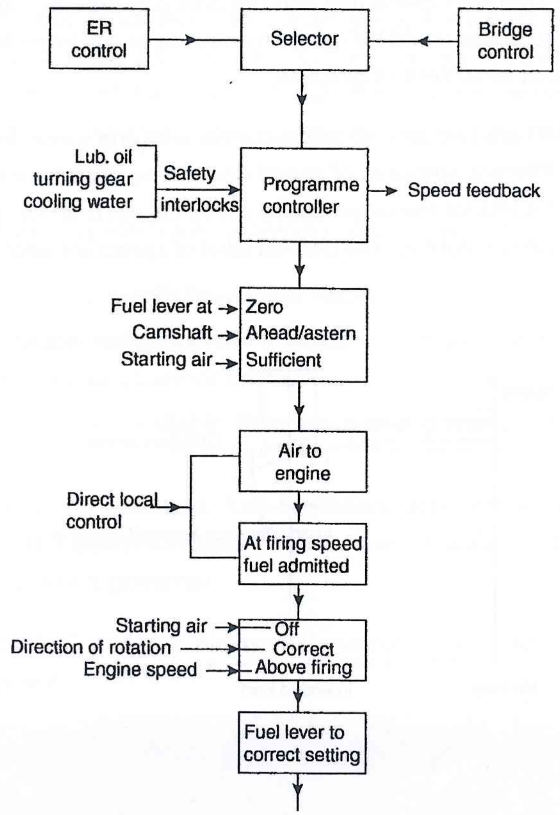


▲ **Figure 6.10** *Load difference*

Bridge Control of Direct Drive Diesel Engine

Two consoles would be provided, one on the bridge and the other in the engine room. For the bridge console the minimum possible alarms and instruments would be provided commensurate with safety and information requirements, for example, low starting air pressure and temperature, sufficient fuel oil, fuel oil pressure and temperature, etc. The engine room console would give comprehensive coverage and overriding control over that of the bridge.

In figure 6.11, for simplification, all normal protective devices are assumed and subsidiary control loops are not considered. The selector would be in the engine room console and the operator can select either engine room or bridge control; with one selected the other is inoperative. Assuming bridge control a programme would be



▲ **Figure 6.11** *Engine control programme*

turning gear in, etc. are satisfied, the programme can be initiated and could follow a sequence of checks and operations such as:

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

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

Further protective considerations are as follows:

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

Outline description

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

Movement of the telegraph lever actuates a variable transformer thus giving signals to the engine room electronic controller which transmits, in the correct sequence, a signal series to operate solenoid valves at the engine. One set of solenoid valves controls starting air to the engine while a second set regulates fuel supply, the latter via the manual fuel admission lever, is coupled to a pneumatic cylinder whose speed of travel is governed by an integral hydraulic cylinder in which rate of oil displacement is governed by flow regulators. This cylinder also actuates a variable transformer giving a reset signal when fuel lever position matches telegraph setting.

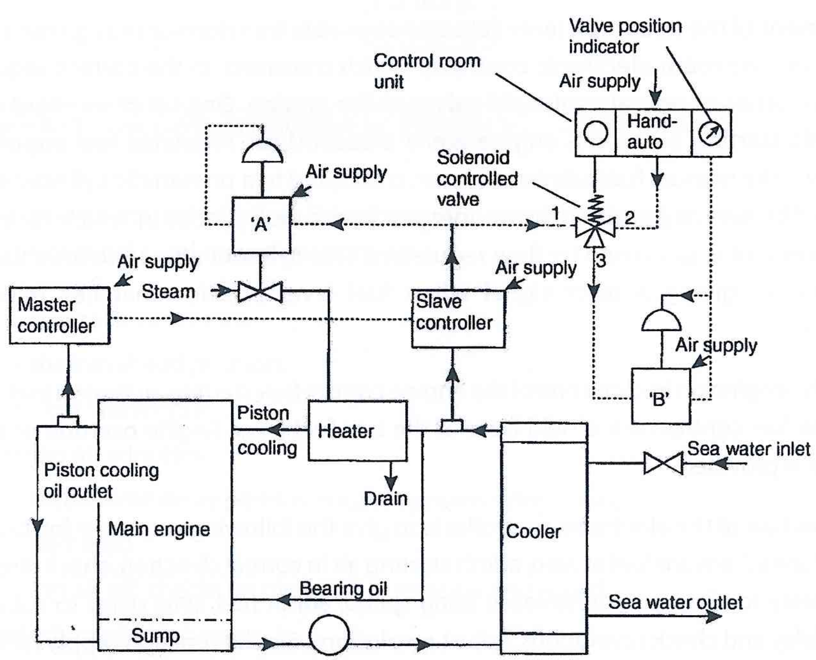
With the engine on bridge control the engine control box starting air lever is ineffective and the fuel control rack is held clear of the box fuel lever. Engine override of bridge control is provided.

The function of the electronic controller is to give the following sequence for, say, start to half ahead: ensure fuel at zero, admit starting air in correct direction, check direction, time delay to allow engine to reach firing speed, admit fuel, time delay to cut off air, time delay and check revolutions, adjust revolutions. Similar functions apply for astern stop to full between adjustable time limits of ½ min and 6 min. Fault and alarm circuits and protection are built into the system.

Piston Cooling and Lubricating Oil Control

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

For piston cooling and lubricating oil control the use of a cascade control system caters effectively for manoeuvring and steady-state conditions. Cascade control means that one controller (the master) is being used to adjust automatically as required the set value of another controller (the slave). In figure 6.12, the two main variables to consider



are sea water inlet temperature and engine thermal load. For simplicity we can consider each variable separately:

1. Assuming the engine thermal load is constant and the seawater temperature varies. The slave controller senses the change in lubricating oil outlet temperature from the cooler and compares this with its set value. It then sends a signal to the valve positioner 'B' to alter the seawater flow.
2. Assuming the seawater temperature is constant and the engine thermal load falls. The master controller senses a fall in piston cooling oil outlet temperature and compares this with its set value. It then sends a signal to the valve positioner 'B' so that the salt water flow will be reduced and the lubicating oil temperature at inlet to the piston is increased.

If the engine thermal load is low or zero then valve positioner 'A' will receive a signal from the slave controller which will cause steam to be supplied to the lubricating oil heater. This means that the slave control is split between valve positioners 'A' and 'B' – this is called 'split range control' or 'split level control'.

Slave controller output range is 1.2–2.0 bar.

Valve positioner 'A' works on the range 1.2–1.4 bar.

Valve positioner 'B' works on the range 1.4–2.0 bar.

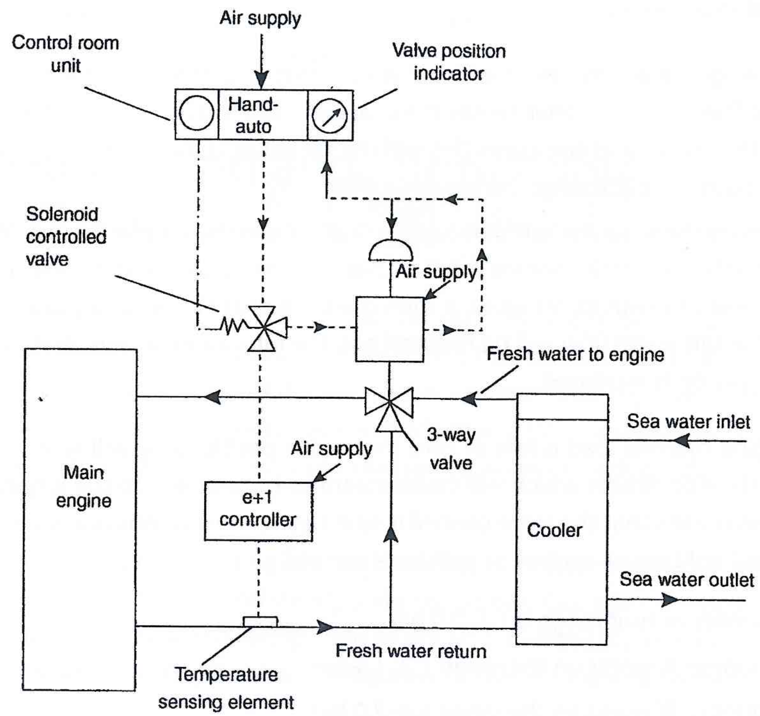
Hence the range is split in the ratio 1.3.

Since the piston cooling oil outlet temperature could be offset from the desired value by 8°C upwards or more, the master controller must give proportional and reset action. In order to limit the variety of spares that must be carried the slave controller would be identical to the master controller (figure 6.13).

It may be necessary to change over from automatic to remote control. This is achieved by position control of the three-way solenoid operated valve and regulation of the air supply to the valve positioner 'B' at the control room unit. The solenoid operated valve would be positioned to communicate air lines 2 and 3, closing off 1.

Hand regulation of the supply air pressure to valve positioner 'B' enables the operator to control the seawater flow to the cooler. Position of the seawater inlet control valve is fed back to control room unit. Lubricating oil temperatures would be indicated on the console in the control room.

An alternative and often preferred arrangement, using a single measuring element, is to have full flow of seawater through the cooler and operate a three-way valve (two inlets,

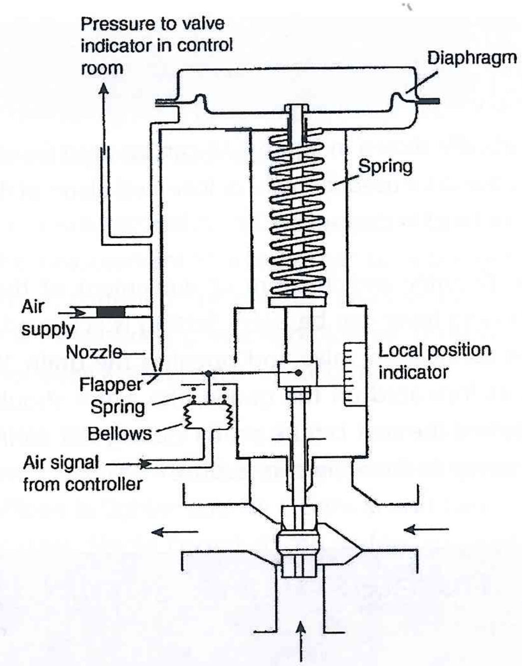


▲ **Figure 6.13** *Jacket (or piston) temperature control*

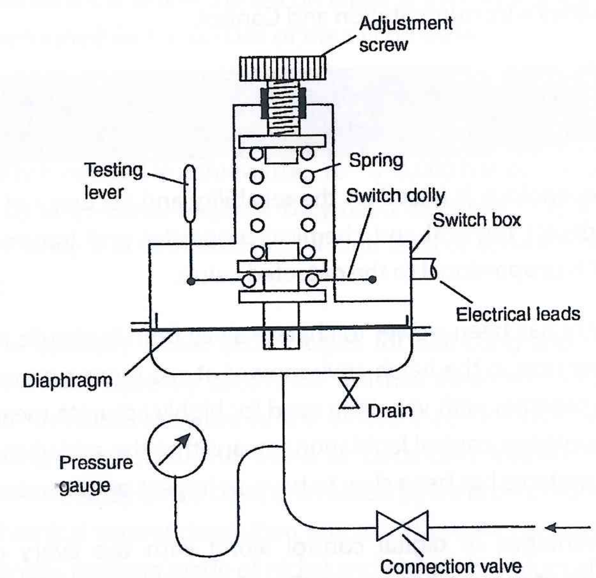
Valve selection for such duties is most important. Maximum pressure and temperature, maximum and minimum flow rate, valve and line pressure drops, etc., must be carefully assessed so that valve selection gives the best results. With correct analysis of the plant parameters and careful valve selection, simple single-element control systems can be employed. This would avoid the extra cost of sophisticated control loops and their attendant, increased maintenance and fall in reliability.

For mixing and by-pass operations a three-way automatically controlled valve with two inlets and one outlet of the type shown diagrammatically in figure 6.14 could be used. An increase in controller output pressure *p* causes the flapper to reduce outflow of air from the nozzle; the pressure on the underside of the diaphragm increases and the valve moves up. As the valve moves, the flapper will be moved to increase outflow of air from the nozzle and eventually the valve will come to rest in a new equilibrium position.

Indication of valve position is given locally and remote, in the latter case by feeding back the diaphragm loading pressure to an indicator possibly situated in the control



▲ **Figure 6.14** *Three-way valve and positioner*



Pressure alarm

The alarm diagrammatically shown in figure 6.15 can be used for either high- or low-pressure warning. It can also be used for high- or low-level alarm of fluids in tanks since pressure is a function of head in the tank.

To test the electrical circuitry and freedom of movement of the diaphragm and switches, the hand testing lever can be used. Setting is achieved, for low-pressure alarm, by closing the connection valve and opening the drain. When the desired pressure is reached, as indicated on the gauge, the alarm should sound. If high-pressure alarm is required the unit can be set by closing the connection valve and coupling a hydraulic pump to the drain connection.

Electrical and Electronic Control Systems

This section is a brief overview of the newly emerging technology that is due to take the industry by storm and students should study this is in parallel with chapter 11 in Volume 8 of this series – Instrumentation and Control.

Sensors

Modern control technology is based on the reliability and accuracy of sensors. These are devices that detect physical and chemical properties and transmit an electrical output signal that is proportional to the detected value.

The marine industry has been slower to change away from hydraulic and pneumatic control systems because in the harsh environment at sea these systems have proved most reliable. This together with very little need for highly accurate measurement (due to lack of engine emission control legislation) means that the added sophistication of electronic control systems has been slow to have an impact on the industry.

However, the advantages of digital control along with the every growing need for reducing engine emissions has meant that there is now a rush to embrace the

- Substrate and hybrid technology for pressure and temperature sensors
- Semi-conductor technology used for monitoring rotational speed
- Micromechanics for measuring pressure and acceleration
- Micro-optics for measuring light, for example, boiler alight confirmation
- Mechatronics (described in the next section).

The advancement in microelectronics and micromechanics is the driving force behind the changes and the measurement of pressure has been one of the problematic areas for the advancement of control. Strain gauges can measure small movements which make them useful for measuring pressure or torque. However, until recently these instruments have not been available in a micro format.

The micromechanical structure – strength and hardness, etc. – of silicon can be compared with steel. However, the other properties of silicon opens up a whole new technology. The silicon is lighter and its electrical and thermal conductivities are quite different for steel. Single crystal silicon wafers are used because they have almost perfect physical response characteristics. Hysteresis and creep are negligible and the single crystal material is very brittle and ruptures if the elastic limit is exceeded.

Bulk micromechanics is where the whole wafer is built up using etching techniques and can be used to produce diaphragms of between 5 and 50 μm for measuring pressure. Surface micromechanics is where the silicon wafer is used as a substrate and moving structures can be formed on the surface of the silicon wafer.

Measuring the high pressures generated in the combustion space and the fuel rail of a diesel engine requires high quality cost-effective sensors. A robust design with long-term stability is needed to achieve the, 10^{10} 0–2,000 bar, operational lifetime load cycles required by large diesel engines. They need electronics capable of processing measured data quickly and demonstrating a high level of accuracy over a wide range of temperatures.

There are several different sensor technologies for detecting and transmitting high pressure but the most frequently used is the resistive measurement method because it references the measured pressure to the ambient conditions. This process makes use of two coating techniques. The first is to electrically insulate the active sensor components from the body of the device, achieved by a silicon dioxide layer being laid down using a chemical vapour deposition process. These are capable of withstanding 500V + AC. Secondly, resistors made of nickel and nickel chromium alloy are sputtered directly onto the silicon dioxide surface to form a Wheatstone bridge measurement circuit. In subsequent steps conventional lithographic techniques are used to create the meander-shaped conductor paths typically used in strain gauges.

The nickel-chromium alloy used to fabricate thin film devices has the additional advantages of having a very high manufacturing reproducibility and temperature stability up to 200°C. The latest high-pressure sensors utilise a non-welded design based on a monolithic body of precipitation-hardened 17–4PH nickel-chromium stainless steel. The sensor diaphragm, which must be of an exactly defined thickness, forms the bottom of a highly accurate deep bore machined into the centre of the device body, where the membrane is designed to operate at pressures in excess of 3,000 bar. The thin film structure is not directly located on the diaphragm but is deposited onto a tiny cantilever beam attached to it at the thinnest point, removing the inefficient need to create the sensor structures on diaphragms individually. To maintain low stress levels in relation to the elastic limit which maximises the operational lifetime, a monolithic sensor body can be used instead of a welded structure. This offers enhanced security under fatigue stress conditions due to the seamless design and lower internal residual stress levels. A stable output signal is generated from the Wheatstone bridge, by the slight stretching of the diaphragm under pressure. However, the signal level is low, and with a *k*-factor of 2 for nickel–chromium strain gauges, outputs of 2 mV/V are measured. This disadvantage is reduced with the help of the latest application-specific integrated circuit (ASIC) technology. The circuit used in the sensor features an offset-free amplifier which operates in a closed control loop, thereby compensating for any offset drift. This allows even small output signals below 2 mV to be amplified and compensated with a high signal-to-noise ratio, at the 5 V supply voltage. Sensors can be tailored to meet specific customer requirements. Analogue to digital conversion can be performed by a sigma-delta converter and error compensation effected by means of a 3D lookup table – the error is evaluated and fed back via a fast pulse width modulated (PWM) signal. Temperature is measured either by the ASIC's on-chip sensor or directly by a nickel resistor on the strain gauge itself, the latter giving a good indication of the temperature of the pressure medium. The signal is measured with a bandwidth of 10 kHz, so that fast transient effects in the fuel injection process can be recorded. Electronics have meet marine electromagnetic compatibility requirements and state the total error shown by a sensor.

The rapid, high amplitude changes in pressure, as well as the easily recognised oscillations in the injector pressure tubing are reproduced without error or delay. In contrast to piezoelectric measurement techniques, the static pressure is measured relative to the ambient value. The non-welded high-pressure sensor design described

bandwidth permits optimisation of the fuel injection modulation process by the motor control system. By using the new sensors to monitor injector needle lift and cylinder pressure, the combustion process in each cylinder can be individually optimised, thereby allowing operators to reduce emissions, increase fuel efficiency and monitor engine health more closely.

Temperature sensors

The common technology employed in temperature sensors is the variation of electrical resistance in materials when the temperature of their surroundings, changes with either a positive or negative coefficient. The sensors predominantly rely on contact with the measured medium but some measure the infra-red radiation given off by a hot surface, for example. Some fire detectors work on this principle.

Sintered ceramic resistors made of heavy metal oxides are an example of semi-conductive materials which display an inverted exponential temperature curve. An example of an extreme precision sensor is a thin film metallic resistor integrated on a single substrate wafer. Temperature neutral 'trimming' resistors are used to give the high accuracy.

Force and torque sensors

Force sensors are either strain measuring or displacement measuring devices. The strain measuring devices might use the piezo-resistive of the magnetoelastic principle. The former being used to measure torque as well as force they are the most widespread and are very reliable. Piezo-resistive sensors work on the fact that there is a change in resistance in the material due to its deformation.

Magnetoelastic devices use the principle that ferromagnetic materials change their magnetic properties under elongation or reduction in length due to the force on the component. This change in magnetic properties is detected, measured and an output signal is generated.

Mechatronics

Mechatronics is the technology that is starting to transform the world of control has taken them to new levels of performance. This has largely been due to the developments in manufacturing which has enabled the production of reliable and robust sensors.

These sensors can detect pressure and temperature to a much finer tolerance and act infinitely faster than any mechanical system. The power of microprocessors and the development of the algorithms required to interpret the signals from thousands of sensors has brought everything together at a time when the industry needs a step forward in efficiency.

Mechatronics is the synergistic integration of mechanical components, electronic devices, computer and software engineering along with embedded control features, coming together to keep machinery operating as close to its design condition as possible. It has been described as 'mechanical engineering for the 21st century'.

During the past 30 years or so, as we have seen, the technology consisted of pneumatic control equipment where precision made mechanical instruments and other devices interpreted the condition of machinery and associated system and transmitted pneumatic signals in an effort to control processes such as the temperature of cooling water or lubricating oil as we have already seen. These control systems were usually localised at the point of use, although some had remote indicators such as that for boiler water level. However, on the whole, control systems were independent of the main machinery alarm systems and the measurement of processes inside hostile environments, such as the engine combustion space, were just not possible.

As discussed we are already seeing the electronic control of diesel engine combustion increasing engine flexibility and lowering emissions. Now, other marine machinery is also being produced with the ability to link into the central control and management system, and as algorithms become ever more sophisticated so will the efficient use of the equipment.

All communication systems need a common language and the focus for the development of mechatronics on-board is now on the topology and protocols being used with modbus and canbus being the top runners. However, some manufacturers are producing equipment that uses the protocol called profibus.

The developments of these standards into a unified system or one protocol is important so that manufacturers can move forward rather than having to cover different systems or having to change standards half way through their production run, which is a possibility at present.

evidence they require ensuring their vessels are complying with legislation. In addition, such information will enable managers to increase efficiency thus decreasing fuel consumption and maximising profits. The capability of mechatronics to assist the industry with its move towards higher efficiencies, lower energy consumption and more ecofriendly vessels is on the horizon. Given the right development, more sophisticated and so-called intelligent systems will be capable of linking the control of machinery with the production of machinery condition data. This will drive efficient management systems enabling ship and head office staff to work together by improving not only the vessel's voyage planning but also the efficient use of machinery. Progression will increasingly mean different types of fuel and/or propulsion types and will also include more efficient maintenance systems and changes in the general management of vessels. The integration of embedded systems into the control of marine machinery will once again bring to the fore the importance of crew competence and focused staff development.

We now have in place the 'Manila amendments' to the STCW convention. Specific outcomes from this update require a focus on the operation of pollution prevention equipment and, more generally, additional emphasis being given to environment management. There will inevitably be a concentration on professional development including diagnostic techniques for ship staff since they will be in the front line in the event of any emergency, for example, if the alarm monitoring system is indicating that the ship is not producing the correct exhaust emissions as the vessel is approaching a SECA area. This issue would clearly have to be resolved urgently so the vessel does not risk incurring a hefty fine. Integrated approach to design has potential for monitor efficiency gains. With sophisticated control systems in place, crew familiarisation will be essential, therefore continuity of staff, efficient handovers and the use of integrated management information systems will be required in running the modern fleet effectively.

Communication systems

The increased use of sensors and electronic control systems will undoubtedly bring a step change in both the accuracy and the amount of information that will be available to the engineers in charge of a marine power plant. The sensors are linked to PLC, microprocessors or full computer systems. However, as with any communication system they all have to be speaking the same language and using the same transportation system. A computer-controlled system employing a universal bus system instead of individually wired circuits is the modern trend for control

Compared to a control system with conventional wiring a computer-controlled bus system has the following advantages:

- Cost, weight and construction savings due to the reduction in wiring (valves, pumps and actuators, etc. are operated by data signals travelling along one communication bus instead of electrical signals travelling down individual circuits).
- Greater redundancy and operational reliability due to a much lower number of plug-in connectors and easy use of multiple transmission paths.
- Much simpler and reduced construction time.
- Reduced commissioning time due to much simpler connecting procedures.
- Multiple use of the sensor signals.
- Simple upgrade procedures.

One of the features of this system that promises to deliver so much is the production of real-time data. Machinery control, performance monitoring and maintenance systems can all be updated with real-time data. This will have substantial benefits for the efficiency of ship's machinery and for energy management. Remote monitoring will be made so much easier and therefore there will be knock on effects for management systems, training and professional development of engineering officers.

Unmanned Machinery Spaces (UMS)

In the past a UMS designated space may have been known under other names such as 'unattended' machinery space; however, IMO has chosen the term 'unmanned' and this is the terminology used in the STCW document as well as 'M' notices in the United Kingdom and by all participating flag states.

Vessels that have a machinery space designated as unmanned does not mean that the engineroom is unattended or unsupervised, in fact quite the opposite is the case. STCW actually describes such an engineroom as 'periodically unmanned engine-room' which is the key description to the system.

The watchkeeping engineering officer is just as much supervising, controlling, monitoring and working closely with the machinery as she/he would in a fully manned engineroom. The difference is that some of the routine monitoring is undertaken by

routine adjustments to the machinery systems and his/her time is freed up to carry out other tasks. It also means of course that if the watchkeeper completes all the checks and duties that the machinery should operate for up to 8 h without the watchkeeper being physically present in the engineroom.

The watchkeeper is however still on-duty and responsible for the supervision of the machinery space but to be effective she/he can monitor the alarm system remotely from the vessels accommodation block. In the event of a machinery alarm the watchkeeper must be present in the machinery control room (MCR) and respond to the machinery alarm within 90 s of the alarm first sounding.

All modern ships are built with a sophisticated alarm and monitoring control system and they will all have the ability to run with UMS; however, not all will be operated in this way. Passenger ships for example have 15,000 alarm points but they still have an engineering watchkeeper in the MCR at all times due to the reassuring message that this sends to the customers.

Cargo ships on ocean passage will be able to operate the machinery space unmanned if the vessel has the appropriate certificate to do so. To get the approval, the vessel must have the following:

1. *Bridge control of propulsion machinery:* The bridge watchkeeper must be able to take engine control action in the event of a vessel emergency such as a navigational manoeuvre. Control and instrumentation must be as simple as possible for the bridge watchkeeper to use.
2. *Centralised control and instruments are required in machinery space:* Engineers may be called to the machinery space to answer a routine alarm or in the case of an emergency and controls must be easily reached and fully comprehensive.
3. *Automatic fire detection system:* Alarm and detection system must operate very rapidly. Numerous well-sited and quick response detectors (sensors) must be fitted.
4. *Fire extinguishing system:* In addition to conventional hand extinguishers a control fire station remote from the machinery space is essential. The station must give control of emergency pumps, generators, valves, ventilators, extinguishing media, etc.
5. *Alarm system:* A comprehensive machinery alarm system must be provided and repeater stations must be available in the accommodation areas.
6. *Automatic bilge high-level fluid alarms and pumping units:* Sensing devices in bilges with alarms and hand or automatic pump cut-in devices must be provided.

7. *Automatic start emergency generator:* Emergency generators must be situated outside the machinery space and connected to separate emergency bus bars. The primary function is to give protection from electrical blackout conditions.
8. *Local hand control of essential machinery.*
9. *Adequate settling tank storage capacity:* The watchkeeper will have to ensure that the engine has enough fuel to operate for the 8 h that the engineroom will be unmanned. If this is not done then the low-level alarms will sound and the watchkeeper will have to complete the task probably at a time that disturbs his/her rest.
10. *Regular testing and maintenance of instrumentation.*

7 ANCILLARY SUPPLY SYSTEMS

Compressed Air

Compressed air is used for starting main and auxiliary diesels, operating whistles or typhons, testing pipe lines (e.g. CO_2 fire extinguishing system) and for workshop services. The latter could include pneumatic tools and cleaning lances as well as other hand tools and specialist tools such as engine exhaust valve or seat grinding wheels. The high pressure compressed air for the starting of diesel engines will usually be stored in two large air receivers at around 30 kg/cm² (bar). The compressors will be low-volume, high-pressure machines usually water cooled. Classification societies require that the outlet temperature of the compressed air be kept 98°C due to the risk of air start line fires. Class also require that at least two compressors are fitted to a vessel and that one must be propelled by an alternative power source (such as a diesel engine). Each compressor must have sufficient capacity to charge the starting air receiver from atmospheric pressure to full pressure in 1 h. The air receivers must have sufficient capacity to provide a minimum of 12 starts for a reversible engine and 6 starts for a non-reversible engine. It is important that air compressors do not carry over oil into the compressed air lines. If a fault develops in any of the air start valves then hot gasses could start a serious fire or explosion in the air start line if oil is present. Due to the air being compressed there would be more oxygen than usual and the result could be violent. It is considered good watchkeeping practice to shut the isolating valves from the main air receivers while the engine is running to reduce the compressed air in the lines from the receivers to the

The compressed air used for powering hand tools such as rotary wire brushes or needle guns is at a much lower pressure than the starting air, but the tools will require a considerable volume of air for them to operate properly and if there are two or three tools in operation at the same time then the air compressors will be working hard to keep up. The working air compressors operate at about 7 kg/cm^2 (bar) and are able to produce a high volume of air. The compressed air required for control engineering needs careful consideration as the air needs to be dry and carefully controlled if delicate controls and instruments are to work correctly. Instrument air compressors can be of the screw type which may also be described as oil-injected or oil-free compressors. The compressors up to about 30 kW are generally air cooled and above 30 kW, fresh water cooling is available. The instrument air system must be free from both oil and water contamination for the instruments and controls to work properly which is why the special 'oil-free' compressors are available. Instrument air can also be provided via a reducer/dryer combination fitted to the main air or working air system.

Air is composed of mainly 23% oxygen, 77% nitrogen by mass and since these are near perfect gases a mixture of them will behave as a near perfect gas, following Boyle's and Charle's laws (see Reeds series, Volume 3). When air is compressed its temperature and pressure will increase as its volume is reduced. There are several theoretical models for this process which are as follows.

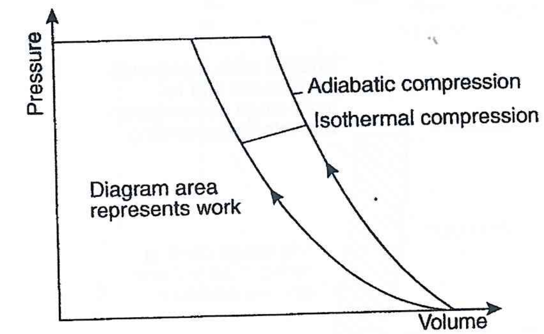
Isothermal compression

Isothermal compression of a gas is compression at constant temperature. This would mean in practice that as the gas is compressed heat would have to be taken from the gas at the same rate as it is being received. This would necessitate a very slow moving piston in a well-cooled small bore cylinder which is not practical for an actual design.

Adiabatic compression

Adiabatic compression of a gas is compression under constant enthalpy conditions, that is, no heat is given to or taken from the gas through the cylinder walls and all the work done in compressing the gas is stored within it. Again this is not easy to build as a practical solution.

In figure 7.1, the two compression curves show that there is extra work done by



▲ **Figure 7.1** *A comparison of compression processes*

compressor were slow running with a small bore perfectly cooled cylinder and a long stroke piston, the air delivery rate would be very low.

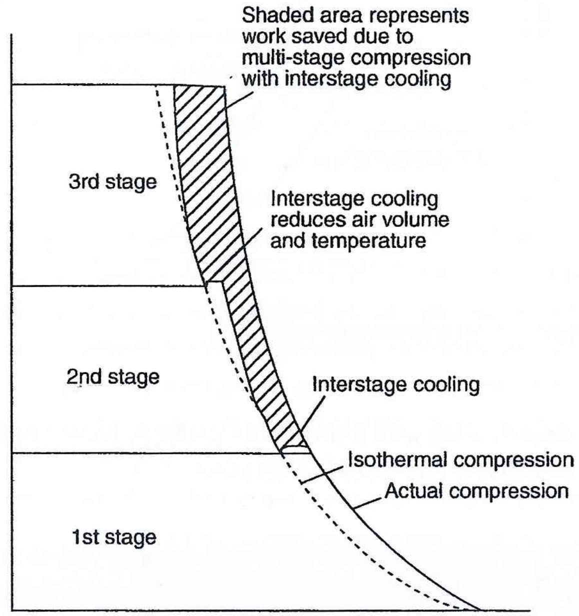
Multi-stage compression

If we had an infinite number of stages of compression with coolers in between each stage returning the air to ambient temperature, then we would be able to compress over the desired range close to isothermal conditions. However, this is still impracticable and therefore a two- or three-stage compression with inter-stage and cylinder cooling is generally used when relatively high pressures have to be reached.

Figure 7.2 shows the work saved by using this method of air compression, but even with efficient cylinder cooling the compression curve is nearer the adiabatic than the isothermal and the faster the delivery rate the more this will be the situation.

To prevent overheating and consequential damage, cylinders have to be water or air cooled and clearance must be provided between piston and cylinder head. This clearance must be as small as practicable.

High-pressure air remaining in the cylinder after compression and delivery will expand on the return stroke of the piston. This expanding air must fall to a pressure below that in the suction manifold before a fresh air charge can be drawn in. Hence, part of the return or suction stroke of the piston is non-effective. This non-effective part of the suction stroke must be kept as small as possible in order to keep capacity to a maximum. This clearance is sometimes referred to as the 'bump clearance'; see the section 'Measuring the air compressor clearance'.



▲ **Figure 7.2** *Three-stage compression*

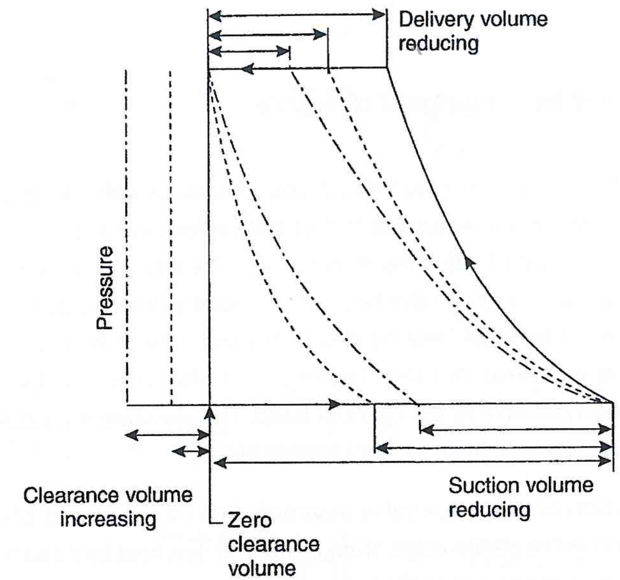
Volumetric efficiency

This is a measure of compressor capacity. It is the ratio of the actual volume of air drawn in each suction stroke to the stroke volume. Figure 7.3 shows what would happen to the compressor volumetric efficiency – and hence capacity – if the clearance volume were increased.

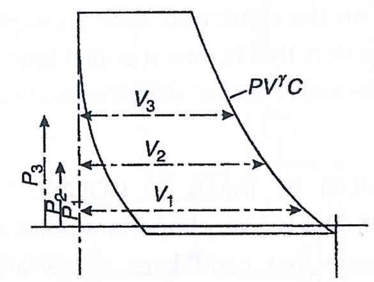
Clearance volume can be calculated from an indicator card by taking any three points on the compression curve such that their pressures are in geometric progression, that is, $P_1/P_2 = P_2/P_3$ hence $P_2 = \sqrt{P_1P_3}$ (figure 7.4). If V_c = clearance volume as a percentage of the readily calculable stroke volume and V_1, V_2, V_3 are also percentages of the stroke volume then:

$$P_1(V_1 + V_c)^n = P_2(V_2 + V_c)^n = P_3(V_3 + V_c)^n$$

$$\text{i.e.}\quad \frac{P_1}{P_2} = \left(\frac{V_2 + V_c}{V_1 + V_c}\right)^n \quad \text{and} \quad \frac{P_2}{P_3} = \left(\frac{V_3 + V_c}{V_2 + V_c}\right)^n$$



▲ **Figure 7.3** *Effects of increasing clearance volume*



▲ **Figure 7.4** *Calculating clearance volume*

$$\text{therefore}\quad \left(\frac{V_2 + V_c}{V_1 + V_c}\right)^n = \left(\frac{V_3 + V_c}{V_2 + V_c}\right)^n$$

$$\text{hence}\quad \frac{V_2 + V_c}{V_1 + V_c} = \frac{V_3 + V_c}{V_2 + V_c}$$

$$V_c = \frac{V_2^2 - V_1V_3}{V_1 + V_3 - 2V_2}.$$

Measuring the air compressor clearance

Correct clearance must be maintained and this is usually done by checking the mechanical clearance, between the top of the piston and the cylinder head, (called the bump clearance) and adjusting it as required by altering the height of the piston relative to the cylinder and cylinder head. This is usually done by using inserts under the palm of the connecting rod. Bearing clearances should also be kept at recommended values because any wear in these bearings will also alter the bump clearance by moving the piston relative to the cylinder head. Two possible methods of ascertaining the mechanical clearance in an air compressor are:

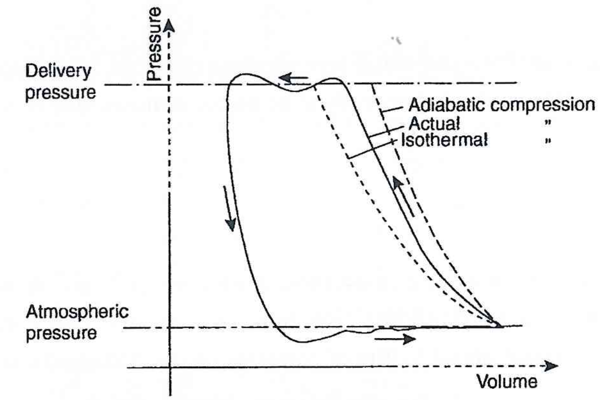
1. Remove suction or discharge valve assembly from the unit and place a small loose ball of lead wire on the piston edge, then rotate the flywheel by hand to take the piston over TDC. Remove and measure the thickness of the lead wire ball.

2. Put crank on TDC, slacken or remove bottom half of the bottom end bearing. Rig a clock gauge with one contact touching some underpart of the piston or piston assembly and the other on the crank web. Take a gauge reading. Then by using a suitable lever bump the piston, that is, raise it until it touches the cylinder cover. Take another gauge reading, the difference between the two readings gives the mechanical clearance.

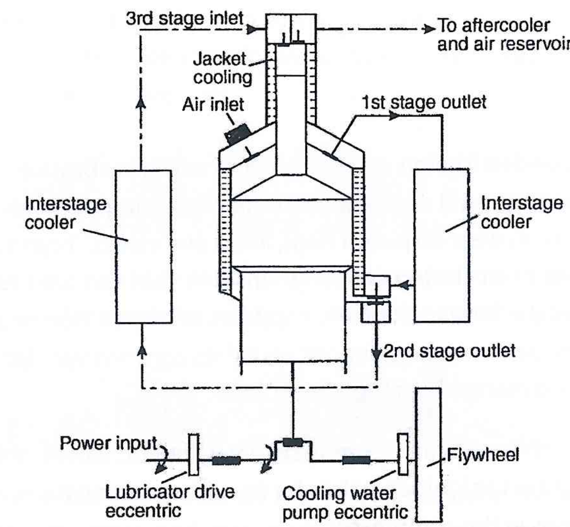
EXTRA SPECIAL CARE MUST BE TAKEN TO ISOLATE THE COMPRESSOR BEFORE UNDERTAKING THIS WORK. This is a very important point and the flag state examiner will be checking to ensure that candidates appreciate the importance of this procedure.

In practice the effective volume of air drawn in per stroke is further reduced by the pressure in the cylinder which on the suction stroke must fall sufficiently below the atmospheric pressure so that the inertia and spring force of the suction valve can be overcome and air under the force of atmospheric pressure will flow into the cylinder. Figure 7.5 shows this effect on the indicator card and also shows the excess pressure above the mean required upon delivery, to overcome delivery valve inertia and spring force and push the compressed air out of the cylinder.

Air compressors are either reciprocating or rotary types, the former are most commonly used at sea for the production of air for starting diesel engines or for driving power tools as outlined at the beginning of this chapter.



▲ **Figure 7.5** *Actual pressure–volume*



▲ **Figure 7.6** *Three-stage air compressor*

compressor, the pressures and temperatures at the various points would be roughly as follows:

	Delivery pressure	Air temperature	
		Before the coolers	After the coolers
First stage	4 bar	110°C	35°C
Second stage	16 bar	110°C	35°C

The above figures are for a salt water temperature of about 16°C. Final air temperature at exit from the after-cooler is generally at or below atmospheric temperature.

Drains

Fitted after each cooler is a drain valve, these are essential. To emphasise, if we consider 30 m³ of free air, relative humidity 75%, temperature 20°C being compressed every minute to about 10 bar, about ½ litre of water would be obtained each minute.

Drains and valves to air storage unit must be open upon starting up the compressor in order to get rid of accumulated moisture. When the compressor is running drains have to be opened and closed at regular intervals.

Filters

Air contains suspended foreign matter, much of which is abrasive. If this is allowed to enter the compressor it will combine with the lubricating oil to form an abrasive-like paste which increases wear on piston rings, liners and valves. It can adhere to the valves and prevent them from closing properly, which in turn can lead to higher discharge temperatures and the formation of what appears to be a carbon deposit on the valves, etc. Strictly, the apparent carbon deposit on valves contains very little carbon from the oil, it is mainly solid matter from the atmosphere.

These carbon-like deposits can become extremely hot on valves which are not closing correctly and could act as ignition points for air–oil vapour mixtures, leading to possible fires and explosions in the compressor.

Hence air filters are extremely important. They must be regularly cleaned and where necessary renewed and the compressor must never be run with the air intake filter removed.

Relieving devices

After each stage of compression a relief valve will normally be fitted. Regulations only require the fitting of a relieving device on the h.p. stage. Bursting discs or some other relieving device are fitted to the water side of coolers so that in the event of a

Lubrication

Certain factors govern the choice of lubricant for the cylinders of an air compressor, these are: operating temperature, cylinder pressures and air condition. The correct grade of oil must be used when topping up the air compressor. This will be detailed on the vessel's oil schedule which is kept in the MCR and will be produced by the oil manufacturers. To comply with the Control of Substances Hazardous to Health (COSHH) regulations the specification of the oil should also be available to the ship's crew.

Operating temperature

This affects oil viscosity and deposit formation. If the temperature is high this results in low oil viscosity, very easy oil distribution, low film strength, poor sealing and increased wear. If the temperature is low, oil viscosity would be high. This causes poor distribution, increased fluid friction and power loss.

Cylinder pressures

If these are high the oil requires to have a high film strength to ensure the maintenance of an adequate oil film between the piston rings and the cylinder walls.

Air condition

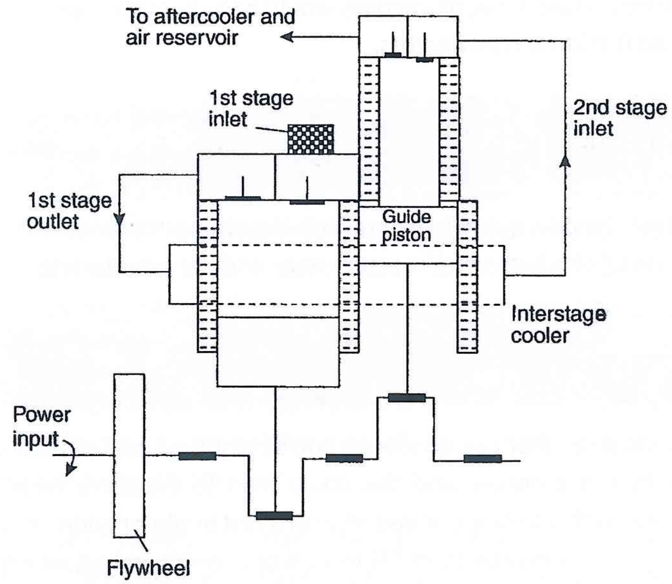
Air contains moisture that can condense out. Straight mineral oils would be washed off surfaces by the moisture and this could lead to excessive wear and possible rusting. To prevent this a compounded oil with a rust inhibitor additive would be used. Compounding agents may be from 5% to 25% of non-mineral oil, which is added to a mineral oil blend. Fatty oils are commonly added to lubricating oil that must lubricate in the presence of water; they form an emulsion which adheres to the surface to be lubricated.

Two-stage air compressors

Most modern diesel engines use starting air at a pressure of about 26–30 bar and to the cylinders, or they could be a combination of a rotary first stage followed by a reciprocating high pressure stage. This latter arrangement leads to a compact, high delivery rate compressor.

Figure 7.7 shows a typical two-stage reciprocating type of air compressor, the pressures and temperatures at the various points would be approximately as follows:

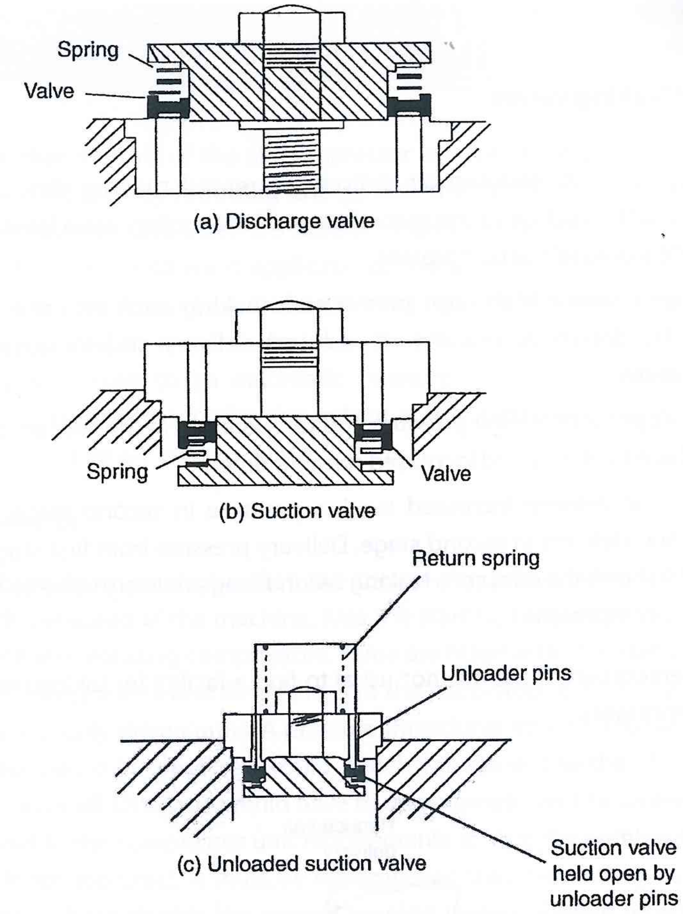
	Delivery pressure	Air temperature	
		Before the coolers	After the coolers
First stage	4 bar	130°C	35°C
Second stage	26 bar	130°C	35°C



▲ **Figure 7.7** *Two-stage air compressor*

Compressor valves

Simple suction and discharge valves are shown in figure 7.8(a–c). These would be



▲ **Figure 7.8** *Compressor valves*

Valve seat

About 0.4% carbon steel hardened and polished working surfaces.

Valve

Nickel steel, chrome vanadium steel or stainless steel, hardened and ground, then finally polished to a mirror finish.

Spring

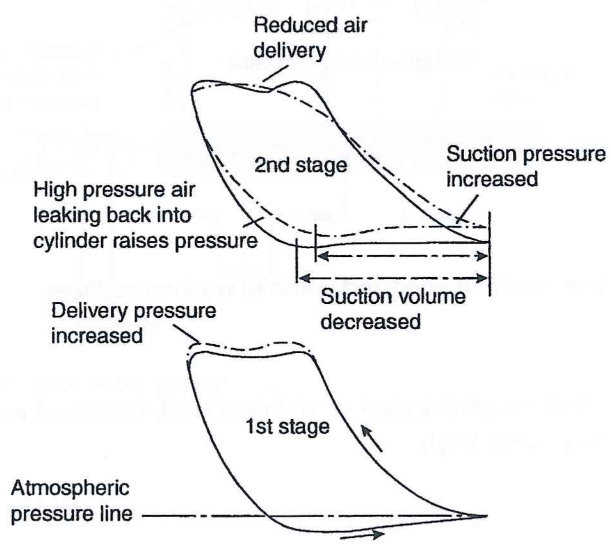
Hardened steel (*Note:* all hardened steel would be tempered).

Valve leakages do occur in practice and this leads to loss of efficiency and increase in

Effects of leaking valves

1. *First-stage suction:* Reduced air delivery, increased running time and reduced pressure in the suction to the second stage. If the suction valve leaks badly it may completely unload the compressor.
2. *First-stage delivery:* With high pressure air leaking back into the cylinder less air can be drawn in, this means reduced delivery and increased discharge temperature.
3. *Second-stage suction:* High pressure and temperature in the second-stage suction line, reduced delivery and increased running time.
4. *Second-stage delivery:* Increased suction pressure in second stage, reduced air suction and delivery in second stage. Delivery pressure from first stage increased. Figure 7.9 shows the effect of a leaking second-stage delivery valve on the indicator cards of a compressor.

It must be remembered that it is not usual to find a facility for taking indicator cards from air compressors.



▲ **Figure 7.9** *Effect of leaking second-stage delivery valves*

Methods of regulating air compressors

We have seen that the role of the air compressor is to raise the pressure of air from a suction to a discharge by inputting energy into the medium. The compressed air has several applications and there are different types of compressor that have been developed to handle the different applications. These different types of compressor also have different methods of control depending upon the type and application of compressor. Therefore, to allow compressors to respond to fluctuations in system demand they are linked to an automatic pressure regulation controller and the controller will start a process to alter the output of the compressor. The most popular methods in use to date are given below.

Start stop control

A general observation would be that the torque required to drive a compressor increases with the speed of the machine. Also the starting torque can be very high as is the case with reciprocating compressors. Some are fitted with star–delta starters but others are still direct online and for this reason the start–stop technology will only be suitable for electrically driven units. A pressure transducer attached to the air receiver set for desired max–min pressures would switch the current to the electric motor's starter either on or off. Drainage would have to be automatic and air receiver relatively large compared to the compressor unit requirements so that the number of starts per unit of time is not too great. It must be remembered that the starting current for an electric motor is about double the normal running current. During its operation the compressor does operate at its optimum efficiency and if the machine is stopped for long periods of time then the overall performance is acceptable.

Constant running control

This method of control is the one used most often for the higher volume machine running at a relatively low pressure. The compressor runs continuously at a constant speed and when the desired air pressure is reached the air compressor is unloaded in some way so that the air is NOT delivered and practically no work is done in the compressor cylinders.

The methods used for compressor unloading vary, but that most commonly used is to shut off the air to the suction side of the compressor. If the compressor receives no air then it cannot deliver any. Or if the air taken in at the suction is returned to the

the compressor cylinder or cylinders and this would provide an economy compared to discharging high pressure air to the atmosphere through a relief valve.

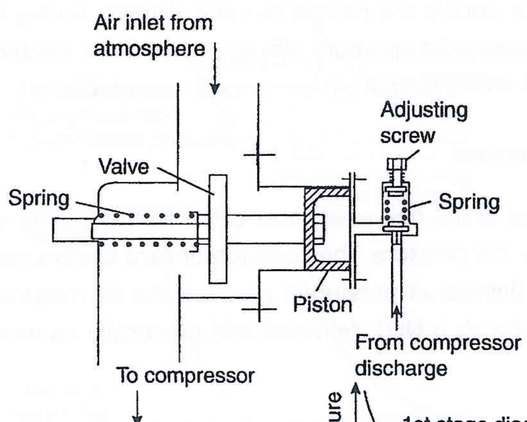
Figure 7.10 shows diagrammatically a compressor unloading valve fitted to the compressor suction. When the discharge air pressure reaches a desired value it will act on the piston causing the spring-loaded valve to close shutting off the supply of air to the compressor.

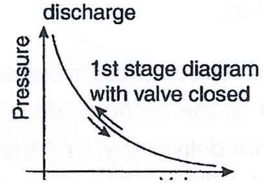
An alternative method of unloading the compressor, while continuing to run it, is to hold the suction valve open. When the compressor is unloaded the suction valve plates are held open by pins which are operated by a relay valve and piston, not unlike that shown in figure 7.10. When the pressure in the air reservoir falls to a preset level, the piston's chamber is vented and return springs push out the holding pins allowing the suction valve to operate normally (figure 7.8).

An alternative method of optimising the use of compressors is to have several smaller machines running in parallel. The number of machines running can be adjusted depending upon the demand.

Variable-speed control

Modern electronics has allowed the development of thyrister control of a.c. synchronous motors and has added another dimension to the efficiency of machinery driven in



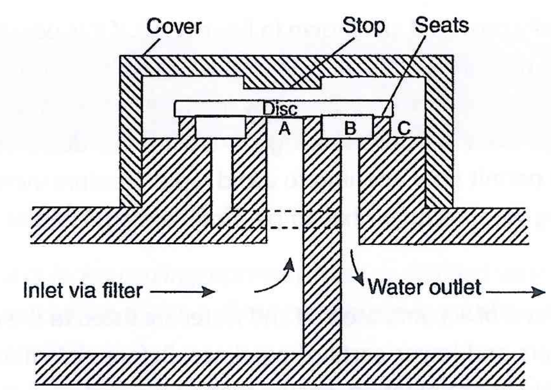


this way. The ability to vary the speed of the compressor presents a number of major advantages over other methods of control. These are:

- A gradual start-up and increase in speed meaning that there are no sudden peaks in the current supply to the motor. The stress of sudden acceleration on the mechanical components is reduced.
- The pressure can be controlled to a much finer tolerance because the speed and therefore the flow rate can be adjusted to match the demand. This reduces the range of the pressure fluctuations and also the stress on the pressure parts of the system. Initially systems can be designed using smaller receivers.
- Due to the efficiency being optimised so is the use of energy and therefore there will be a fuel saving for the vessel.
- Variable speed control is also suited to compressors operating in parallel. Here one of the machines can be optimised by speed control while the others operate on an on–off basis.

Automatic drain

Figure 7.11 shows an automatic air drain trap which functions in a near similar way to a steam trap. With water under pressure at the inlet the disc will lift, allowing the water to flow radially across the disc from A to the outlet B. When the water is discharged and air now flows radially outwards from A across the disc, the air expands increasing in velocity ramming air into C and the space above the disc, causing the disc to close on the inlet. Because of the build-up of static pressure in the space above the disc in this way, and the differential area on which the pressures are acting, the disc is held firmly closed. It will remain so unless the pressure in the space above the disc falls.



In order that this pressure can fall, and the trap reopen, a small groove is cut across the face of the disc communicating B and C through which the air slowly leaks to the outlet.

Obviously this gives an operational frequency to the opening and closing of the disc which is a function of various factors, for example, size of groove, disc thickness and volume of space above the disc. Therefore, it is essential that the correct trap be fitted to the drainage system to ensure efficient and effective operation. These traps should be checked by the watchkeeper by listening for their operation. After a while in operation debris in the water can cause grooves to form across the disc and they stop working.

Air Vessels

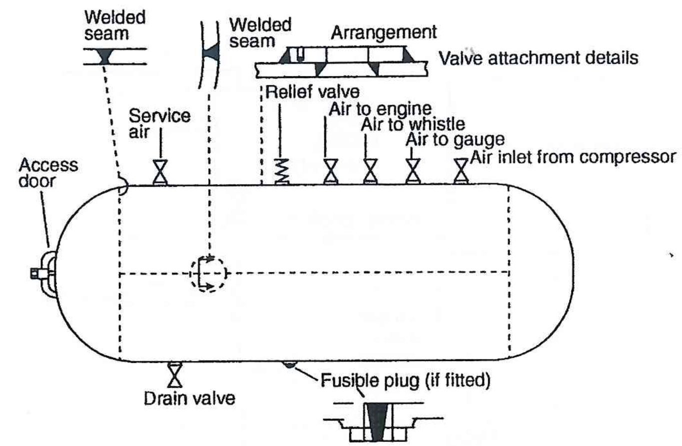
Material used in the construction must be of good quality low carbon steel similar to that used for boilers, for example, 0.2% carbon (max.), 0.35% silicon (max.), 0.4% manganese, 0.05% sulphur (max.), 0.05% phosphorus (max.), u.t.s. 460 MN/m^2.

Welded construction has superseded the rivetted types and welding must be done to class 1 or class 2 depending upon operating pressure. If above 35 bar approximately, then class 1 welding regulations apply.

Some of the main points relating to class 1 welding are that the welding must be radiographed, annealing must be carried out at a temperature of about 600°C and a test piece must be provided for bend, impact and tensile tests together with micrographic and macrographic examination.

Mountings generally provided are shown in figure 7.12. If it is possible for the receiver to be isolated from the safety valve then it must have a fusible plug fitted, melting point approximately 150°C, and if carbon dioxide is used for fire fighting it is recommended that the discharge from the fusible plug be led to the deck. Stop valves on the receiver generally permit slow opening to avoid rapid pressure increases in the piping system, and piping for starting air has to be protected against the possible effects of explosion.

Drains for the removal of accumulated oil and water are fitted to the compressor, filters, separators, receivers and lower parts of pipelines. Before commencing to fill the air vessel after overhaul or examination, ensure that:



▲ **Figure 7.12** *Air reservoir*

2. Check pressure gauge against a master gauge.
3. All doors are correctly centred on their joints.

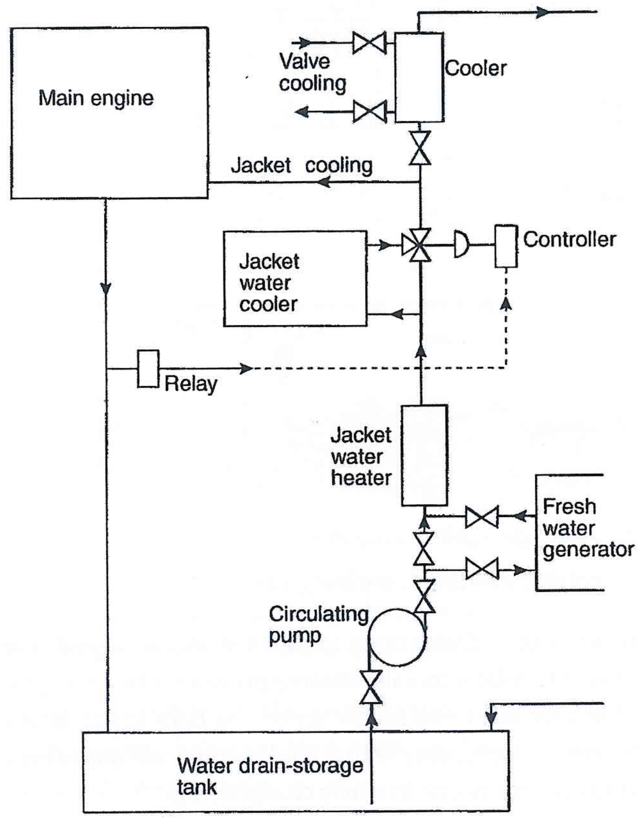
Run the compressor with all drains open to clear the lines of any oil or water, and when filling open drains at regular intervals, observe pressure. After filling close the air inlet to the bottle, check for leaks and follow up on the door joints. When emptying the receiver prior to overhaul, etc., ensure that it is isolated from any other interconnected receiver which must, of course, be in a fully charged state.

Cleaning the air receiver internally must be done with caution. Any cleaner which gives off toxic, inflammable or noxious fumes should be avoided. A brush down and a coating on the internal surfaces of some protective, harmless to personnel, such as a graphite suspension in water could be used.

Cooling Systems

These can conveniently be grouped into sections.

1. *Cylinder cooling or jacket cooling:* normally fresh or distilled water (figure 7.13). This may incorporate cooling of the turbine or turbines in a turbo-charged engine and exhaust valve cooling.
2. *Fuel valve cooling:* This would be a separate system using fresh water or a fine

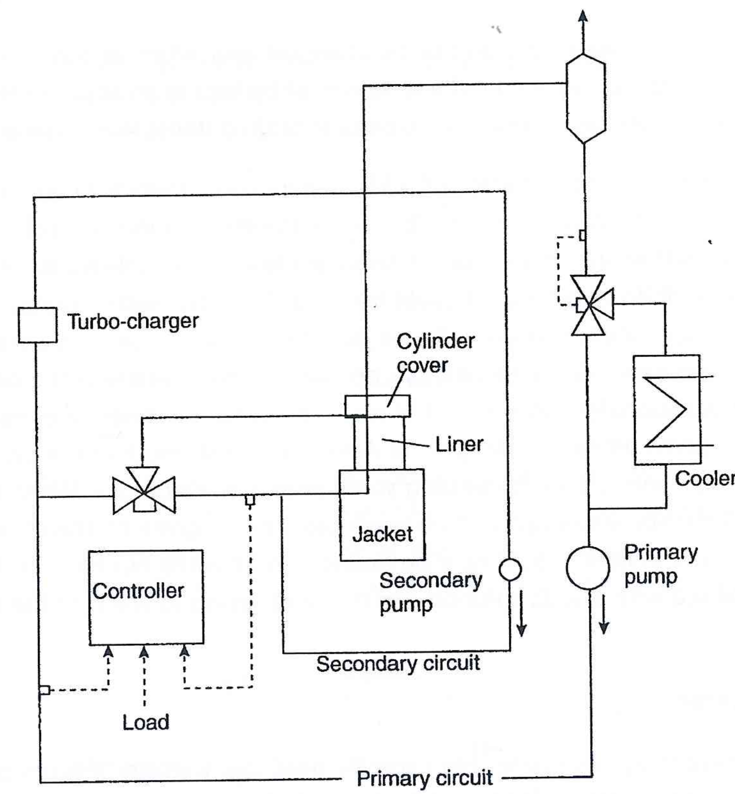


▲ **Figure 7.13** *Jacket cooling system*

3. *Piston cooling:* This may be lubricating oil, distilled or fresh water. If it is oil the system is generally common with the lubrication system. If water, a common storage tank with the jacket cooling system would generally be used.
4. *Charge air cooling:* This is normally seawater.

Load-controlled cylinder cooling

In an effort to reduce the danger of local liner corrosion over the whole engine load some manufacturers are employing cooling systems that are load dependant. In such a system, shown in figure 7.14, the cooling flow is split into a primary circuit, bypassing the liner, for cylinder head cooling. In the secondary circuit uncooled water from engine



▲ **Figure 7.14** *Load-controlled cylinder cooling*

The advantages claimed for such a system include:

1. Possible savings in cylinder lubrication oil feed rate.
2. Omission of cylinder bore insulation.
3. Reduced cylinder liner corrosion.

Comparison of coolants

Fresh water

Inexpensive, high specific heat, low viscosity. Contains salts which can deposit, obstruct flow and cause corrosion. Requires treatment. Leakages could contaminate lubricating oil system leading to loss of lubrication, possible overheating of bearings and bearing corrosion. Requires a separate pumping system.

It is important that water should not be changed very often as this can lead to increased deposits. Leakages from the system must be kept to an absolute minimum, so a regular check on the replenishing-expansion tank contents level is necessary.

If the engine has to stand inoperative for a long period and there is a danger of frost, (a) drain the coolant out of the system, (b) heat up the engine room or (c) circulate the system with heating on. It may become necessary to remove scale from the cooling spaces and the following method could be used. Circulate, with a pump, a dilute hydrochloric acid solution. A hose should be attached to the cooling water outlet pipe to remove gases. Gas emission can be checked by immersing the open end of the hose occasionally into a bucket of water. Keep compartment well ventilated as the gases given off can be dangerous. Acid solution strength in the system can be tested from time to time by putting some onto a piece of lime. When the acid solution still has some strength and no more gas is being given off then the system is scale free. The system should now be drained and flushed out with fresh water, then neutralised with a soda solution and pressure tested to see that the seals do not leak.

Distilled water

More expensive than fresh water, high specific heat, low viscosity. If produced from evaporated salt water it would be acidic. No scale-forming salts. Requires separate pumping system. Leakages could contaminate the lubricating oil system, causing loss of lubrication and possible overheating and failure of bearings, etc.

Additives for cooling water

Those generally used are either anti-corrosion oils or inorganic inhibitors. If pistons are water cooled an anti-corrosion oil is recommended as it lubricates parts which have sliding contact. The oil forms an emulsion and part of the oil builds up a thin unbroken film on metal surfaces. This prevents corrosion but is not thick enough to impair heat transfer. Inorganic inhibitors form protective layers on metal surfaces guarding them against corrosion.

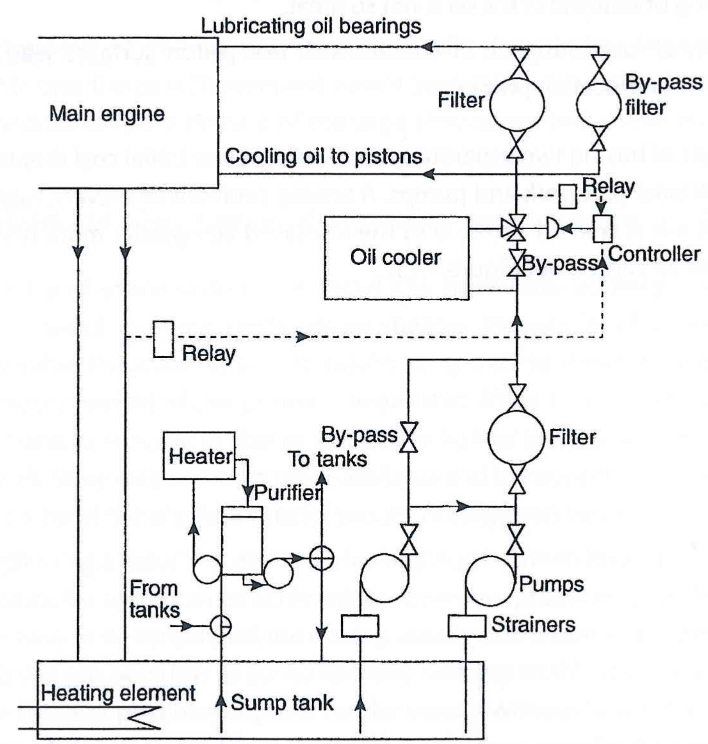
It is important that the additives used are not harmful if they find their way into drinking water – this is possible if the jacket cooling water is used as a heating medium in a fresh water generator. Emulsion oils and sodium nitrite are both approved additives, but the latter cannot be used if any pipes are galvanised or if any soldered joints exist. Chromates cannot be used if the cooling water is used in a fresh water generator and it

Lubricating oil

This is expensive and generally there is no separate pumping system required (see Figure 7.15) since the same oil is normally used for lubrication and cooling. Leakages from cooling system to lubrication system are relatively unimportant provided they are not too large; otherwise one piston may be partly deprived of coolant with subsequent overheating.

Due to reciprocating action of pistons some relative motion between parts in contact with the coolant supply and return system must occur; oil will lubricate these parts more effectively than water. No chemical treatment required. Lower specific heat than water, hence a greater quantity of oil must be circulated per unit time to give the same cooling effect.

If the lubricating oil is subject to a high temperature it can burn leaving carbon deposit as it does so. This deposit on the underside of a piston crown could lead to impairment of heat transfer, overheating and failure of the metal. Generally the only effective method of dealing with the carbon deposit is to dismantle the piston and physically



remove it. Since oil can burn in this way a lower mean outlet and inlet temperature of the oil has to be maintained. In order to achieve this more oil must be circulated per unit time.

Some engines may use completely separate systems for oil cooling of pistons and bearing lubrication. The advantages gained by this method are as follows:

1. Different oils can be used for lubrication and cooling, a very low viscosity mineral oil would be better suited to cooling than lubrication.
2. Additives can be used in the lubricating oil that would be beneficial to lubrication, for example, oiliness agents, e.p. agents and V.I. improvers, etc.
3. Improved control over piston temperatures.
4. If oil loss occurs, then with separate systems the problem of detection is simplified and in the case of total oil loss in either system, the quality to be replaced would not be as great as for a common system.
5. Contamination of the oil in either system may take place. In the event the problem of cleaning or renewal of the oil is not so great.
6. Oxidation of lubricating oil in contact with hot piston surfaces leads to rapid reduction in lubrication properties.

Disadvantages of having two separate systems are: greater initial cost due to separate storage, additional pipework and pumps. A sealing problem to prevent mixing of the two different oils is created and due to the increased complexity more maintenance would have to be carried out (figure 7.15).

8 MEDIUM-SPEED DIESELS

The term medium speed refers to diesels that operate within the approximate speed range of 300–800 rev/min. High speed is usually 1,000 rev/min and above.

The development of the medium-speed, usually four-stroke, engine has been considerable over the past 20 years and now it is a serious competitor for applications which were once only the domain of the large slow-speed two-stroke engines or the steam turbine.

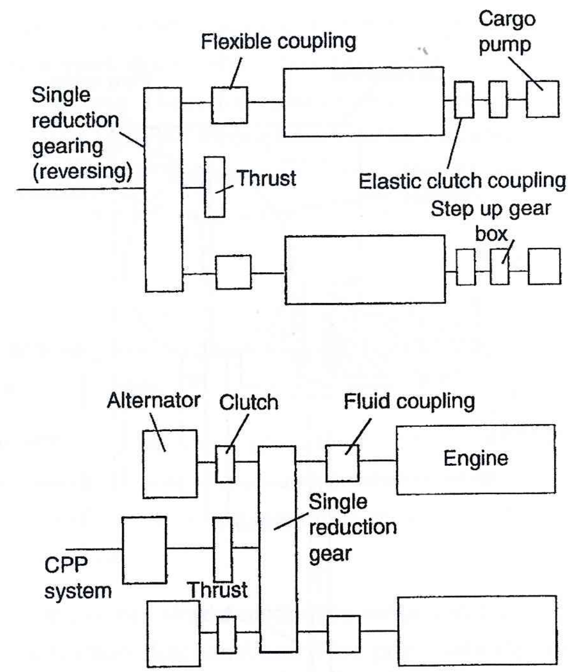
The advantages and salient features of the medium-speed diesel are as follows:

1. Compact and space saving. The vessel can have reduced height and broader beam – useful in some ports where shallow draught is of importance. The considerable reduction in engine height compared to direct drive engines and the reduced weight of components means that lifting tackle, such as the engine room crane, is reduced in size as it will have lighter loads to lift through smaller distances. More cargo space is made available and because of the higher power to weight ratio of the engine a greater weight of cargo can be carried.
2. Through using a reduction gear a useful marriage between ideal engine speed and ideal propeller speed can be achieved. For optimum propeller speed hull form and rudder have to be considered, the result is usually a slow turning propeller (for large vessels this can be as low as 50–60 rev/min). Gearing enables the naval architect to design the best possible propeller for the vessels without having to consider any dictates of the engine. Engine designers can ignore completely propeller speed

3. Modern tendency is to utilise unidirectional medium-speed geared diesels coupled to either a reverse reduction gear, controllable pitch propeller (CPP) or electric generator. The second two of these methods are the ones primarily used and the advantages to be gained are considerable. They include:
   a. Less starting torque required, clutch disengaged or CPP in neutral.
   b. Reduced number of engine starts, hence starting air capacity can be greatly reduced and compressor running time minimised. Classification society requirements are six consecutive starts without air replenishment for non-reversible engines and twelve for reversible engines. Cylinder liner wear rate increases during starting.
   c. Engines can be tested at full speed with the vessel alongside a quay without having to take any special precautions.
   d. With the mechanical drive arrangement and the engine or engines running continuously, power can be taken off via a clutch or clutch/gear drive for the operation of electric generators or cargo pumps, etc. Hence the main engine has become a multi-purpose 'power pack'.
   e. Improved manoeuvrability, vessel can be brought to rest within a shorter distance by intelligent use of the engines and CPP.
   f. Staff work load during 'stand-by' periods is reduced and the system lends itself ideally to simple bridge control.
4. With two engines coupled via gearing one may be disengaged, while the other supplies the motive power, and overhauled. This reduces off hire time as the voyage is continued at slightly reduced speed with a fuel saving.
5. Spare parts are easier to store and manhandle, therefore unit overhaul time will be greatly reduced.

Engine Couplings, Clutches and Gearing

Various arrangements of geared engines coupled together are possible, the basic arrangement depends upon the services the engine has to supply, for example, a high electrical load in port may have to be catered for with the alternator being driven at a higher speed than the engine. Hence a step up gear box would be required along

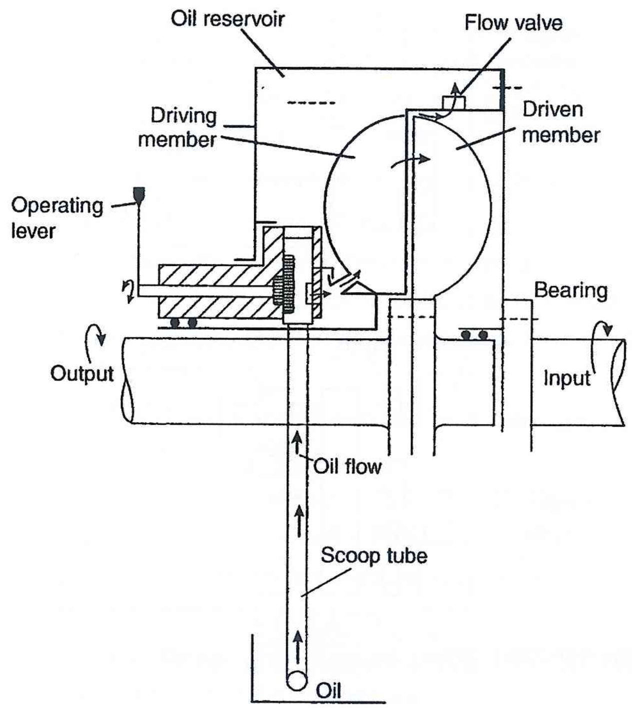


▲ **Figure 8.1** *Engine arrangements*

Fluid couplings (Figure 8.2)

These are completely self-contained, apart from a cooling water supply, they require no external auxiliary pump or oil feed tank. A scoop tube when lowered picks up oil from the rotating casing reservoir and supplies it to the vanes for coupling and power transmission; withdrawal of the scoop tube from the oil stops the flow of oil to the vane which then drains to the reservoir. During power transmission a flow of oil takes place continuously through the cooler and clutch.

Fluid clutches operate smoothly and effectively. They use a fine mineral lubricating oil and have no contact and hence no wear between driving and driven members. Torsional vibrations are dampened out to some extent by the clutch and transmitted speeds can be considerably less than engine speed if required by suitable adjustment of the scoop tube. It is possible to have a dual entry scoop tube for reversible engines, this obviates the use of c.p. propellers or reversible reduction gears but the control problem is considerably more complex with reversible engines, they have to be stopped and started and if four-stroke engines are used camshafts have to be moved,



▲ **Figure 8.2** *Fluid coupling (vulcan)*

Reverse reduction gear

These gear systems are mainly restricted, at present, to powers of up to about 4800 kW for twin-engined single-screw installations. Their obvious advantages are as follows:

1. Uni-directional engine.
2. No c.p. propeller required.
3. Ability to engage or disengage either engine of a twin-engine installation from the bridge by a relatively simple remote control.
4. Improved manoeuvrability, etc.

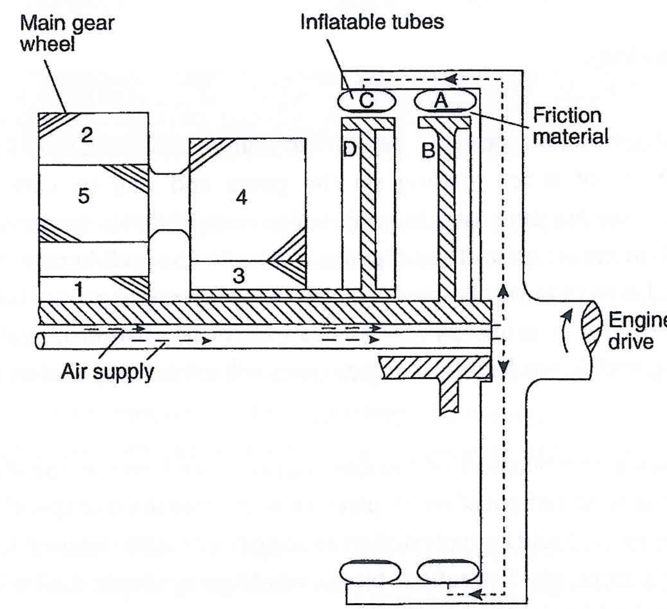
When dealing with higher powers the friction clutches used in the system can become excessively large, great heat generation during engagement may require a cooling

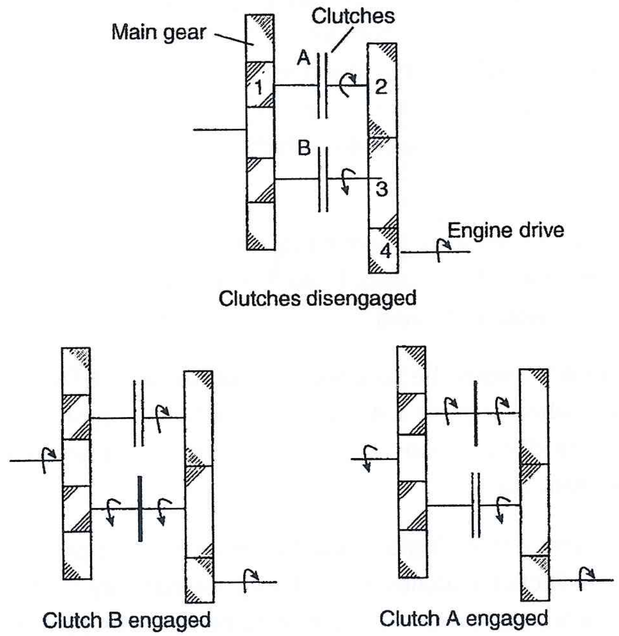
Two systems of reverse reduction gear are shown in figures 8.3 and 8.4. In figure 8.3, the engine drives a steel drum which has two inflatable synthetic rubber tubes bonded to its inner surface. These tubes have friction material, like brake lining, on their inner surface. Air is supplied through the centrally arranged tube, or the annulus formed by the tube and shaft hole to one or the other of the inflatable tubes. Two flanged wheels are connected via hollow shafts and gears to the main gear wheel and shaft.

For operation ahead, air would be supplied to inflatable tube A. which would then by friction on flanged wheel B bring gears 1 and 2 up to speed; gears 3, 4 and 5 together with flanged wheel D would be idling.

For astern operation, air would be supplied to inflatable tube C (A evacuated) and by friction on flanged wheel D gears 3, 4, 5 and 2 would be brought up to speed, gear 1 and drum B would be idling. For single reduction, gears 3 and 4 would be the same size and so would be gears 1 and 5.

An alternative system, either single or double reduction but probably the latter, is shown in figure 8.4. Friction clutches A and B are pneumatically controlled from some remote position. Gears 1, 2, 3 and 4 would have to be the same size if the gear were to be single reduction – but this is most unlikely.



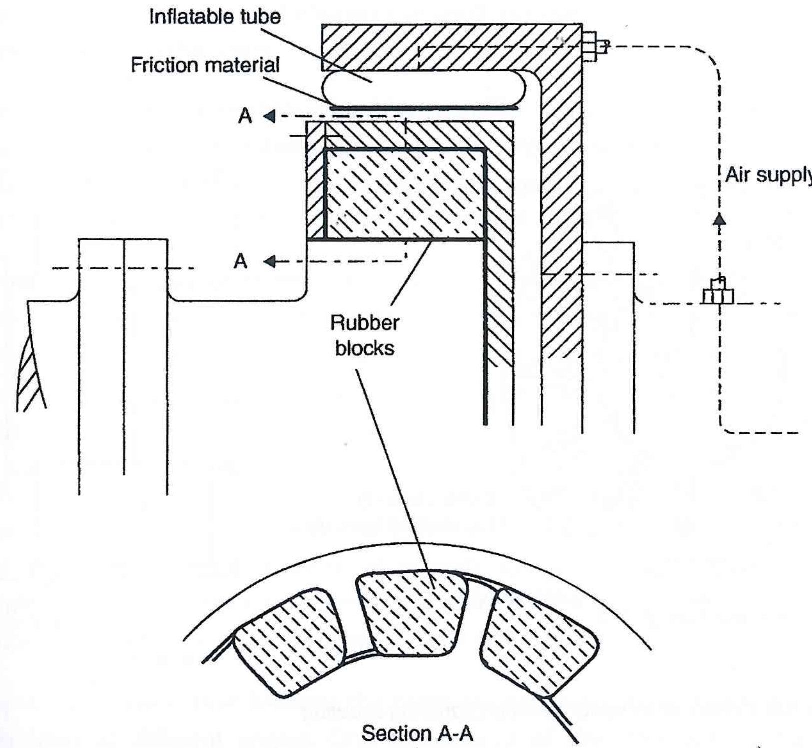


▲ **Figure 8.4** *Reversible reduction gear*

Flexible couplings

These are used between engine and gearbox to dampen down torque fluctuations, reduce the effects of shock loading on the gears and engine, cater for slight misalignments. They are also used in conjunction with clutches for power take-off when required. In construction they may be similar to the well-known multi-tooth type to be found in turbine installations or employ diaphragms or rubber blocks. Those types that use rubber or synthetic rubber, such as Nitrile, give electrical insulation between driving and driven members, but all types will minimise vibration and reduce noise level.

Figure 8.5 shows a combination of flexible couplings and pneumatically operated friction clutch, the arrangement of which gives a smooth transition of speed and torque during engagement; it could be typical of an arrangement for the take-off for electrical power or cargo pumps, etc. The rubber blocks would be synthetic if oil is likely to be present as natural rubber is attacked by oil.

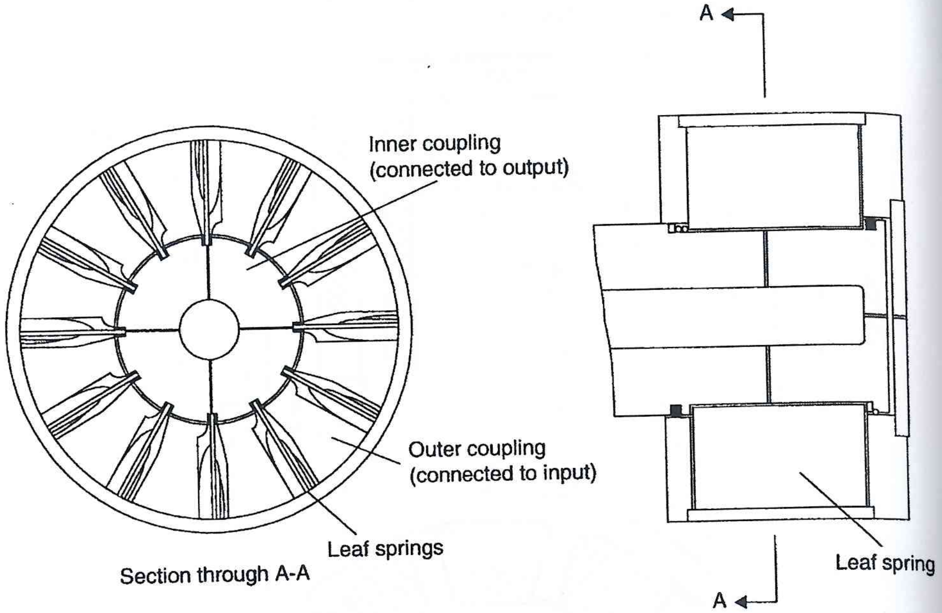


▲ **Figure 8.5** *Flexible clutch coupling*

The Geislinger coupling

The main function of a Geislinger coupling is to assist in the damping out of torsional vibrations. This is accomplished by connecting the engine crankshaft to the load via flexible steel leaf springs arranged radially in the coupling, which is also filled with oil. As torsional fluctuations occur they are absorbed by the leaf springs which deflect and displace oil to adjacent chambers, slowing down the relative movement between the inner and outer components of the coupling. The makers claim that this effective damping is achieved without problems of wear because of the absence of friction (figure 8.6).

Damping oil is supplied from the engine oil system through the centre of the coupling. It is returned to the engine through hollow coupling bolts. Maintenance is limited to cleaning, inspection and the replacement of 'O' rings.



▲ **Figure 8.6** *Geislinger torsional vibration damping coupling*

Gearboxes, Thrust Blocks, Shafting and Controllable Pitch Propeller (CPP)

Shafting and CPPs are covered in more detail in chapter 6 of Volume 8 in this marine engineering series. However, the use of CPP operating through a gearbox and coupled with medium-speed diesels is a fairly common arrangement for higher power ocean going vessels such as Rolls-Royce ships and product carrying tankers and therefore needs to be described as part of this chapter.

Gear boxes are very interesting and need a great deal of care in both manufacture and on-going care. They have to transmit sometimes large forces through relatively small areas of contact. The metal obviously transmits that power but metal-to-metal contact would mean that the component parts would not last for long. This means that the quality of the oil and the oil supply is vital to the on-going success of the gearbox.

... said to be meshing they are actually sliding over one another

is catastrophic if metal gets between the teeth on the gearwheel due to the small clearance between the gears.

Gearboxes transmit power through a drive train. The gears can be arranged to increase or decrease the speed of rotation of input and output shafts or they can be used to transmit power through an angle so that the output shaft is pointing in a different direction to the original shaft.

The basic arrangement is to have straight gears around the outside of a wheel, fixed to the end of an input shaft, linking with a second wheel, of a different diameter, fixed to the start of an output shaft. The dimensions of the gear wheels will determine the different input and output speeds to and from the gearbox. Any combination of speeds can be chosen to suit the designer's needs.

Straight cut gears present a problem in so much as they have the minimum surface-to-surface contact area through which to transmit the power. Therefore they have to be sized accordingly. If the teeth are set at an angle across the end of the wheel to form a helical gear then the area for transmitting power in increased and the gear wheel can be more compact than a gear transmitting the same power using straight teeth.

The problem here is that because the teeth are set at an angle there will be forces transmitted at different angles. One component of the force will be transmitted through the gear as required and another will be transmitted along the shaft as a vector component of the total force from the input shaft.

This will result in a lateral thrust being transmitted along the shaft. The value of the thrust will depend upon the angle of the teeth and the total power from the input shaft. This means that with a single helical gearwheel a thrust block of some sort will be needed to counteract the thrust from the gearwheel.

Another answer to this problem is to arrange for half the width of the gearwheel to have helical teeth set in one direction and the other half of the wheel to have teeth set in the opposite direction. This means that the thrust from one set of teeth is offset by the thrust from the other set of teeth and the need of a thrust block has been overcome.

The profile of the teeth is very important to the smooth operation of the gearbox because for the teeth at the end of the input shaft to mesh and transmit power they have to slide into the space in-between the gears on the output shaft. It is also important for the tip of the gear not to make any contact with the root of the opposing gear wheel as this will also impose forces on the gears resulting in gearbox failure.

Smaller gearboxes and low-power gearboxes might be lubricated by relying on the oil are not so well protected and wear can occur during this time. Larger or more powerful arrangements will have the oil pumped into the gearbox where it is arranged to spray directly onto the meshing gears ensuring the even at the start-up stage the gear teeth are well lubricated.

Gearboxes do need to be checked and looked after. Any unusual noises must be investigated and routine inspections must be made at the appropriate intervals. It is not a good idea to make frequent visual inspections because there is more chance of introducing foreign materials inside the casing.

When an inspection is made the engineer should be looking out for the following:

- Broken teeth on the gearwheels
- Discolouration anywhere on the gearwheel of teeth (indicating overheating)
- Excessive wear on the faces of the gear teeth (indicating a lack of lubrication)
- Condition of the oil.

A sample of oil can be sent away for further analysis. This will be checked for any metal or water content and from this analysis a picture of the condition of the gearbox can be formed. This process obviously takes some time therefore some companies such as Kittywake International are now supplying analysis kits that can be used on-board. The next step is to offer online real-time testing of lubricating oil. This will then start to move the industry towards a CBM approach which is described in more detail in chapter 12 of Volume 8 in this marine engineering series of books.

Propellers

Although normally described under naval architecture propellers have become the focus of efficiency gains in recent years and therefore will come under engineering knowledge as it could be an area where fuel savings could be made by retrofitting an updated system not available when the vessel was built.

Exhaust Valves

12-cylinder V engines this gives a total of 48 exhaust valves. A not inconsiderable quantity, and if the plant is to burn fuel of high viscosity, the maintenance problem for these valves could be considerable.

In order to minimise maintenance and to prolong valve life, bearing in mind that burning of high viscosity oil is essential due to the higher cost of light diesel oil, certain design parameters and operating procedures must be followed. These are:

1. Separately caged exhaust valves are preferred even though they increase the initial cost. If they are made integral with the cylinder head and used with poor quality fuel then there will be an increased frequency of valve replacement and overhaul. Cylinder head removal each time becomes a tedious time-consuming operation and the caged valves save a lot of time. However part load or short trip operation can be a problem as the exhaust valves could be running at a temperature where the dew point of the gasses is reached. Some cross-channel operators have in the past had problems with acid erosion of exhaust valves spindles on uprated Pielstick PC2.5 because they had water-cooled exhaust valve cages. The previous version of the engine running on the short voyages did not have the same problem.
2. All connections to the valves, cooling, exhaust, etc. should be capable of easy disconnection and re-assembly.
3. Materials that have to operate at elevated temperatures must be capable of withstanding the erosive and corrosive effects of the exhaust gas. When burning oils of high viscosity which contain sodium and vanadium deposits can form on the valve seats which, at high temperatures (in excess of 530°C at the valve seat), become strongly corrosive sticky compounds which lead to burnt valves. Hence, the need for materials that can withstand the corrosion and for intense cooling arrangements for valve seats.
4. Stellite valve seats have started the quest for improved durability of exhaust valves. Stellite is a mixture of cobalt, chromium and tungsten extremely hard and corrosion resistant that is fused on to the operating surfaces.

Low temperature corrosion due to sulphur compounds can occur during prolonged periods of running under low load conditions. The valve spindle and guide, which would be at a relatively low temperature, are the principal places of attack due to the effective cooling in this region. Ideally, valve cooling should be a function of engine load with the valve being maintained at a uniform temperature at all times, as stated this could prove complicated and expensive to arrange for part load and low load conditions.

Further to the use of Stellite, a nickel-chromium alloy, strengthened by additions of titanium, aluminium and carbon called Nimonic 80A, has gained favour for use in exhaust valve construction. Recently MAN have found that welding a high-temperature resilient Ni-Cr alloy onto a stainless steel spindle would dramatically improve the hardness and ductility of the valve seat as well as its resistance to cracking when compared to chromium- and nickel-based hard facings including Nimonic 80A.

In the first stage of the process, the stainless steel DuraSpindle is placed through a new robotic welding procedure where Inconel, an alloy traditionally used in gas turbines, is welded into the groove of an exhaust spindle valve seat.

Once the alloy has been welded in place, the DuraSpindle is then machined after which more than 10 tonnes of force is used during the special rolling process to work harden the Inconel weld to 500 HV. While the spindle is being rolled and rotated three or four concentric grooves, depending on the spindle size, are etched into the seat at a depth of several millimetres. This further hardens a relatively ductile material.

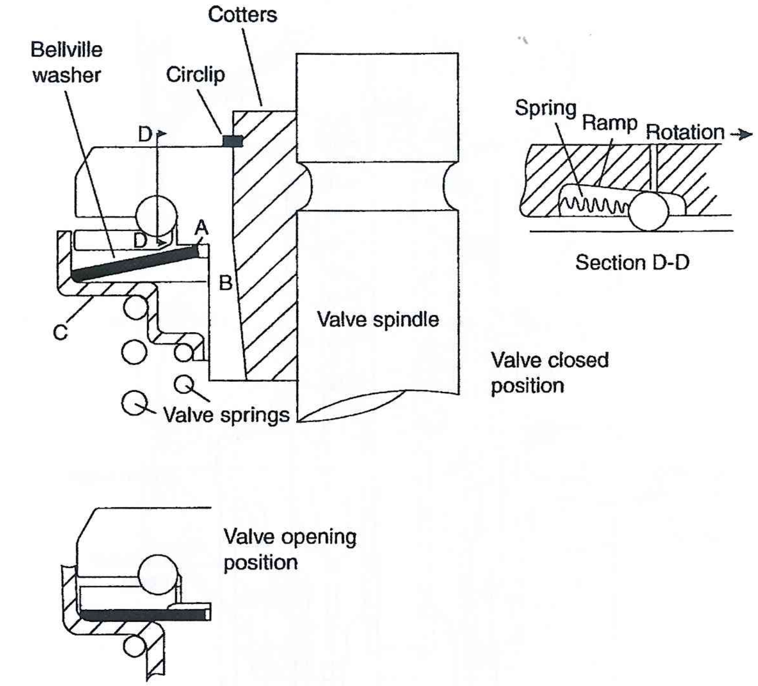
The rolling process provides compressive stresses into the component, as opposed to tensile stresses which may cause cracking in the seat area. Compressive stressing significantly reduces the probability of cracking even in the advent of welding defects.

The hard facing on the spindle seat is further hardened by heating the material up to 600–700°C. The metallurgical reaction, called precipitation hardening, further hardens the seat to 600 HV.

Compared with an Alloy 50-type hard facing material DuraSpindle is 20% harder and 50% harder if compared to a spindle with Stellite hard facing or Nimonic 80A.

5. Effective lubrication of the valve spindle is necessary to avoid risk of seizure and possible mechanical damage due to a valve 'hanging up'. In order to minimise lubricating oil usage the lubrication system for the valves would be similar to that used for cylinder lubrication and since the amount of oil used would therefore be in small quantities, any contamination of the oil by combustion products and water, etc. would be minimal, and this would also increase the life of crankcase lubricating oil.

Rotocap

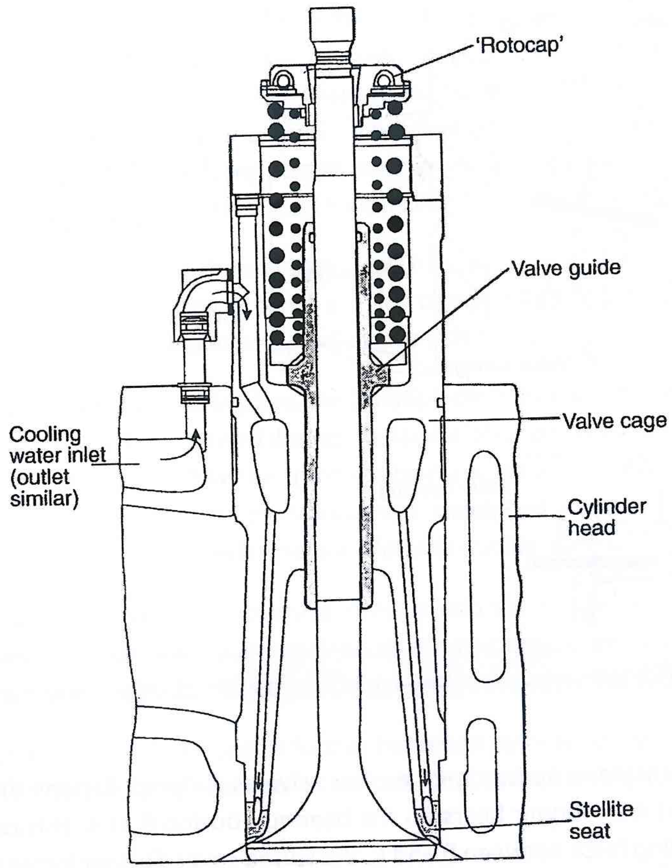


▲ **Figure 8.7** *Rotocap*

follows: an increase in spring force on the valve as it opens flattens the belleville washer so that it no longer bears on the bearing housing B at A. This removes the frictional holding force between B and C, the spring cover. Further increase in spring force causes the balls to move down the ramps in the retainer imparting as they move a torque which rotates the valve spindle. As the valve closes, and load from the belleville washer is removed from the balls and they return to the position shown in section D–D.

Figure 8.8 shows an exhaust valve with welded stellited seat around which cooling water flows keeping the metal temperature at full load conditions well below 500°C, minimising the risk of attack by sodium-vanadium compounds. The valve is housed in a 'cage' which can be easily removed for maintenance without disturbing the cylinder cover.

It has been stated in Chapter 2 that modern medium-speed four-stroke engines usually have four valves per cylinder head to maximise the CSA of the ports and thus improve gas flow through the engine. The gas flow of a typical four-valve cylinder head is shown

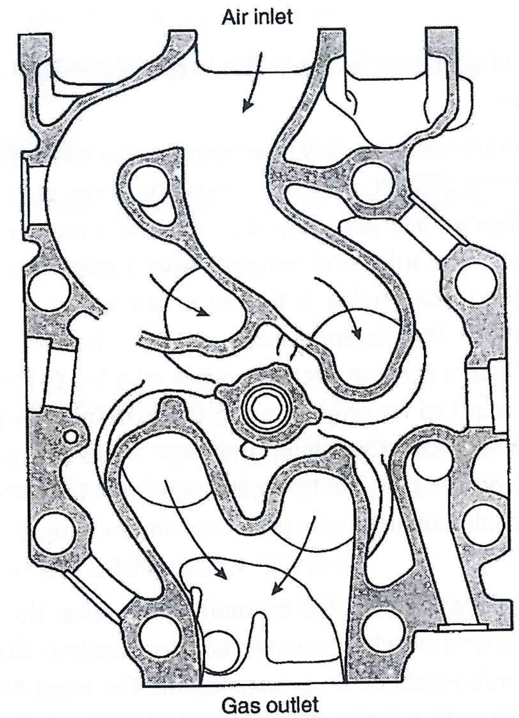


▲ **Figure 8.8** *Exhaust valve*

Engine Design

The principal design parameters for medium-speed diesel engines are:

1. High power/weight ratio.
2. Simple, strong, compact and space saving.
3. High reliability.



▲ **Figure 8.9** *Gas flow of typical four-valve cylinder head*

understand arrangements are inherent features of good design (figures 8.10 and 8.12).

6. Easily capable of adaption to unmanned operation.
7. Low fuel and lubricating oil consumption.
8. High thermal efficiency.
9. Low cost and simple to install.
10. Four-stroke design leads to electronic control and use of advanced environmental techniques such as the Miller cycle.

Types of engine configuration

Either two- or four-stroke cycle single acting turbo-charged with 'in line' or 'V' cylinder configuration. The main choice is, certainly at present, for the four-stroke engine and there are various reasons for this.

1. They are capable of operating satisfactorily on the same heavy oils as slow-speed two-stroke engines.
2. Effective scavenging is relatively easy to achieve in slow-speed two-stroke engines but it becomes more difficult with an increase in mean piston speed. Modern medium-speed engines are generally, but not exclusively, of the four-stroke configuration. With large inlet and exhaust valve overlap effective scavenging can be accomplished. Scavenging is further improved by utilising high turbo-charger pressure ratios. The current versions of turbo-chargers using single-stage aluminium compressors achieve pressure ratios of 4.5–5.0. However, two-stage turbo-charging is required for engines using the extreme Miller cycle. Using the Diesel or Otto cycle, good scavenging and high turbo-charger pressure ratios result in engines producing high BMEP figures. The use of the Miller cycle reduces the maximum possible but also reduces the maximum temperature and also the NOx produced.
3. The mean piston speed is calculated by multiplying twice the stroke times the rev/s. For medium-speed diesels it would be approximately 9–10 m/s and for slow-speed diesels 7–9 m/s would be an average figure. The latest MAN engines have been type approved with a mean piston speed of 8.97 m/s for the S80ME-C9.2 and 8.49 m/s for the G80ME-C9.2 as can be seen from the ultra-long stroke engine that has a reduced piston speed over the super-long stroke engine. However the cyclic stresses involved are greater for the medium-speed engine. In order that greater power can be developed in the cylinder the working fluid must be passed through the engine faster, hence the higher the mean piston speed for a given unit the greater the power. Practical limitations govern the piston speed, such as the relation between cylinder CSA and areas of exhaust and air inlet, method of turbo-charging and inertia forces are the main limitations. To reduce inertia forces designers have in the past utilised aluminium alloy for piston skirts and in some cases entire pistons. However, as the output of medium-speed engines have increased the limitations of aluminium become apparent. Designers of high output engines now specify cast or forged steel for piston crowns and nodular cast iron for piston skirts. The greater mass of this type of piston means that the higher inertia forces result and cognisance of this must be made when designing the connecting rod and bottom end arrangements. Inertia forces must be taken into account for bearing loads – important in trunk piston engines (i.e. the majority of medium- and high-speed diesels) where the guide surface is the cylinder liner, a smaller side thrust means less friction and cylinder liner and piston wear.
4. Engine can operate with the turbo-charger out of commission: this would present

5. Turbo-charger size and power can be reduced.
6. Specific fuel consumption is comparable with the two-stroke engines.

Typical 'V' type engine

The following is a brief description of a medium-speed diesel engine currently in use:

- Cylinder bore = 400 mm
- Stroke = 560 mm
- BMEP = 23 bar
- Maximum cylinder pressure = 160 bar
- Four-stroke turbo-charged with up to 18 cylinders developing approximately 700 kW (MCR) per cylinder at approximately 600 rev/min.

Overall dimensions of a 18 cylinder 'V' type

- Length = 10.25 m
- Height = 5.0 m
- Width = 4.0 m
- Dry weight = 145 tonnes.
- Specific fuel consumption = 175 g/kWh.

Bedplate and cylinder blocks are of heavy section cast iron, this gives a strong compact arrangement with good properties for damping out vibrations.

The crankshaft, of an 'underslung' design, is a solid forging. The connecting rod is also forged but is of the 'marine-type' bottom end and is two pieces. Pistons are of a composite design with forged steel crown and a cast iron skirt. Piston crown is bore cooled. Liners are of good quality grey cast iron alloy and are bore cooled in the vicinity of the combustion space.

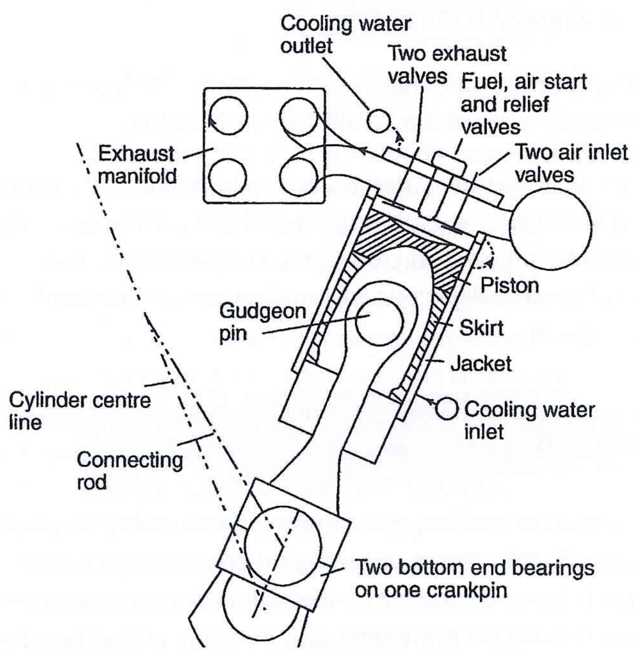
Future development

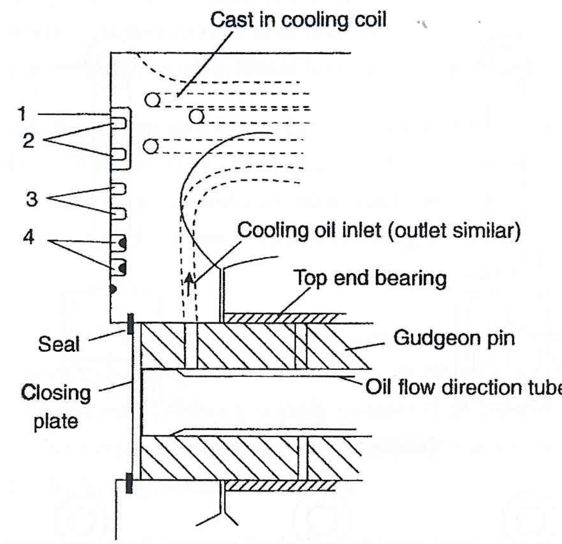
The trend in the field of the medium-speed engine is towards higher power outputs per cylinder, with high reliability, when operating on cheaper high-viscosity fuels. Much development work is being carried out by manufacturers to improve the combustion process. This work focuses on the timing and duration of fuel injection to achieve reliable combustion and manufacturers are now testing engines operating with firing pressures in excess of 210 bar.

- Cylinder bore = 580 mm
- Stroke = 600 mm
- Speed = 450 rev/min
- Power per cylinder = 1,250 kW.

Typical lubrication and piston cooling system

A pump, which could be main engine driven, supplies oil to a main feeder pipe wherein oil pressure is maintained at approximately 6 bar. Individual pipes supply oil to the main bearings from the feeder, the oil then passes through the drilled crankshaft to the crankpin bearing then flows up the drilled connecting rod to lubricate the small end bush. It then flows around the cooling tubes cast in the piston crown then back down the connecting rod to the engine sump. Oil would also be taken from the main feeder to lubricate camshaft gear drive, camshaft bearings, pump bearings, etc. (figure 8.10).



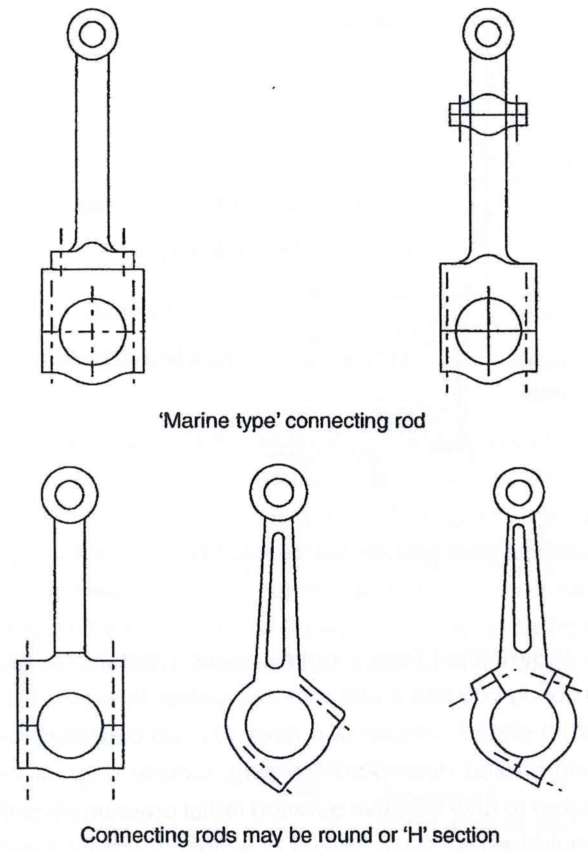


▲ **Figure 8.11** *Piston cooling*

Figure 8.11 shows in simplified form a typical cooling system for alloy pistons, cast in the piston is a cooling coil and a cast iron ring carrier (marked (1) in the diagram); (2) are two chromium-plated compression rings; (3) two copper-plated compression rings; (4) two spring-backed downward scraping, scraper rings of low inertia type. They are spring backed to give effective outward radial pressure since the gas pressure behind the ring would be very small. The oil flow direction tube is expanded at each end into the gudgeon pin and it is so passaged to direct oil flow and return to their respective places without mixing.

Due to complex vibration problems that can arise in medium-speed engines of the 'V' type it would appear important to have a very strong and compact arrangement of bedplate, etc. Excessive vibration of the structure can lead to increased cylinder liner wear and considerable amounts of lubricating oil being consumed.

Alkaline lubricating oil of the type used in these engines is expensive and because the engines are mainly trunk type consumption rates can be high. Positioning, and type, of oil scraper ring is important. With some engines they have been moved from a position below the gudgeon pin to above since considerable end leakage sometimes occurred from the gudgeon bearing. The rings should scrape downwards and there may be two scraper rings fitted each with two downward scraping edges, spring backed and of low inertia (figure 8.12).



▲ **Figure 8.12** *Variations of connecting rod design*

One future trend for four-stroke engines is to use LNG as a fuel

One view for the future development of the four-stroke medium-speed engine is to use LNG fuel as a solution for lower exhaust emissions. Rolls-Royce has developed its Bergen gas engine range to span powers from 1,460 kW to 7,800 kW.

Engine manufacturers are facing up to the challenge of increasingly strict requirements for exhaust emissions and Rolls-Royce is no exception. There is growing pressure to reduce CO_2 and IMO Tier II regulations on NOx emissions will be superseded by much tougher Tier III limits in 2016.

B35:40 and C26:33 gas engines have NOx emissions lower than the strict Tier III limits and net CO_2 equivalent emissions (which also take into account methane slip) and are about 22% less than an engine burning diesel fuel, with negligible SOx.

For many applications the gas engine is a natural choice but acceptance was retarded by complexity of safety rules and lack of LNG bunkering infrastructure. An acceptable regulatory structure is now in place and the infrastructure is being filled out. With the price difference between liquid fuel and LNG increasing, the case for gas is becoming even stronger.

The market for gas engines is advancing. Bergen gas engines in marine applications have now accrued more operating hours in vessels as diverse as Rolls-Royce ships, feed supply vessels, ferries and offshore supply vessels that are now equipped with Bergen gas engines. Gas tanks and the gas supply system to the engine are established technology, within the Rolls-Royce scope of supply.

The C26:33 series combines well-proven Rolls-Royce lean burn gas engine technology with the main mechanical components of the compact C25:33 diesel engine range. The first-generation engines will be produced with six, eight or nine cylinders in line, and an introductory power range from 1,469 to 2,430 kW at 900/1,000 rev/min for generator and mechanical drive applications.

CO_2 equivalent emissions are reduced by 22% compared with engines burning liquid fuel, NOx emissions are cut by 92% while emissions of SOx and particulates are negligible. The design of the C26:33 cuts methane slip, which has been seen as a disadvantage of gas engines, to very low levels. The engine meets both IMO Tier III and the forthcoming emission limits for SOx.

With the BV35:40 and C26:33 gas engines in service, Rolls-Royce developed an inline version of the B35:40 to complete a seamless range of Bergen marine gas engines spanning power requirements from 1,460 kW to 7,800 kW. The new C26:33 takes over from the K-series gas engine, which proved highly successful both on land and in pioneering marine applications, going through four generations before reaching its limit of development.

There is a growing awareness and discussion about the feature of IC engines know as methane slip. Driven by the fact that methane is more than 20 times more effective at global warming than CO_2 the subject has the potential to be fuelled by sentiment rather than a study of the facts.

Methane (CH_4) slip is the pheromone where some of the methane from the fuel moves through the engine and out of the exhaust without being burnt. Some manufacturers rather than the Diesel cycle. However, the industry is confident that as the mechanics of the methane slip become better understood, so changes in combustion design will reduce the problem. Some suggestions for how methane can by-pass the combustion process include being injected early or late in the combustion cycle and the gas is therefore caught in the scavenge port and gets sucked though during the overlap period. Another possibility is that the air/gas mix in the Otto cycle can be caught just above the piston ring where it remains unburnt and escapes with the exhaust.

It then follows that older, fuel oil, combustion space designs could be more prone to these imperfections than would new engines that are designed with methane slip in mind. It also follows that any reduction in fuel injection performance could make the situation worse.

As the engine design improves so will the combustion efficiency and therefore less unburnt fuel will pass through the process making the modern purpose built 'gas' engine less and less prone to methane slip. The wider industry view is that methane slip is a real issue but is only part of the issue CIMAC discussions focused on reducing all engine emissions and not looking at any one part in isolation.

Using gas as a fuel reduces the CO_2 considerably, cuts the NOx by 90% and reduces the SOx emissions to practically zero according to the in-service experience of Rolls-Royce who now have in excess of 30,000 h operational experience from which to draw upon. In the face of so much saving of emissions a temporary small amount of methane is a good transient solution.

9 ELECTRIC PROPULSION SYSTEMS

Introduction

Electric propulsion systems are gaining popularity as the flexible propulsion system for merchant vessels. However, it is now required that much more knowledge about these systems is gained by the operational engineering staff required to look after the machinery on a daily basis.

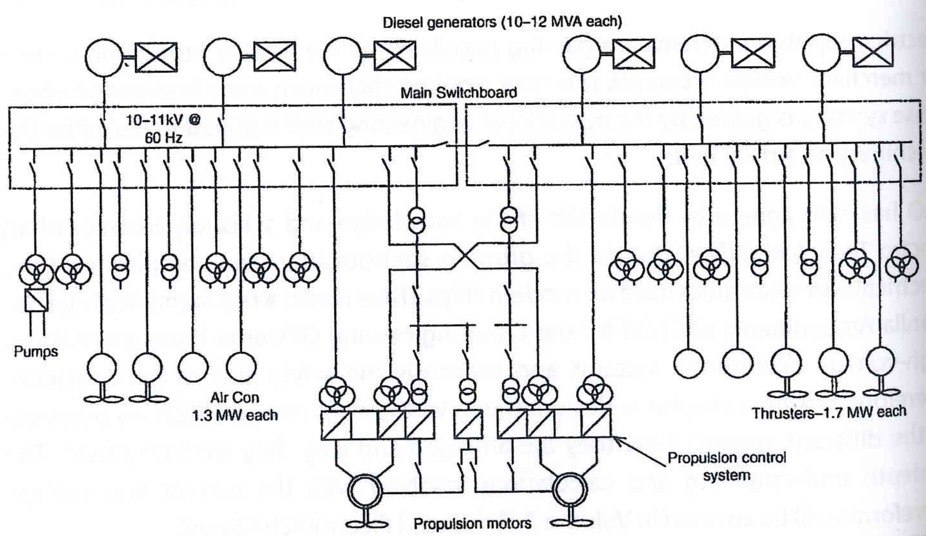
IMO has now agreed to the details of the knowledge and skills requirement of an Electro Technical Officer due to the growing sophistication and complexity of the electrical and electronics used on modern ships.There is also a requirement under the Manila Amendments of STCW for the Chief Engineering Officer to know more about high-voltage distribution systems and especially the safety side of such systems. Therefore, this new chapter is designed to give the engineering officer an overview of the different systems, how they are arranged and why they are being used. The in-depth understanding and calculations involved with the current and voltage waveforms will be covered in Volume 7, 'Advanced Electrotechnology'.

General Arrangements

There have been diesel electric and turbo-electric marine propulsion systems in the past.The P&O passenger liner Canberra, for example, had a turbo-electric drive system fitted as new when the vessel was built in 1961 and the Queen Elizabeth II was re-engined in 1987, and a diesel electric drive system replaced the original conventional steam turbine shaft line drive.

The initial driving force behind fitting an electric drive system to a ship was the flexibility in propulsion plant layout. This was particularly relevant for the passenger liner/cruise ship part of the business as the propulsion plant could be arranged so that additional 'revenue earning' passenger space could be accommodated for a given power output.

The second consideration, again initially for cruise ships, was the adoption of the 'power station' principle of operation. The electrical load of the passenger ship is considerable, even without electric drive motors. Therefore, it is often more economical for a vessel to be able to call upon a number of smaller generators, that can be matched to the load, than it is to have a smaller number of larger generators running at part load. When electric propulsion motors are added to the design of the vessel then the advantages of the 'power station' principle are even greater (figure 9.1).



Refinements to the different systems are happening all the time as the advancement in technology takes place. The developments in power electronics has allowed considerable efficiency gains and as different energy sources start to become more effective the idea of the direct current (d.c.) bus transmission system allowing a 'plug-n-play' style of system is becoming more relevant.

In the past d.c. motors and control systems provided excellent speed control with electrical drive systems. However d.c. motors are complicated, heavy and need more power than the equivalent-sized a.c. machine. The problem is that a.c. motors rely on the frequency of the supply system to operate. The rotor of the a.c. motor follows the sinusoidal waveform of the supply which is determined by the frequency of the a.c. system. This means that until recently a.c. motors have been single-speed machines.

This might not be so much of a problem if an a.c. propulsion motor is coupled to a CPP which would then be used to provide the variable propulsion required to manoeuvre the vessel. There is however considerable energy saving potential in reducing the speed of the motor when the full speed is not required.

To enable a reduction in the speed of an a.c. motor the frequency of the electrical supply must also be reduced and there are now a number of different methods that are being used to control the speed of marine propulsion motors. The method that is currently the most popular in the systems is called the PWM. Here the voltage between the different phases of the supply is switched on and off, or modulated, at high speed. This switching changes the wave form of the flux density which in effect changes the magnetic field set-up within the motor and alters the speed. The PWM control comes from a variable frequency inverter.

Harmonics is the term used to describe a distortion in the behaviour of rapidly changing physical quantities such as noise and electricity. Pure notes are noises vibrating at a given frequency but when we talk about 'harmonics' we are describing the overtones of the pure note produced by some interference or distortion. In electrical systems the distortion can be extensive and accumulative and therefore the fundamental frequency of 60 Hz would have a secondharmonic of 120 Hz and a thirdharmonic of 180 Hz. The values can keep on rising where the number of the harmonic distortion is multiplied by the fundamental frequency to give the distortion value,for example, the tenthharmonic will be 600 Hz.

In speed control circuits of a.c. propulsion motors the high-speed switching action of the electronic components, in the power converters, will cause a harmonic distortion of the original 'pure' waveform of the original supply from the generators.

In marine electrical installations, electric variable speed drives are the main load on the system and therefore the harmonic disturbance of the fundamental frequency does, in turn, have an affect on all the connected loads regardless of their position in the system.Symptoms of harmonic distortion in the electrical system are as follows:

- Occasional unexplained occurrences, such as:
  - o flickering lights
  - o alarms sounding
  - o fuses, circuit breakers and earth leakage devices tripping for no apparent reason
  - o cables running hot
  - o hot switchboards
  - o overheating motors
  - o frequent need to replace your motor's bearings andinsulation.

Some of the common and unpredictable effects of excessive harmonic distortion on marine installations include:

- Overheating and sustained damage to bearings, laminations and winding insulation on generators, transformers and induction motors causing early life failure, which could potentially result in fire.
- Overheating of the stator and rotor of fixed speed electric motors; risk of bearing collapse due to hot rotors. This is especially problematic on explosion-proof motors with increased risk of explosion, more especially with ExN (non-sparking motors).
- Overheating of cables and additional risk of failure due to resonance. Harmonics also decrease the ability to carry rated current due to 'skin effect', which reduces a cable's effective CSA.
- Disruption in the operation of uninterruptible power supplies (UPS).
- Spurious tripping or failure of sensitive electronic and computer equipment, measurement and protection relays.
- Voltage resonances leading to transient overvoltage and overcurrent failures in the electrical network.
- Electromagnetic interference (EMI) resulting in disruption to communication equipment.
- Malfunction of circuit breakers and fuses.

Total Harmonic Distortion (THD)

Harmonic distortion can be multiples of either the voltage or current waveforms and the THD is a term used to describe the contribution of all the harmonic waveforms in the electrical power generation and distribution system. It is expressed as a percentage of the ratio of the root mean square (RMS) value of the total harmonic content to the RMS value of the fundamental frequency.

Lloyd's Register rules on harmonic distortion of voltage state are as follows. Unless specified otherwise, the THD of the voltage waveform at any a.c. switchboard or sectionboard is not to exceed 8% of the fundamental for all frequencies up to 50 times the supply frequency and no voltage at a frequency above 25 times supply frequency is to exceed 1.5% of the fundamental of the supply voltage. All other classification societies place a limit of 5% on THD of voltage (THDv). The Institution of Electrical and Electronic Engineers' (IEEE) Recommended Practice for Electrical Installations on Shipboard (IEEE Standard 45–2002) states:

A dedicated propulsion bus should normally have a voltage total harmonic distortion of no more than 8%. If this limit is exceeded in the dedicated propulsion bus, it should be verified by documentation or testing that malfunction or overheating of components does not occur. A non-dedicated main generation/distribution bus should not exceed a voltage total harmonic distortion of 5%, and no single voltage harmonic should exceed 3%.

IEC 60034–1, 2004, Rotating Electrical Machines – Part 1: Rating and Performance, requires that the THDv for synchronous motors above 300 kW output should not exceed 5%. It does not specify distortion levels for individual harmonics. However, keeping low THD values on a system will further ensure proper operation of equipment and a longer equipment lifespan.

Importance of mitigating THD

There are several methods used to counter the effects of harmonic distortion in marine power systems, including:

- Active or passive filters.
- Increasing the number of pulses in power converters by using multiple-phase shifted secondary windings in propulsion motor supply transformers.

The predominant harmonics that are expected to occur in the electrical power conversion systems are calculated at the design stage.

Keeping low THD values on a system will further ensure proper operation of equipment and a longer equipment life span.

Power Quality Measurement

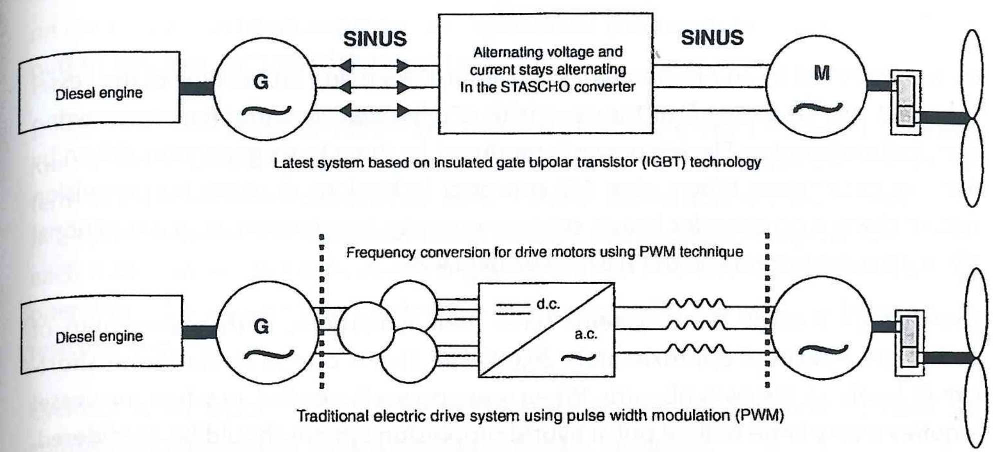
Land-based utilities monitor their power quality as a matter of routine. In a marine vessel where harmonic distortion has the potential to disrupt its electrical network, possibly leading to a blackout and loss of control in restricted waters, the need for power quality surveillance is even more significant.

Regular monitoring of power quality, using a predetermined pattern of propulsion motor loading, with a complete record of operational parameters, would help ensure that the harmonic distortion levels on-board are closely monitored as the vessel and its equipment age and operating configurations change.

An online monitoring system, that records all the parameters and can be triggered to make specific recordings of transient voltage spikes or resonances, would be invaluable in assessing the on-going quality of power. It would also be a very useful tool to investigate the root cause of accidents caused by anomalies in the electrical network and to identify incipient faults in these systems.

PWM control strategies were introduced in the early 1980s to overcome the heating and torque pulsations of the then 'square wave drives' (also known as 'quasi-square wave' or 'six-step drives'). The purpose was to reduce the output harmonics, especially the low-order harmonics, to the motor. Since that time, the various PWM strategies have been improved significantly such that present series of drives usually have output current waveforms (i.e. not the output voltage) which are relatively sinusoidal. This was achieved due to a combination of PWM techniques and advances in fast power semi-conductors such as insulated gate bipolar transistors (IGBTs). One example of this system is included in figure 9.2.

Such systems are termed active front end (AFE) systems. Compared to similarlyrated conventional six-pulse a.c. PWM drives the AFE drive has significantly higher conducted and radiated EMI emissions, and therefore, special precautions and installation



▲ **Figure 9.2** *IGBT technology a.c. electric propulsion drive system*

Advantages of the electric and hybrid drive systems

Rolls-Royce has designed and equipped many vessels with various combinations of diesel-electric, gas-electric and hybrid propulsion solutions for offshore support vessels and related multi-role vessels that demonstrate substantial fuel savings and reduced emissions compared with mechanical systems. This system is particularly effective where that is a requirement for the sudden acceleration of a diesel engine. Vessels such as tug boats, vessels fitted with dynamic positioning and vessels working extensively in ice. The advantages of the electric and hybrid drive systems include:

- Up to 50% reduction in fuel consumption resulting in reduced NOx/CO_2 emissions compared to diesel-mechanical propulsion.
- High levels of flexibility and redundancy in the configuration of the propulsion system and electrical plant.
- Fast AFE frequency control allows rapid and easy manoeuvring of the vessel.
- Up to 30% reduced maintenance due to fewer running hours and less mechanical stress when using frequency control drives.
- Reduced noise and vibration giving greater crew comfort.
- No need for heavy, space-consuming transformers.

A modern electric propulsion system today uses AFE technology. The system consist motor controlled by an AFE frequency converter. Depending on the type of the vessel there will also be two to four thrusters with AFE frequency control for manoeuvring and position keeping. Electric power is produced by three to six generators driven by diesel or gas engines. When using AFE converter technology in an electric propulsion system there is no need for heavy, space-consuming transformers as in a traditional 12- or 24-pulse system and the THD will be below 2%.

Low-voltage electric drive systems have their limitations, with a maximum of power generation of approximately 20,000 kW. This is due to limitation of short-circuit levels on the switchboards. When more power is needed, or when the vessel requires a very large bollard pull a hybrid propulsion system should be considered. Efficient hybrid propulsion systems combine mechanical and diesel/gas electric transmissions. Individual systems are tailored to the vessel and its operating profile in terms of total installed power and how much of this power travels the diesel electric route.

A typical vessel with the Rolls-Royce system will have a twin screw layout with CPP. Each shaftline comprises a medium-speed engine, a reduction gearbox and a clutch between gearbox and engine. At the forward end of each main engine are a second clutch and a large shaft generator. A frequency-controlled variable speed electric motor is connected to a power take-in (PTI) drive in the gearbox. In addition to the two main shaftlines there are two or more auxiliary gensets and there are also tunnel and azimuth thrusters to assist in manoeuvring and positioning.

Efficiency under all operating conditions is the defining principle, achieved by running only the number of engines actually required, and avoiding having powerful engines operating at low part loads with a resulting high specific fuel consumption. Further, utilisation of frequency-controlled electric motors eliminates the zero-pitch propeller losses which may become significant for long periods of operations at low load. At the same time the energy losses associated with electric transmissions are reduced at higher powers by routing all or most of the propulsion power through a low-loss mechanical transmission.

Cycloconverters are a common form of electrical variable speed drive in the higher power range and, as such, are used for main propulsion drives. Unlike other forms of a.c. drives, such as a.c. PWM drives and load commutated inverters (LCI), both of which have an intermediate stage (i.e. d.c. bus) to facilitate dual conversion (a.c. to d.c. and d.c. to a.c.), the cycloconverter is a direct conversion drive converting one frequency ... without the need for an intermediate stage. Cycloconverters have been in

on mercury arc rectifiers and they have operational constraints by having a maximum output frequency of 33% of the input frequency.

Developments of speed control of synchronous induction motors, known as the Static Kramer drive, uses a cycloconverter to further enhance the system under the new term of 'static Scherbius' drive. When used in power conversion systems the operation of a cycloconverter is complex with both positive and negative bridges necessary for each motor phase. To briefly describe their operation it is necessary to consider the operation of a single-phase-to-single-phase device with full-wave rectifiers and a resistive load.

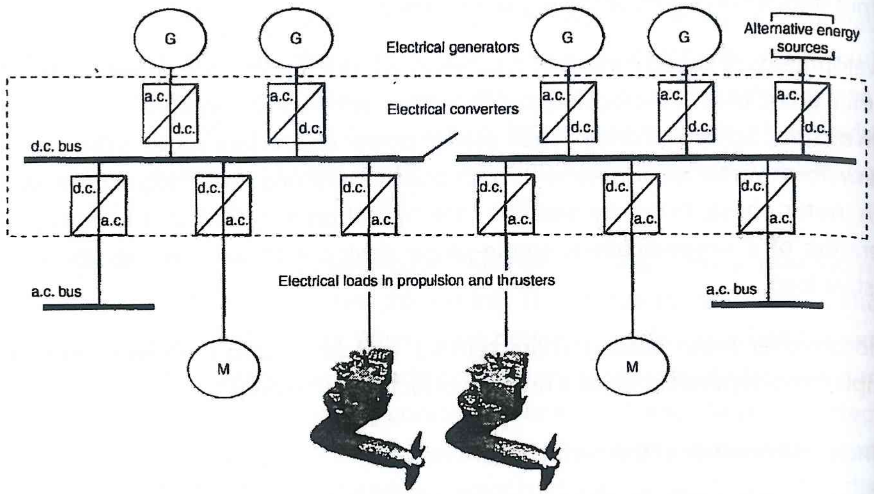
Cycloconverter input current characteristics and associated harmonic content are complex and dependent upon a number of factors, including:

- The pulse number of the cycloconverters
- The relative magnitude of the output fundamental voltage
- The ratio of the input and output frequencies
- The displacement power factor of the load
- The firing control strategy.

In applications with large drives, six-pulse drives are not common. Multi-pulse drives, including 12-pulse, are the norm to minimize the input harmonic currents and associated disruption of the power supply system. One development of the a.c. motor control is based upon IGBT transistors, thyristors and by-pass switching.

The basic reason for the adoption of electric drive systems is to improve the efficiency of converting the energy from the fuel into useful propulsive and power generation. The more energy conversion steps in the line the more potential there is for losses.The use of a d.c. grid or distribution system is an effort to reduce the number of changes in energy between the fuel and the use of the power output.

For example, converting the fuel directly into thrust is not yet possible. Using mechanical components only is inefficient and as we have seen earlier in this chapter there are a large number of changes involved in the a.c. drive system using the PWM or AFE system where we have fuel converted into mechanical force and then through the magnetic field to an a.c. output; next, it is changed to another a.c. frequency then to d.c. and back to a.c. before being converted once again to mechanical power and finally trust. Much of these changes can be removed with the use of the d.c. grid as shown in figure 9.3.



▲ **Figure 9.3** *On-board d.c. grid (electric propulsion drive system)*

10 WASTE HEAT RECOVERY SYSTEMS

General Details

Reference should be made to Chapter 1 for general comments relating to heat balance. Figure 1.3 details an approximate heat balance for an IC engine showing significant losses to the exhaust and cooling. Every attempt made to utilise energy in WHR from both exhaust and coolant is established practice. Sufficient energy potential can be available in exhaust gas at full engine power to generate sufficient steam, in a waste heat boiler, to supply total electrical load and heating services for the ship. The amount of heat actually recovered from the exhaust gases depends upon various factors such as steam pressure, temperature, evaporation rate required, mass flow of gas, condition of heating surfaces, etc. Waste heat boilers can recover up to about 60% of the loss to atmosphere in exhaust gases. Heat recovery from jacket cooling water systems at a temperature of 70–80°C is generally restricted to supplying heat to the fresh water generator.

Combustion equipment

Most modern ships have boiler arrangements for raising steam. A thermal fluid system alternative is available but the preferred system is steam auxiliary boilers. During low engine power conditions or when the main engine is not in use the boiler has to combust fuel to provide the heat source. It is therefore appropriate to repeat some very general remarks on combustion with details of the typical boiler equipment in use on-board ship. A more detailed explanation of boilers appears in chapter 3 of Volume 8 of this book series.

Good combustion is essential for the efficient running of the boiler as it gives the best possible heat release and the minimum amount of deposits upon the heating surfaces. To ascertain if the combustion is good we measure the % CO_2 content (and in some installations the % O_2 content) and observe the appearance of the gases.

If the % CO_2 content is high (or the % O_2 content is low) and the gases are in a non-smokey condition then the combustion of the fuel is correct. With a high % CO_2 content the % excess air required for combustion will be low and this results in improved boiler efficiency since less heat is taken from the burning fuel by the small amount of excess air. If the excess air supply is increased then the % CO_2 content of the gases will fall.

Condition of burners, oil condition pressure and temperature, condition of air registers, air supply pressure and temperature are all factors which can influence combustion.

Burners

There are two basic types of burners, the pressure jet and the rotary cup. The pressure jet as its name suggests relies on the fuel oil supply pressure to force the fuel through a series of small nozzles in the end of a long tube. The holes are set at an angle and will therefore give a spin to the fuel as it exits from the burner. This spin or swirl gives the fuel the right action to mix thoroughly with the air delivered by the air register and therefore when the mixture hits the flame front it is ignited. The rotary or spinning cup type of burner does not rely on the fuel pressure to give atomisation. The low-pressure fuel oil is released into the centre of the rotary cup that is spinning at about 5,000 rev/min. As the fuel follows the conical cup it eventually comes to the rim where centrifugal

are dirty or the sprayer plates damaged then effective atomisation will not be achieved, resulting in poor combustion.

Oil

If the oil is dirty it can foul up the burners. (Filters are provided in the oil supply lines to remove most of the dirt particles but filters can get damaged. Ideally the mesh in the last filter should be smaller than the holes in the burner sprayer plate.)

Water in the oil can also affect combustion, it could lead to the burners being extinguished and a dangerous situation arising. It could also produce panting (unstable combustion leading to pressure fluctuations) which can result in structural defects.

If the oil temperature is too low the oil does not readily atomise since its viscosity will be high, this could cause flame impingement, overheating, tube and refractory failure. If the oil temperature is too high the burner tip becomes too hot and excessive carbon deposits can then be formed on the tip causing spray defects. These could again lead to flame impingement on adjacent refractory and damage could also occur to the air swirlers. Oil pressure is also important since it affects atomisation and lengths of spray jets.

Air register

Good mixing of the fuel particles with the air is essential, hence the condition of the air registers and their swirling devices are important, if they are damaged mechanically or by corrosion then the air flow will be affected.

Air

The combustion air supply is governed by the combustion controller fuel/air ratio setting. If this is set too low then insufficient air will be supplied resulting in incomplete combustion and the generation of black smoke. If the fuel/air ratio is set too high then too much air will be supplied for combustion resulting in a greater percentage of free oxygen in the uptakes than is desirable, causing the boiler efficiency to fall.

It is generally considered that the appearance of the boiler uptake gases will give an accurate indication of the effectiveness of combustion. While this is undoubtedly true air, resulting in a reduction in boiler efficiency. To achieve maximum boiler efficiency the fuel/air ratio setting should be reduced until the setting for optimum combustion, commensurate with clear uptake gases, is reached.

Boiler operation

Boilers are potentially one of the most dangerous places in the engineroom of a ship. For this reason the Flag State examiner issuing a certificate of competency to a motor ship marine engineer will not do so unless she/he is sure that the marine engineer can cope with the dangers of the steam raising plant.

Auxiliary boilers on modern ships are usually fire tube boilers operating with a working steam pressure of about 7 bar. This is enough pressure to supply all the necessary heating required on-board the vessel. However, because the fire tube boiler has a relatively large amount of water, for the size of boiler, it also has a greater potential for causing a lot of damage if there was a structural failure.

It must be remembered that if the steam is at 7 bar pressure then any parts in contact with the steam are also at 7 bar pressure. If the pressure on the water within the boiler was suddenly reduced to atmospheric pressure, due to some form of structural failure, then the water would flash off into steam.

Steam requires 1,600 times the volume of water; therefore, when the pressure is released and the water flashes into steam a considerable force is released and large sections of the boiler can be moved at considerable speed.

One of the most important dangers to guard against is loss of water. The metal furnace close to the burner relies upon the cooling effect of water on the other side of the furnace to ensure that it does not overheat and fail. There are a number of reasons for a loss of feed water and the motor engineer will need to understand his/her system and be able to explain to the examiner how to guard against a loss of water in the auxiliary boiler.

The gauge glass is the primary source of information about the water level in the boiler. There are always two gauge glasses just in-case one becomes blocked, however on a marine boiler they are situated on opposite sides of the boiler. The reason for this is that if the vessel is rolling and one glass is empty then the other should be showing a high level and vice versa. More details can be found in Volume 8 of this series.

sometimes temperamental in their operation they have to be opened, cleaned and adjusted from time to time. Oil builds up around the furnace front and can be the cause of a fire if engineers are not careful.

The watchkeeper will be responsible for the safe operation of the auxiliary boiler during his/her watchkeeping duty period; therefore, it is essential that she/he understands the following safety-related start-up and operating procedure. Safe start-up procedure involves a purge cycle. This means that the boiler will run the forced draught fan for a few seconds before trying to light the boiler. This is to ensure that any unburnt hydrocarbons from the previous cycle are taken away from the burner that will be lighting up soon.

With the air operating correctly the next step is to introduce the heat source. This is usually in the form of a high voltage passing from one electrode to another via an air gap (a bit like the spark in a car). When the heat and the oxygen are both in place it will then be okay to introduce the fuel and start the burn. Feedback that to the boiler controls saying that the boiler is alight is provided by a photoelectric cell. Using this sequence there is very little chance of unburnt gasses accumulating in the furnace and causing a violent start-up or an explosion.

The watchkeeping engineer could be called upon to start a boiler that has 'locked out'. It would only do this if it had failed to light for some reason. It is very important that the watchkeeper carries out some basic checks before trying to relight the boiler.

The first and most important check is to look at the water level in the boiler and make sure that the boiler has not 'locked out' due to low water level. Make sure that you look at both gauge glasses and if you are unsure you need to follow the gauge glass check procedure described in chapter 3 of Volume 8 in this book series. If a low water level is suspected then it is very important that you DO NOT start the burner until the correct level is restored. The problems here could range from failure of the feed water pump to deliver a sufficient quantity of water to a malfunction of the feed water control float that dictates when and how much water is sent to the boiler.

The feed water pump operates under difficult conditions because not only does it have to pump water that is close to its boiling condition but it also has to pump it at sufficient pressure to overcome the boiler's working pressure. Sometime these pumps 'gas up', in other words, the water's vapourisation condition is met and it starts to turn to steam inside the pump stopping the flow of water.

The other, not uncommon, problem is with the float sticking in the feed water controller. I have seen some of these corroded so much that the float has broken away from the

the problem then the boiler can be started again following the restoration of the feed water to the boiler.

The watchkeeper should monitor the boiler as it works through its safety purge cycle and then the starting sequence described earlier. Some boilers are fitted with a burner viewing port which is to be treated with great respect. These should never be used when the boiler is starting up as serious injury has been caused in the past due to 'blow back' as some unburnt fuel has caught alight.

A malfunction in the burner starting sequence is another reason for the boiler 'locking out'. If the boiler fails to light – some control systems may allow two cycles before giving an alarm – then the watchkeeper will need to identify the reason and rectify the fault.

If the 'lock out' is due to the burner then there are three conditions to check:

1. Boiler fan working to give the correct amount of air to the burner
2. Igniter working and in the correct position
3. Fuel supply to the burner nozzles or spinning cup.

Conditions 1 and 3 are relatively easy to check because the fan not working or the air register blocked will be easy to spot as will be the lack of fuel. Igniter faults and subsequent set-up however is generally more difficult to deal with but is a more common fault than the other two.

Sometimes there could be quite a bit of pressure to get the boiler back online for example. If the heavy fuel oil to the generator was cooling down due to lack of steam there might not be much time in which to have the boiler up and running again or to change the generator over to a light distillate fuel.

The problem with the igniter is that it has to sit right in the most turbulent area of the air/fuel mix and the forces involved are enough to knock it out of alignment occaisionally and therefore it stops working. It has to be reset to the correct position which is just where the air/fuel ratio is correct for combustion. If it is placed too close to the burner the mixture is too rich for ingintion and if it is placed too far away the mixture could be too lean or it might miss the spray of fuel entering the combustion space. Then, the burner will cut-out due to the length of time trying to light the fuel.

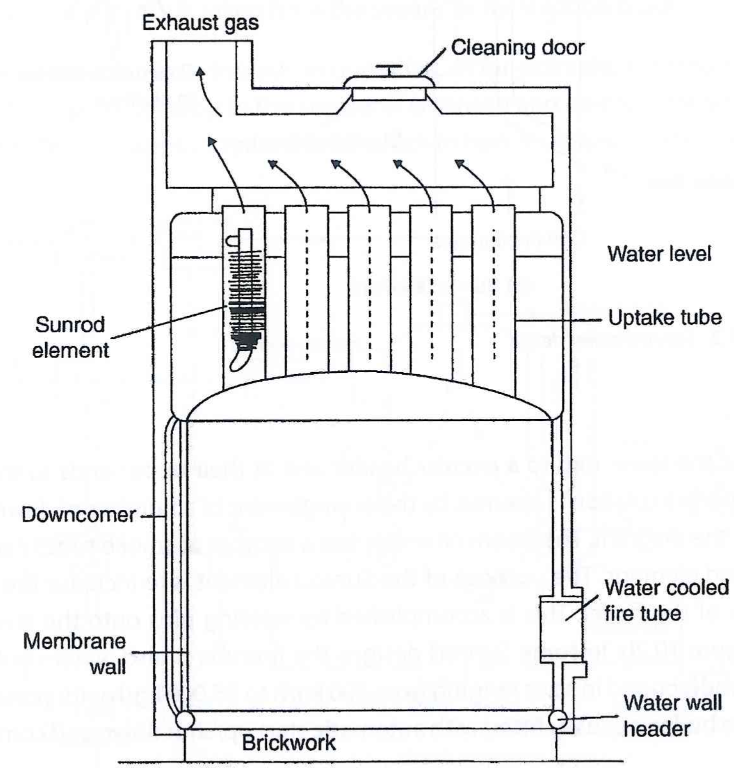
Generally the engineers will know the settings and be able to work away at getting

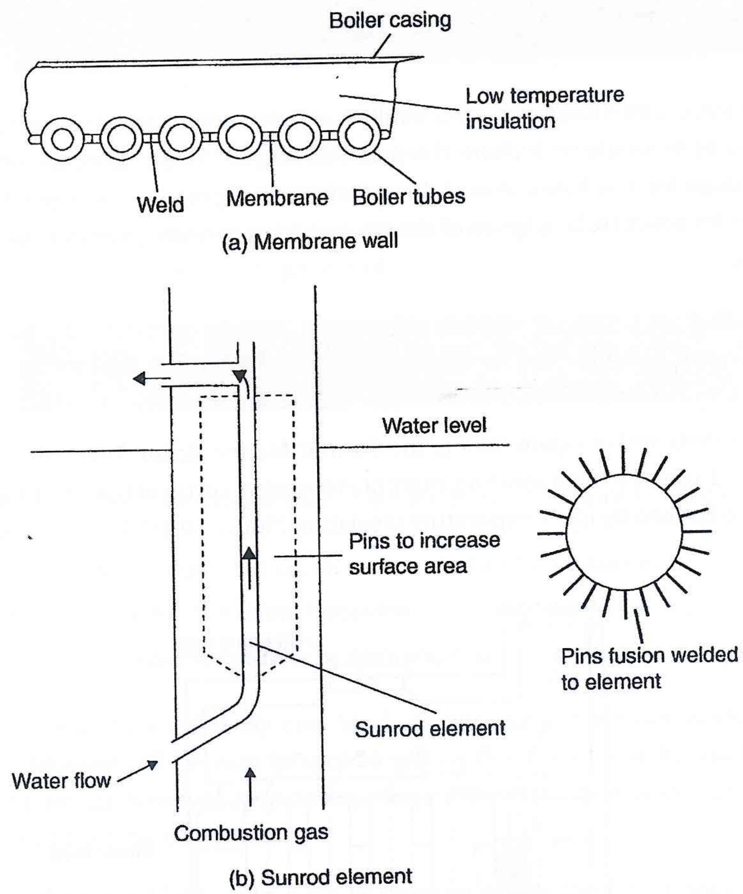
Package Boilers

Although such boilers are not necessarily involved with waste heat systems it is considered appropriate to include them at this stage. These boilers are often fitted on motorships for auxiliary use and the principles and practice are a good lead into general boiler practice. Two types of design involving modern principles will now be considered.

Sunrod vertical boiler

The design sketched in figure 10.1 is the Sunrod Marine Boiler. This boiler utilises a water-cooled furnace incorporating membrane-walled construction. The membrane water wall is backed by low temperature insulation (figure 10.2a). The water wall tubes





▲ **Figure 10.2** *Sunrod boiler detail*

are joined at the lower end to a circular header and at their upper ends to the steam chamber. Good circulation is assured by the arrangement of a number of downcomers as shown in the diagram. The steam chamber has a number of smoke tubes each fitted with a 'Sunrod element'. The purpose of the Sunrod element is to increase the heating surface area of the boiler. This is accomplished by welding pins onto the element as shown in figure 10.2b. In some Sunrod designs the firetube is also water-cooled. This design is manufactured in sizes ranging from 700 kg/h to 35,000 kg/h with pressures up to 18 bar. The boiler is usually fitted with automatic start up/shut down and combustion control.

when the boiler is shut down, by simply removing the cleaning doors, opening the drain and spraying with high-pressure fresh water.

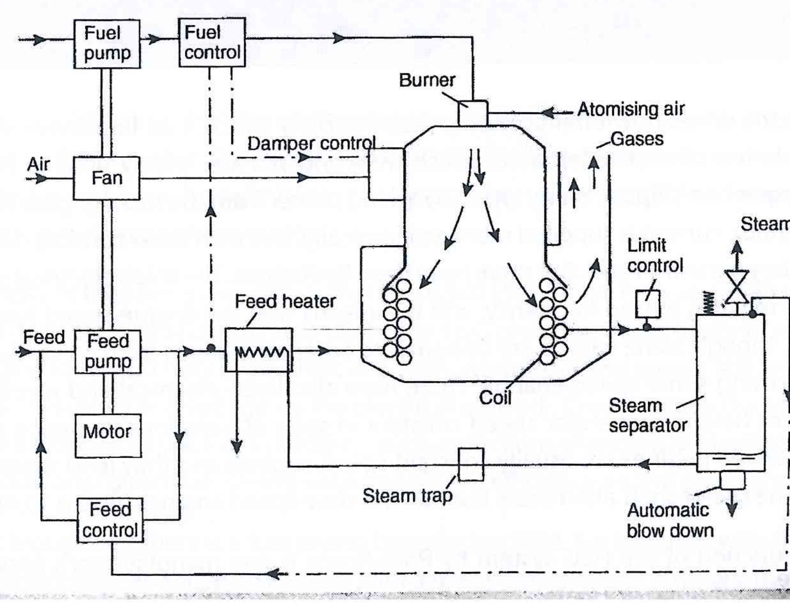
Pressure control of the steam is accomplished by flashing the boiler when pressure drops below a preset level during periods of high steam load and, dumping steam to the condenser when the pressure rises due to low steam load.

Vapour vertical boiler (coiled-tube)

Figure 10.3 shows in a simplified diagrammatic form a coiled-tube boiler of the stone-vapour type. It is compact, space saving, designed for UMS operation, and is supplied ready for connecting to the ships services. A power supply, depicted here by a motor, is required for the feed pump, fuel pump (if fitted), fan and controls.

Feed water is force circulated through the generation coil wherein about 90% is evaporated. The un-evaporated water travelling at high velocity carries sludge and scale into the separator, which can be blown out at intervals manually or automatically. Steam at about 99% dry is taken from the separator for shipboard use.

The boiler is completely automatic in operation. If, for example, the steam demand is increased, the pressure drop in the separator is sensed and a signal, transmitted to the feed controller, demands increased feed, which in turn increases air and fuel supply.



With such a small water content explosion due to coil failure is virtually impossible and a steam temperature limit control protects the coil against abnormally high temperatures. In addition the servo-fuel control protects the boiler in the event of failure of water supply. Performance of a typical unit could be:

- Steam pressure = 10 bar
- Evaporation = 3,000 kg/h
- Thermal efficiency = 80%
- Full steam output in about 3–4 min.

Note: Atomising air for the fuel may be required at a pressure of about 5 bar.

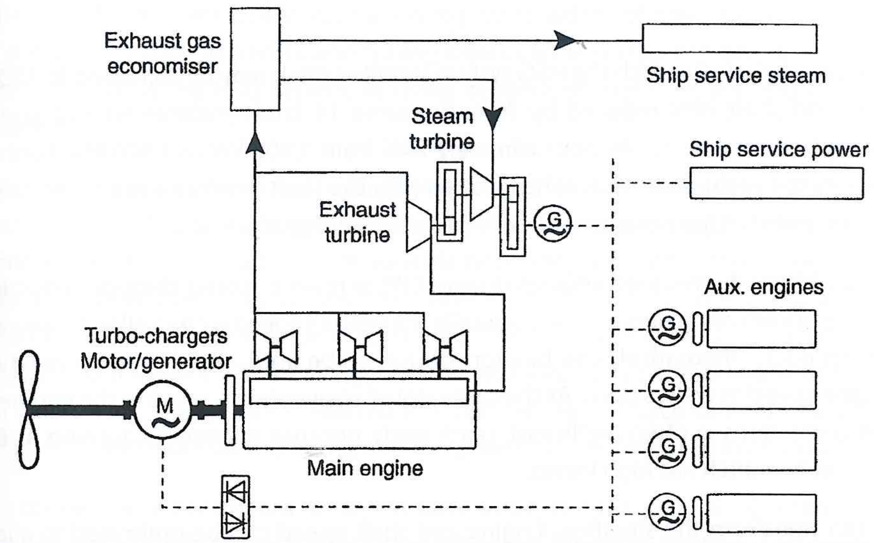
Steam-to-steam generation

In vessels which are fitted with water tube boilers a protection system of steam-to-steam generation may be used instead of desuperheaters and reducing valves, etc. (see later).

Hybrid and Power Take-Off and Take-In Systems

New electric drives now offer economy with flexibility which is an improvement over the simple use of shaft alternators which were and still are, widely used to provide electric power on ships of many types. By taking power from the main engine to drive the generator, current is supplied more economically, and with fewer running hours on the auxiliary generator sets. But there have been limitations. The ship's electrical system normally requires a fixed frequency, and this means that the engine speed has to be constant, though some vessels are designed to accept variations from 60 Hz down to 50 Hz allowing some speed change. There have also been electrical and mechanical systems for holding generator speed constant in spite of variations in engine speed. Medium-speed engines are usually constant speed engines and they lend themselves more to the use of shaft alternators than do the slow-speed engines (figure 10.4).

The introduction of the HSG system by Rolls-Royce is one manufacturer's answer to improve the efficiency of main engine driven alternators. HSG stands for hybrid shaft



▲ **Figure 10.4** *Wärtsilä WHR system*

and frequency, and the correct phase angle to match other generator sets running in parallel. This opens the way for much more flexible use of engine and propeller speed variations to maximise both propeller and engine efficiencies by running them at their design points. HSG gets away from the straitjacket of fixed speed. Hence, for some applications the HSG concept can give remarkable fuel reductions over a comparable diesel mechanical installation. The system also helps to reduce exhaust CO_2 and NOx emissions.

HSG can in addition control the shaft generator to enable it to act as a motor, feeding in power to the propeller. This would add flexibility to the power plant by allowing the generated power from the swithboard to move the vessel in a very limited way without running the main propulsion motor.

An example of the savings possible is that one 6,500 kW CPP will have about 900 kW loss at fixed nominal RPM and with zero pitch. The HSG gives the opportunity of reducing the engine RPM, and hence the shaft line RPM, down to idling speed, still with fixed nominal frequency and voltage on the electrical network. Consequently the zero pitch losses are reduced by 800 kW. In addition, such reductions of engine RPM will give up to 5–8% additional direct fuel saving based on higher efficiency on the diesel engine.

Even at higher load there is a fuel saving by reducing RPM. For instance, with the same example propeller as above on a vessel with a maximum speed of 20 knots, the normal and reduced pitch. But with the HSG system installed pitch may be increased to 100%, engine and shaft RPM reduced by 30%, the same 14 knots maintained, but power consumption may be cut by approximately 20% from 1,900 kW to 1,500 kW. Further, the ability to use the same HSG concept to employ the shaft generator as a power take-in motor gives further potential for optimising the energy efficiency.

Vessels with medium-speed engines driving CPP or main thrusters through reduction gears are prime candidates for HSG. Typically the vessel's speed is controlled by varying propeller pitch. Alternatively combinator control can be used, allowing some variation in engine speed as well as pitch. As the case quoted above shows, running the engine(s) at full revolutions is often inefficient, particularly because a propeller turning at full speed but low pitch has high losses.

HSG can transform the situation. Engine and shaft speed can be optimised to allow power production at its most economical, while the propeller operates at its maximum efficiency speed and pitch for the given conditions. The shaft generator continues to function down to very low shaft speeds, feeding the main switchboard and supplying the ship's electrical load, avoiding the need to run auxiliary generator sets.

Using the shaft generator as a PTI motor with the HSG concept can be attractive where a vessel may have to steam very slowly, or loiter waiting, for a place at the quay. The main engine can be shut off instead of idling, and power generated instead by a genset operating at an efficient load.

The drive is also applicable to merchant ships with direct-coupled low-speed diesel engines and fixed pitch propellers. The traditional problem with shaft generators in this type of propulsion system is that all ships' speed control is by altering engine revolutions. Even when on passage, weather conditions cause small engine speed variations, enough to cause problems with fixed frequency shaft generator systems. HSG concept allows the generator to follow the speed changes, but the electrical consumers still receive the normal voltage and frequency. Much more use can therefore be made of a shaft generator deriving its power from a big diesel engine operating with a high thermal efficiency, eliminating the need to have auxiliary gensets running continually.

Refrigerated cargo vessels can save even more, because a shaft generator can supply the power for cooling down prior to loading, probably reducing the number and size of generator sets. Offshore vessels use shaft generators extensively. For vessels with hybrid electrical/mechanical transmission the cost and complication of dedicated PTI ... HSG ... another level of flexibility to multi-mode operation.

Other vessel types with operation modes requiring widely different amounts of power for propulsion, such as yachts and fishing boats can also reap the benefits of hybrid propulsion using the HSG system, in terms of reduced fuel burn and less noise and vibration.

The HSG concept works on the basis of a two-step electrical conversion. It uses power electronics and AFE technology developed by Rolls-Royce, avoiding the need for bulky transformers. The first step is from variable frequency a.c. to d.c. The second step is from d.c. to fixed frequency a.c., with the added feature of a 'speed droop' characteristic which makes the system appear to the switchboard and other connected power suppliers as if it were a standard generator set running in parallel and sharing load in a stable way. The system is housed in a standard Rolls-Royce cabinet. The drive is suitable for 440 V or 690 V systems, and the power ranges from 100 kW to 5.000 kW.

This concept is not restricted to new vessels. Converting ships in service can be cost effective and a significant help in reducing emissions. Where an existing shaft generator is driven by a secondary PTO from the gearbox, conversion can be simple. The HSG concept can handle both synchronous and asynchronous electrical machines, so the existing generator might be retained, conversion involving installation of the drive cabinet and some switchboard alterations. Other types of vessel should be evaluated on a case-by-case basis.

Turbo-Generators

Such turbines are fairly standard l.p. steam practice and reference, where necessary, could be made to Volume 9. Steam from the exhaust gas boiler can be used to drive a Turbine. Detailed instructions are provided on-board ship for personnel unfamiliar with turbine practice. For the purposes of this chapter the short extract description given below should be typical and adequate.

Turbine

A single cylinder, single axial flow, multistage (say 5) impulse turbine provided with steam through nozzles at 10 bar and 300°C preferably with superheat to limit exhaust moisture to 12%. Axial adjustment of rotor position is usually arranged at the thrust block and protection for overspeed, low oil pressure and low vacuum are provided. Materials and construction for the turbine unit and single reduction gearing are

Electrical

The turbine at 100–166 rev/s drives the alternator and exciter through a reduction of about 6:1 to produce typically 450–600 kW at 440 V, 3 ph., 60 Hz. A centrifugal shaft-driven motorised governor arranged for local or switchboard operation would operate the throttle valve via a hydraulic servo. Straight line electrical characteristics normally incorporate a speed droop adjustment to allow ready load sharing with auxiliary diesel generators or an extra turbo unit.

Ancillary plant

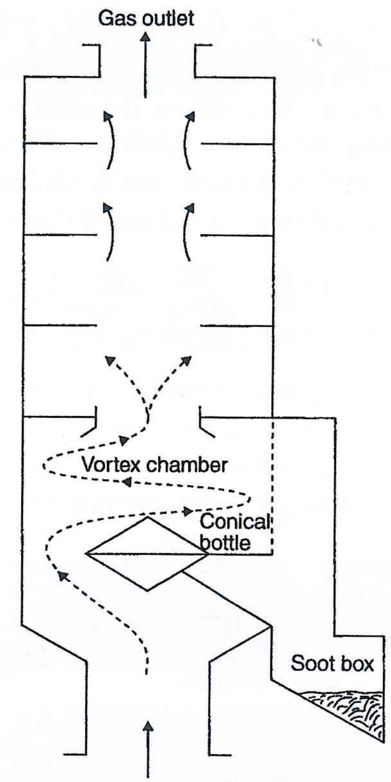
This is normally provided as a package unit with condenser, air ejector, auto gland seals, gland condenser, motorised and worm-driven oil pumps, etc. A feed system is provided either integral or divorced from the turbine-gearbox-alternator unit. Exhaust can be arranged to a combined condenser incorporating cargo exhaust. Control utilises gas by-pass, dumping steam, etc.

Silencers

Normally waste heat boilers act as spark arresters and silencers at all times. The silencer sketched in figure 10.5 would not usually be fitted if such boilers were used but a short description of the silencer may be useful.

Three designs have been utilised. The tank type has a reservoir of volume about 30 times the cylinder volume. Baffles are arranged to give about four gas reversals. The diffuser type has a central perforated discharge pipe surrounded by a number of chambers of varying volume. The orifice type is sketched in figure 10.5 and the construction should be clear. Energy pulsations and sound waves are dissipated by repeated throttling and expansion.

Gas Analysis



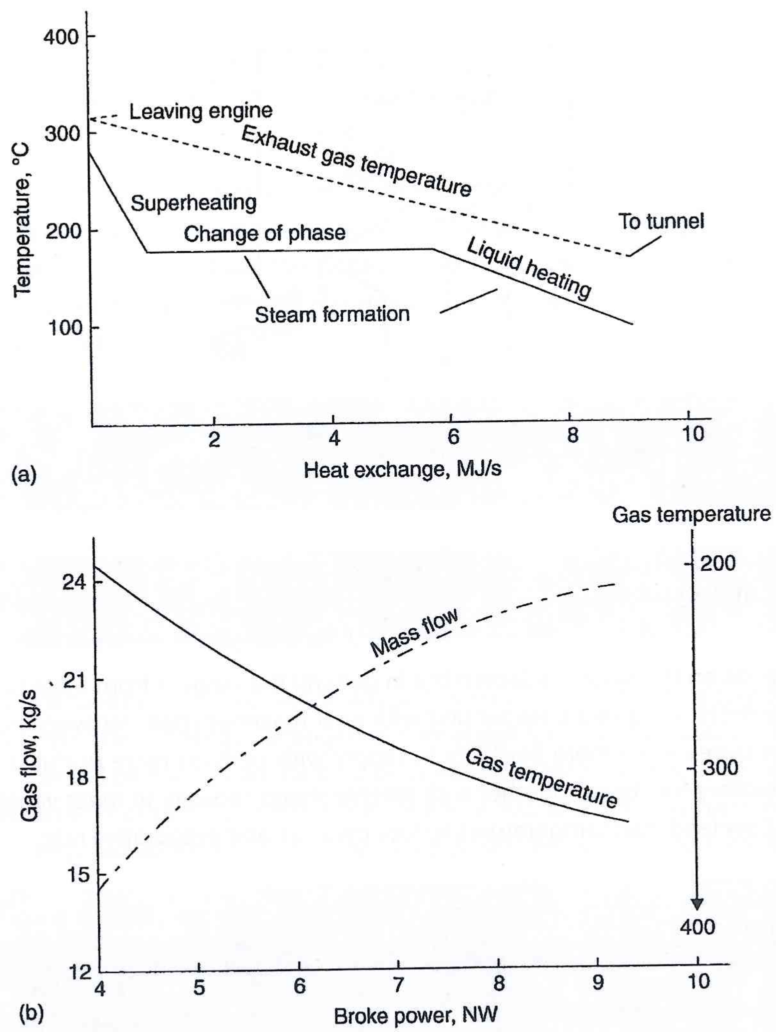
▲ **Figure 10.5** *Silencer and spark arrester*

Optimum pressure

This depends on the system adopted but in general the range is from 6 bar to 11 bar. The lower pressures give a cheaper unit with near maximum heat recovery. However, higher pressures allow more flexibility in supply with perhaps more useful steam for certain auxiliary functions together with reserve steam capacity to meet variations in demand. Low feed inlet temperatures reduce pressure and evaporative rate.

Temperature

A minimum temperature differential obviously applies for heat transfer. Temperature difference, fouling, gas velocity, gas distribution, metal surface resistance, etc., are all and temperature increase if the power is maintained constant. A similar effect will be apparent under operation in tropical conditions. The effect of increased back pressure will be to raise the gas temperature for a given air inlet temperature. Figure 10.6 illustrates: (a) typical heat transfer diagram, and (b) gas temperature/mass–power curves. A common temperature differential is about 40°C, that is, water inlet 120°C and gas exit 160°C.



Corrosion

The acid dew point expected is about 110°C with a 3% sulphur fuel and a high rate of conversion from SO_2 to SO_3 is possible. Minimum metal temperatures of 120°C for mild steel are required.

Exhaust system

The arrangement must offer unrestricted flow for gases so that back pressure is not increased. Good access is required for inspection and cleaning. On designs with alternate gas-oil firing provision must be made for quick and foolproof change-over with no possibility of closure to atmosphere and waste heat system at the same time.

Gas/Water Heat Exchangers

Waste heat economisers

Such units are well proven in steamship practice and similar all-welded units are reliable and have low maintenance costs in motorships. Gas path can be staggered or straight through with extended surface element construction. Large flat casings usually require good stiffening against vibration. Water wash and soot blowing fittings may be provided.

Waste heat boilers

These boilers have a simple construction and fairly low cost. At this stage a single natural circulation boiler will be considered and these normally classify into three types, namely: simple, alternate and composite.

Simple

These boilers are not very common as they operate on waste heat only. Single- or two-pass types are available, the latter being the most efficient. Small units of this type have in conjunction with another boiler. A gas change valve to direct flow to the boiler or atmosphere is usually fitted as described below.

Alternate

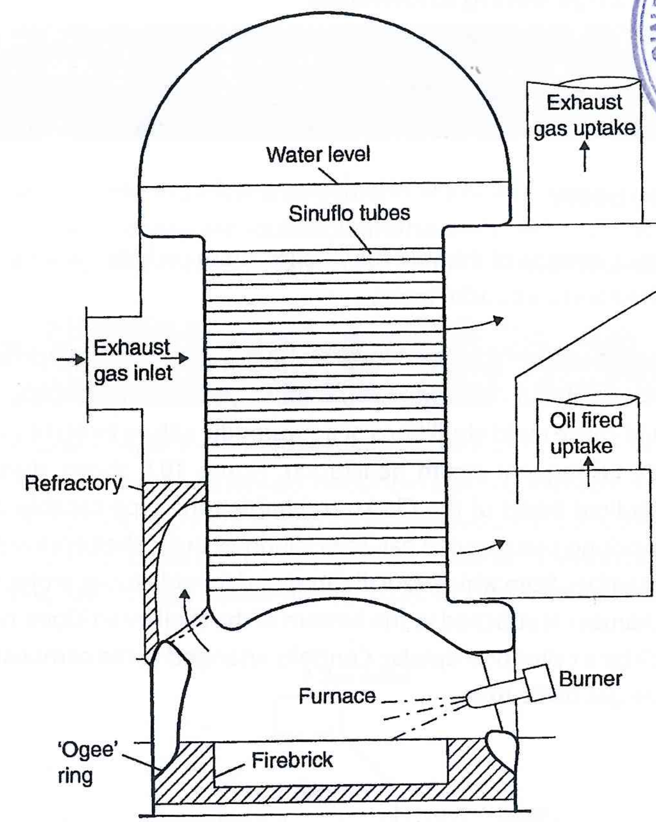
This type is a compromise between the other two. It is arranged to give alternate gas and oil firing with either single- or double-pass gas flow. It is particularly important to arrange the piping system so that oil fuel firing is prevented when exhaust gas is passing through the boiler. A large butterfly type of change-over valve is fitted before the boiler so as to direct exhaust gas to the boiler or to the atmosphere. The valve is so arranged that gas flow will not be obstructed in that as the valve is closing one outlet the other outlet is being opened. The operating mechanism, usually a large external square thread, should be arranged so that with the valve directed to the boiler, fuel oil is shut off. A mechanical system using an extension piece can be arranged to push a fork lever into the operating handwheel of the oil fuel supply valve. When the exhaust valve is fully operated to direct the gas to atmosphere the fork lever then clears the oil fuel valve handwheel after changeover travel is completed. It is also very important to ensure full fan venting and proper fuel heating-circulation procedures before lighting the oil fuel burners.

Composite

Such boilers are arranged for simultaneous operation on waste heat and oil fuel. The oil fuel section is usually only single pass. Early designs utilised Scotch boilers, with, say, a three-furnace boiler, it may mean retaining the centre or the wing furnaces for oil fuel firing. The gas unit would often have a lower tube bank in place of the furnace, with access to the chamber from the boiler back, thus giving double pass. Alternative single pass could be arranged with gas entry at the boiler back. Exhaust and oil fuel sections would have separate uptakes and an inlet change-over valve was required. In general Scotch boilers as described are nearly obsolete and vertical boilers are used. As good representative, and more up-to-date, common practice, two types of such boilers will be considered.

Cochran boiler

The Cochran boiler whose working pressure is normally of the order of 8 bar is available in various types and arrangements, some of which are as follows: single-pass composite, that is, one pass for the exhaust gases and two uptakes, one for the oil fired system and another for exhaust system; double-pass composite, that is, two passes for the exhaust



▲ **Figure 10.7** *Diagrammatic arrangement of a single-pass composite Cochran boiler*

as a simple type. Or, double pass alternatively fired, that is, two passes from the furnace for either exhaust gases or oil fired system with one common uptake.)

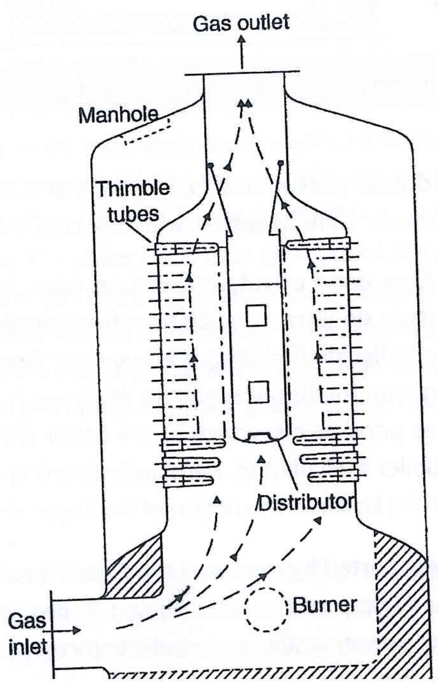
The boiler is made from good quality low carbon open hearth mild steel plate. The furnace is pressed out of a single plate and is therefore seamless. Connecting the bottom of the furnace to the boiler shell plating is a seamless 'Ogee' ring. This ring is pressed out of thicker plating than the furnace, greater thickness is necessary since circulation in its vicinity is not as good as elsewhere in the boiler and deposits can accumulate between it and the boiler shell plating. Hand hole cleaning doors are provided around the circumference of the boiler in the region of the 'Ogee' ring.

The tube plates are supported by means of tube stays and by gusset stays, the gusset stays supporting the flat top of the tube plating. Tubes fitted, are usually of special design (Sinuflo), being smoothly sinuous in order to increase heat transfer by promoting turbulence. The wave formation of the tubes lies in a horizontal plane when the tubes are fitted, this ensures that no troughs are available for the collection of dirt or moisture. This wave formation does not in any way affect cleaning or fitting of the tubes.

Thimble tube boiler

There are various designs of thimble tube boiler, these include: oil fired, exhaust gas, alternatively fired and composite types.

The basic principle with which the thimble tube operates was discovered by Thomas Clarkson. He found that a horizontally arranged tapered thimble tube, when heated externally, could cause rapid ebullitions of a spasmodic nature to occur to water within the tube, with subsequent steam generation. Figure 10.8 shows diagrammatically an alternatively fired boiler of the Clarkson thimble tube type capable of generating steam with a working pressure of 8 bar. The cylindrical outer shell encloses a cylindrical combustion chamber, from which, radially arranged thimble tubes project inwards. The combustion chamber is attached to the bottom of the shell by an 'Ogee' ring and to the top of the shell by a cylindrical uptake. Centrally arranged in the combustion chamber is an adjustable gas baffle tube.



Exhaust Gas Heat Recovery Circuits

Many circuits are possible and a few arrangements will now be considered. Single boiler units as discussed, while cheap, are not flexible and have relatively small steam generating capacity. The systems now considered are based on multi-boiler installations.

Natural circulation multi-boiler system

It is possible to have a single-exhaust gas boiler located high up in the funnel, operating on natural circulation whereby a limited amount of steam is available for power supply while the vessel is at sea. In port or during excessive load conditions, the main boiler or boilers are brought into operation to supply steam to the same steam range by suitable cross-connecting steam stop valves (figure 10.9). In port, the exhaust gas boiler is