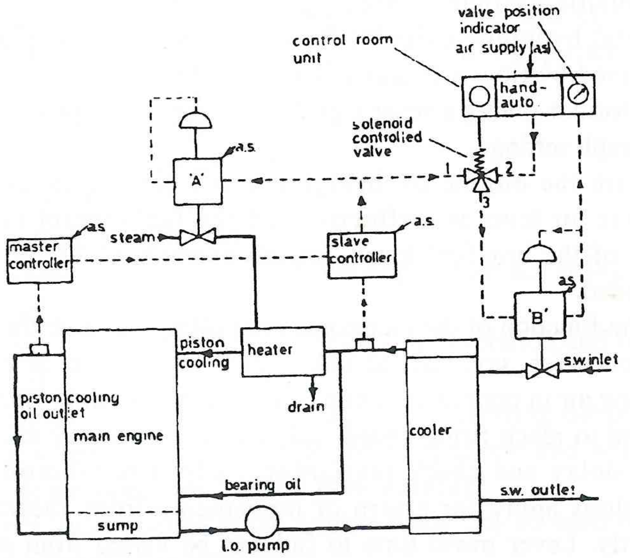
FIG 114
COOLING AND LUB OIL CONTROL



lub. 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 sea water flow.

2. Assuming the sea water 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 lub.oil temperature at inlet to the piston 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 lub. 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 to 2·0 bar.
Valve positioner "A" works on the range 1·2 to 1·4 bar.
Valve positioner "B" works on the range 1·4 to 2·0 bar.
Hence the range is split in the ratio 1·3.

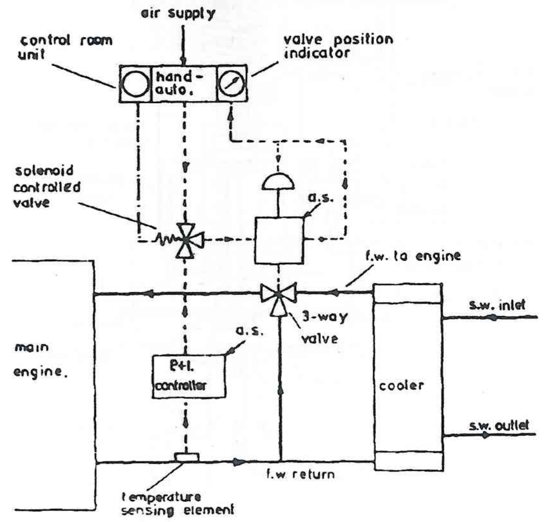
Since the piston cooling oil outlet temperature could be offset from the desired value by upwards of 8°C or more, the master controller must give proportional and reset action. In order to limit the variety of spares that must be carried the slave controller would be identical to the master controller.

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 sea water flow to the cooler. Position of the sea water 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 sea water through the cooler and operate a three way valve (2 inlets, 1 outlet) in the engine fresh water, or cooling circuit that by-passes the cooler.

FIG 115
JACKET (OR PISTON) TEMPERATURE CONTROL

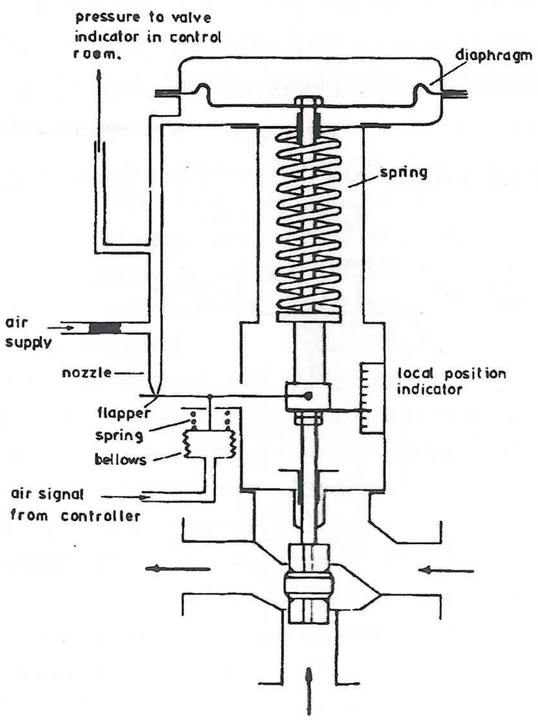


Valve selection for such duties is most important. Maximum pressure and temperature, maximum and minimum flow rate, valve and line pressure drops, etc., must be carefully assessed so that valve selection gives the best results. With correct analysis of the plant parameters and careful valve selection, 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 inlet and one outlet of the type shown diagrammatically in Fig. 116. 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

FIG 116
3-WAY VALVE AND POSITIONER



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.

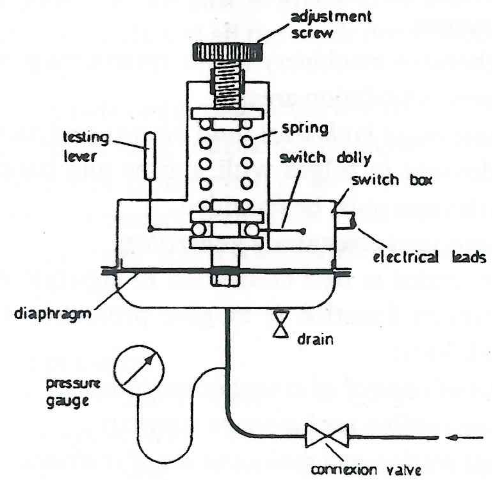
Indication of valve position is given locally and remote, in the latter case by feeding back the diaphragm loading pressure to an indicator possibly situated in the control room. the valve positioner gives accurate positioning of the valve and provides the necessary muscle to operate the valve against the various forces.

Pressure Alarm

The alarm diagrammatically shown in Fig. 117. 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.

FIG 117
PRESSURE ALARM



Unattended Machinery Spaces

These are designated u.m.s. in regulations and in the case of the diesel engines they are gradually increasing in number. A controllable pitch propeller driven by geared unidirectional medium or high speed diesels is a relatively uncomplicated system that lends itself to direct control from the bridge. However, irrespective of the type of installation, certain essential requirements for u.m.s. particularly unmanned engine rooms at night, must be fulfilled. They could be summarised as follows:

1. *Bridge control of propulsion machinery*

The bridge watchkeeper must be able to take emergency engine control action. Control and instrumentation must be as simple as possible.

2. *Centralised control and instruments are required in machinery space*

Engineers may be called to the machinery space in 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 for control and 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*

Such a generator is best 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.*

10. *Regular testing and maintenance of instrumentation.*

CHAPTER 7

ANCILLARY SUPPLY SYSTEMS

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, etc.

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. When air is compressed its temperature and pressure will increase as its volume is reduced.

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.

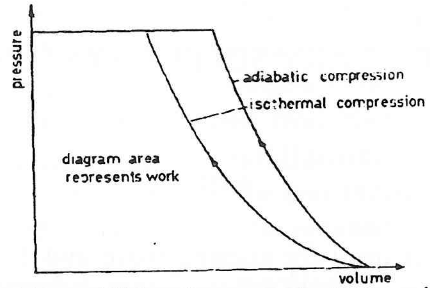
Adiabatic compression of a gas is compression under constant enthalpy conditions, *i.e.* no heat is given to or taken from the gas through the cylinder walls and all the work done in compressing the gas is stored within it.

In Fig 118. the two compression curves show clearly the extra work done by compressing adiabatically, hence it would be more sensible to compress isothermally. In practice this presents a problem – if the 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 under near

FIG 118
A COMPARISON OF COMPRESSION PROCESSES

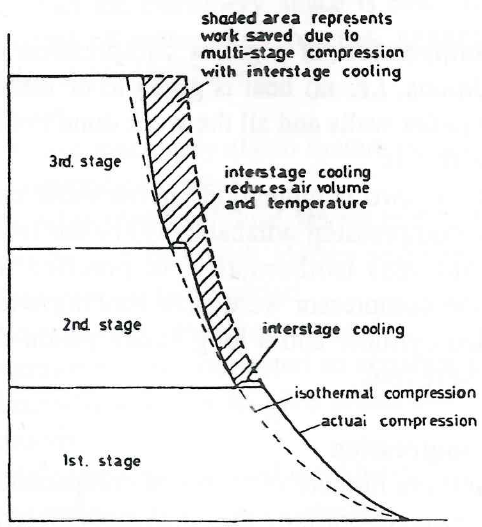


isothermal conditions. This of course is impracticable so two or three stage compression with interstage and cylinder cooling is generally used when relatively high pressures have to be reached.

Fig. 119. shows clearly 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 so.

To prevent 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.

FIG 119
3 STAGE COMPRESSION



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.

Volumetric efficiency 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. Fig. 120. 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, *i.e.* $P_1/P_2 = P_2/P_3$ hence $P_2 = \sqrt{P_1 P_3}$. (Fig. 121.). 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:

FIG 120
EFFECTS OF INCREASING CLEARANCE VOLUME

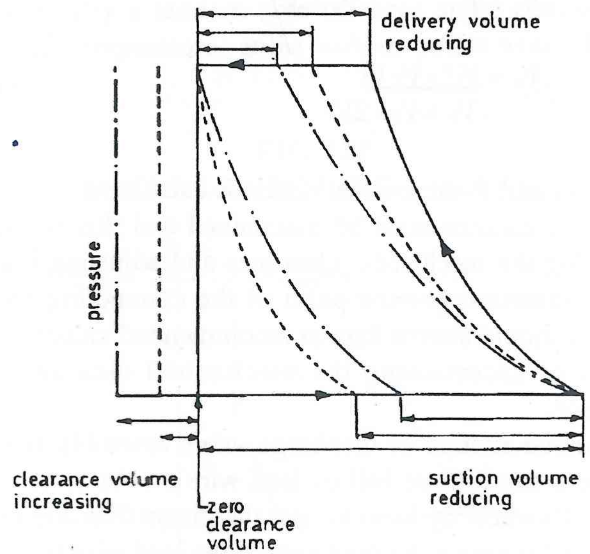
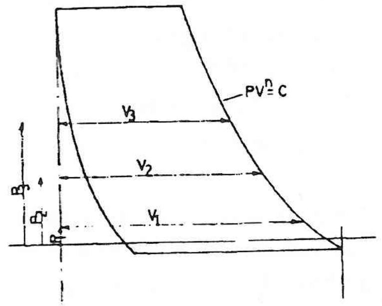


FIG 121
CALCULATING CLEARANCE VOLUME



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

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

now $\quad \dfrac{P_1}{P_2} = \dfrac{P_2}{P_3}$

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

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

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

Since V_1, V_2 and V_3 are known V_c can be calculated.

Correct clearance must be maintained and this is usually done by checking the mechanical clearance and adjusting it as required by using inserts under the palm of the connecting rod. Bearing clearances should also be kept at recommended values.

Methods of ascertaining the mechanical clearance in an air compressor:

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 top dead centre. Remove and measure the thickness of the lead wire ball.

2. Put crank on top dead centre, 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, *i.e.* raise it until it touches the cylinder cover. Take another gauge reading, the difference between the two readings gives the mechanical clearance.

In practice the effective volume drawn in per stroke is further reduced since the pressure in the cylinder 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. Fig. 122. shows this effect on the actual indicator card and also the excess pressure above the mean required upon delivery, to overcome delivery valve inertia and spring force.

Air compressors are either reciprocating or rotary types, the former are most commonly used at sea for the production of air for purposes outlined at the beginning of this chapter. The latter types are used to produce large volumes of air at relatively low pressure and are used at sea as integral parts of main engines for scavenging and for boiler forced draught.

Reciprocating air compressors at sea are generally two or three stage types with inter-stage cooling. Fig. 123. shows diagrammatically a tandem type of three stage compressor, the pressures and temperatures at the various points would be roughly as follows:

FIG 122
ACTUAL PRESSURE-VOLUME

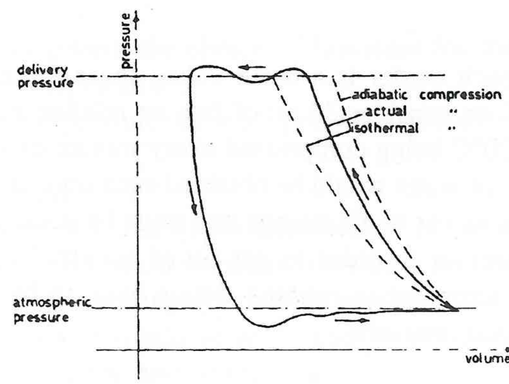
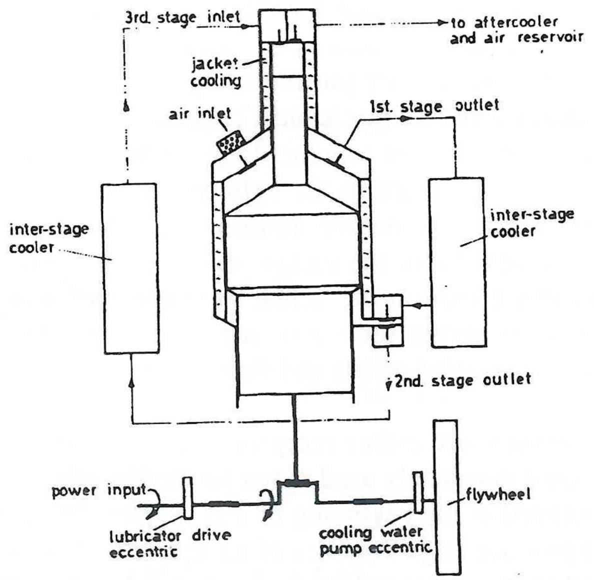


FIG 123
3 STAGE AIR COMPRESSOR



	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
Third stage	40 bar	70°C	25°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 1/2 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 compressed air carrying tube bursting, the sudden rise in pressure of the surrounding water will not fracture the cooler casing. In the event of a failure of a bursting disc a thicker one must not be used as a replacement.

Lubrication

Certain factors govern the choice of lubricant for the cylinders of an air compressor, these are:

Operating temperature, cylinder pressures and air condition.

Operating Temperature

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.

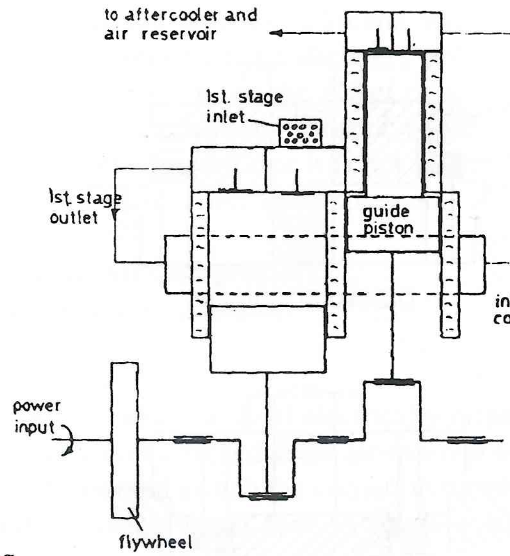
Two Stage Air Compressor
Most modern diesel engines use starting air at a pressure of about 26 bar and to achieve this a two stage type of compressor would be adequate. These compressors are generally of the reciprocating type, with various possible arrangements of the cylinders, or they could be a combination of a rotary first stage followed by a reciprocating high pressure stage. This latter arrangement leads to a compact, high delivery rate compressor.

Fig. 124. shows a typical two-stage reciprocating type of air compressor, the pressures and temperatures at the various points would be approximately as follows:

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

Compressor Valves
Simple suction and discharge valves are shown in Fig. 125. These would be suitable diagrams for reproduction in an examination. Modern valves are somewhat more streamlined and lighter in order to reduce friction losses and valve inertia. Materials used in the construction are generally:

FIG 124
2 STAGE AIR COMPRESSOR



Valve Seat
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. (N.B. all hardened steel would be tempered).

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

Effects of Leaking Valves

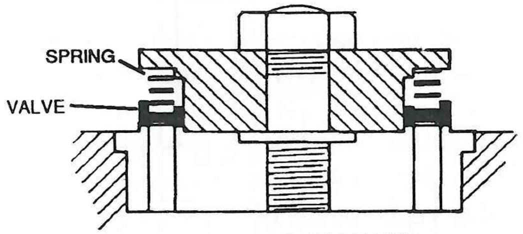
1. *First stage Suction*

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.

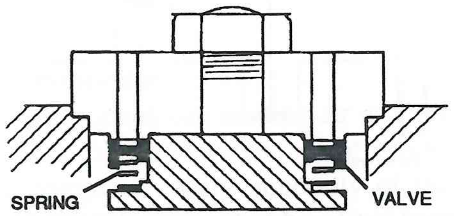
2. *First Stage Delivery*

With high pressure air leaking back into the cylinder less air can be drawn in, this means reduced delivery and increased discharge temperature.

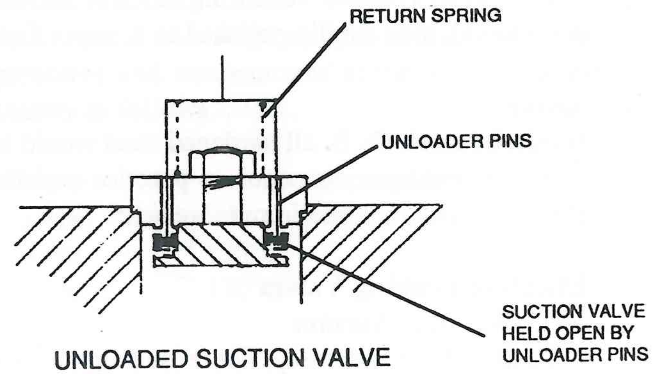
FIG 125
COMPRESSOR VALVES



DISCHARGE VALVE



SUCTION VALVE



UNLOADED SUCTION VALVE

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. Fig. 126. 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.

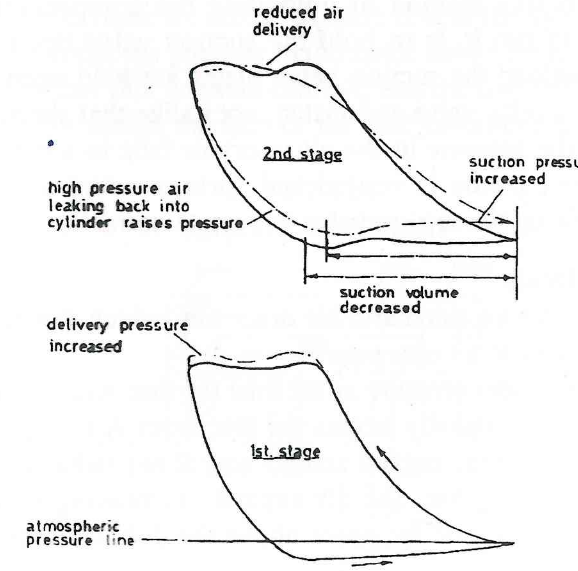
Regulation of Air Compressors

Various methods are available:

Start Stop Control

This is only suitable for small electrically driven types of unit. A pressure transducer attached to the air receiver set for desired max-min pressures would switch the current to the electric motor either on or off. Drainage would have to be automatic and air receiver

FIG 126
EFFECT OF LEAKING 2ND STAGE DELIVERY VALVES



relatively large compared to the compressor unit requirements so that the number of starts per unit time is not too great. It must be remembered that the starting current for an electric motor is about double the normal running current.

Constant Running Control

This method of control is the one most often used. 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 no air is delivered and practically no work is done in the compressor cylinders.

The methods used for compressor unloading vary, but that most commonly used is on 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 suction no air will be delivered. In either case virtually no work would be done in the compressor cylinder or cylinders and this would provide an economy compared to discharging high pressure air to the atmosphere through a relief valve.

Fig. 127. 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.

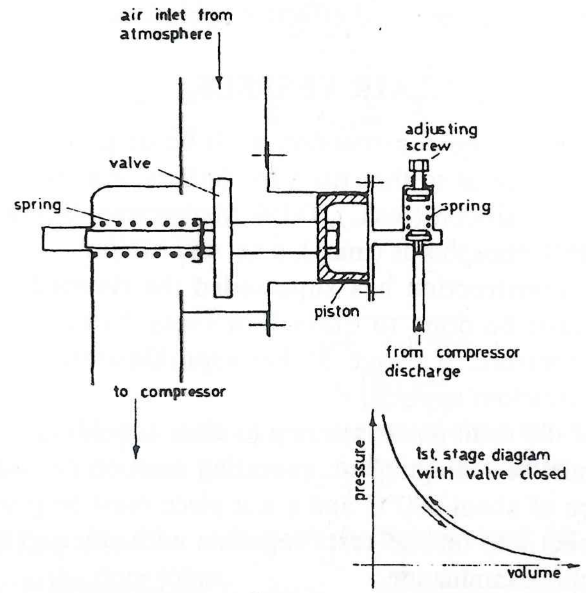
An alternative method of unloading the compressor, while continuing to run it, is to hold the suction valve open. During periods of unload the suction valve plates are held open by pins operated by a relay valve and piston, not unlike that shown in Fig. 127. When the pressure in the air reservoir falls to a preset level, the unloading piston is vented and springs withdraw the pins holding allowing the suction valve to operate normally. Fig. 125.

Automatic Drain

Fig. 128. 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

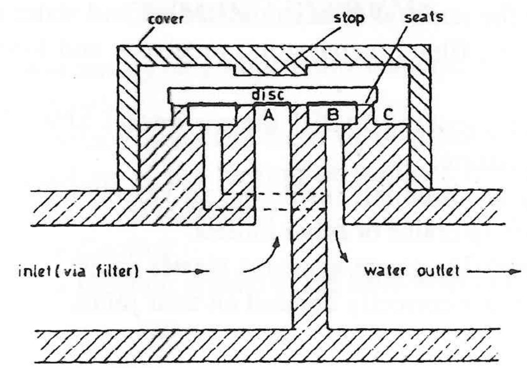
FIG 127
COMPRESSOR UNLOADING VALVE



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 re-open, a small groove is cut across the face of the disc communicating B and C

FIG 128
AIR DRAIN TRAP



through which the air slowly leaks to outlet.

Obviously this gives an operational frequency to the opening and closing of the disc which is a function of various factors, *e.g.* size of groove, disc thickness, volume of space above the disc, etc. It is therefore essential that the correct trap be fitted to the drainage system to ensure efficient and effective operation.

AIR VESSELS

Material used in the construction must be of good quality low carbon steel similar to that used for boilers, *e.g.* 0·2% Carbon (max.), 0·35% silicon (max.), 0·1% 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.

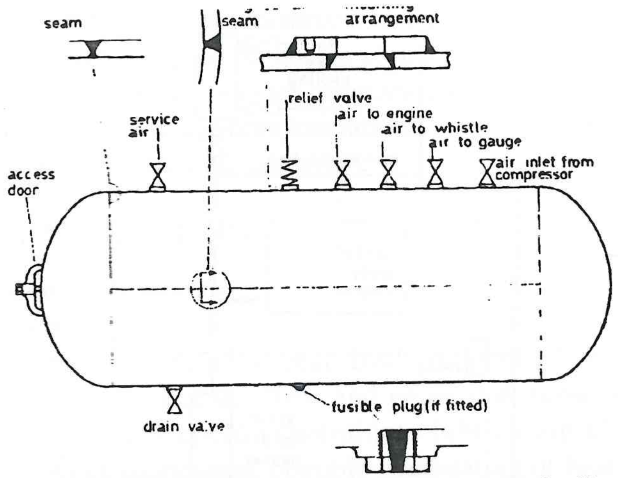
Mountings generally provided are shown in Fig. 129. If it is possible for the receiver to be isolated from the safety valve then it must have a fusible plug fitted, melting point approximately 150°C, and if carbon dioxide is used for fire fighting it is recommended that the discharge from the fusible 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 pipe-lines.

Before commencing to fill the air vessel after overhaul or examination, ensure:

1. Nothing has been left inside the air vessel, *e.g.* cotton waste that could foul up drains or other outlets.
2. Check pressure gauge against a master gauge.
3. All doors are correctly centred on their joints.

FIG 129
AIR RESERVOIR



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.

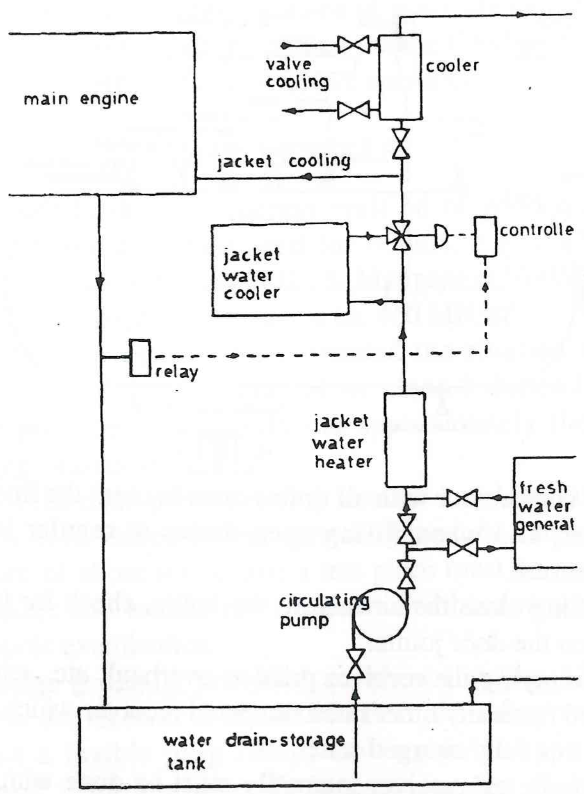
1. *Cylinder Cooling*

Or jacket cooling: normally fresh or distilled water. This may incorporate cooling of the turbine or turbines in a turbocharged engine and exhaust valve cooling.

2. *Fuel Valve Cooling*

This would be a separate system using fresh water or a fine mineral oil.

FIG 130
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 sea water.

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 Fig. 131., the cooling flow is split into a primary circuit, bypassing the liner, for cylinder head cooling. In the secondary circuit uncooled water from engine outlet is directed to cool the liner. To avoid vapour formation as a result of maintaining higher cooling temperatures the system is pressurised to 4 to 6 bar.

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

1. *Fresh Water*

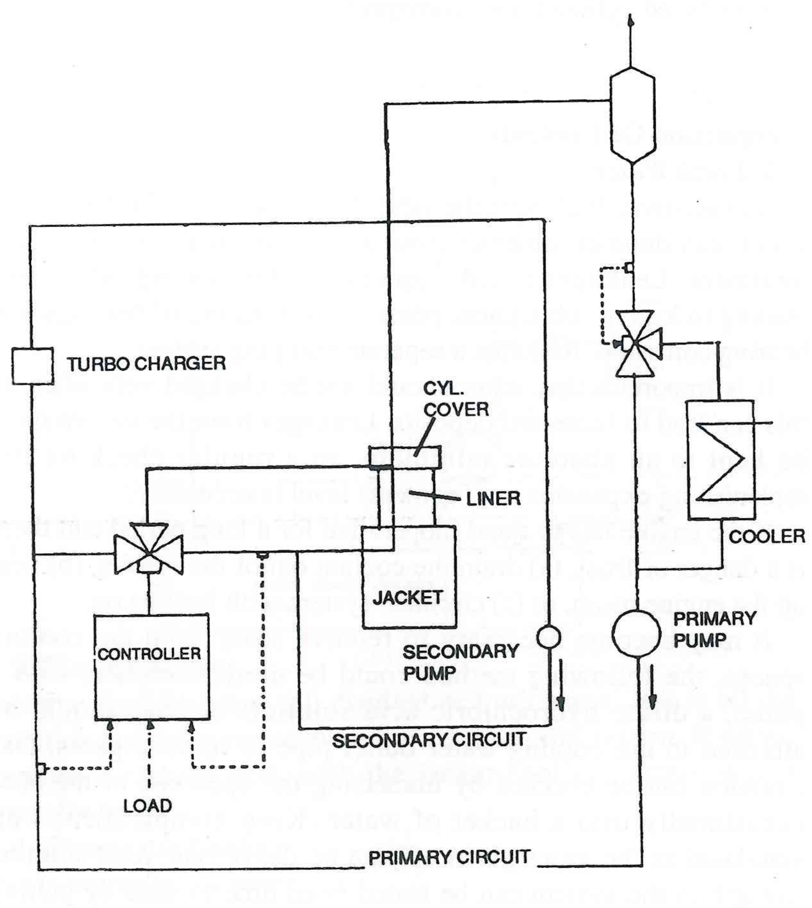
Inexpensive, high specific heat, low viscosity. Contains salts which can deposit, obstruct flow and cause corrosion. Requires treatment. Leakages could contaminate lubricating oil system leading to loss of lubrication, possible overheating of bearings and bearing corrosion. Requires a separate pumping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

It may become necessary to remove scale from the cooling spaces, 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 on to 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.

FIG 131
LOAD CONTROLLED CYLINDER COOLING



2. *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 is a chemical that must be handled with care.

3. *Lubricating Oil*

Expensive. Generally no separate pumping system required since the same oil is normally used for lubrication and cooling. Leakages from cooling system to lubrication system are relatively unimportant providing 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 in 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 lubricating oil encounters high temperature it can burn leaving as it does so carbon deposit. 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:

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, *e.g.* oiliness agents, e.p. agents and V.I. improvers, etc.

3. Improved control over piston temperatures.

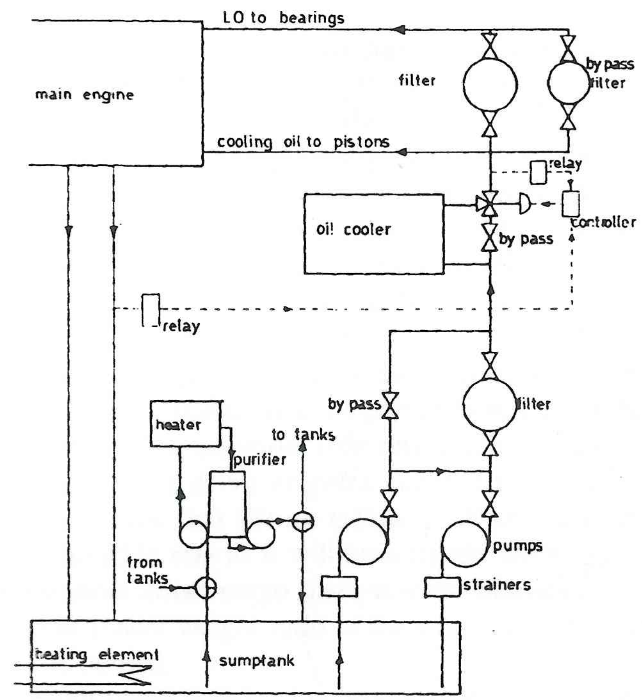
4. If oil loss occurs, then with separate systems the problem of detection is simplified and in the case of total oil loss in either system, the quality to be replaced would not be as great as for a common system.

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.

FIG 132
LUBRICATING AND COOLING OIL SYSTEM



CHAPTER 8

MEDIUM SPEED DIESELS

The term medium speed refers to diesels that operate within the approximate speed range 300 to 800 revolutions per minute. High speed is usually 1000 rev/min and above.

The development in the medium speed engine has been such that it is now a serious competitor for applications which were once only the domain of slow-speed 2-stroke engines. If the advantages and salient features of the medium speed diesel are examined the reader will appreciate why this swing, for certain vessels tonnages, is taking place. They are as follows:

1. Compact and space saving. 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 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 to 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 and concentrate solely upon producing an engine that will give the best possible power weight ratio.

3. Modern tendency is to utilise uni-directional medium speed geared diesels coupled to either a reverse reduction gear, controllable pitch propeller or electric generator. The first two of these methods are the ones primarily used and the advantages to be

gained are considerable, they are:

(a) Less starting torque required, clutch disengaged or controllable pitch propeller 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 occurs upon 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 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 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 c.p. propeller.

(f) Staff load during 'stand-by' periods is reduced, system lends itself ideally to simple bridge control.

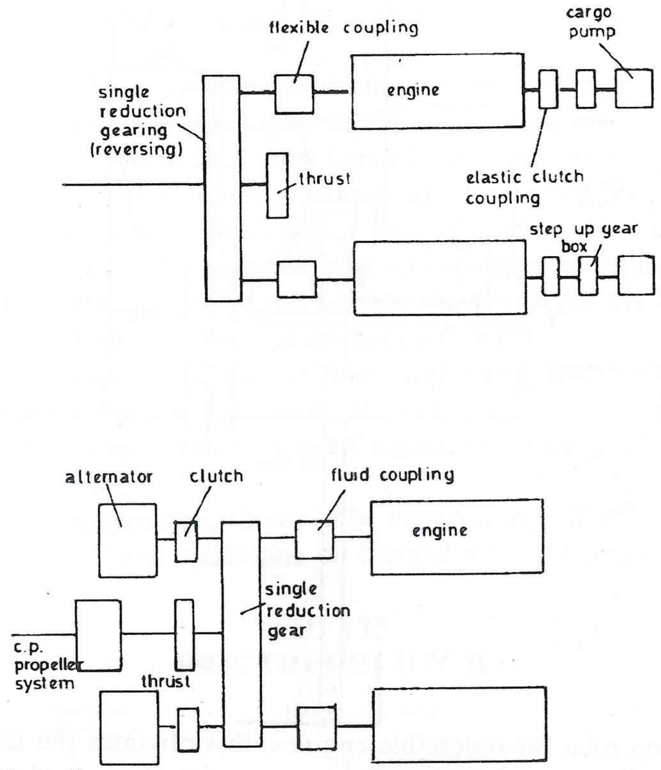
4 With two engines coupled via gearing one may be disengaged, whilst the other supplies the motive power, and overhauled. This reduces off hire time and voyage is continued at slightly reduced speed with a fuel saving.

5. Spare parts are easier to store and manhandle, unit overhaul time will be greatly reduced.

ENGINE COUPLINGS, CLUTCHES AND GEARING

Various arrangements of geared coupled engines are possible, the basic arrangement depends upon the services the engine has to supply, *e.g.* 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 with some form of clutch. Large capacity cargo pumps operating at high speed would require a similar arrangement. Fig. 133. shows different types of arrangements with different types of clutches or couplings being used.

FIG 133
ENGINE ARRANGEMENTS

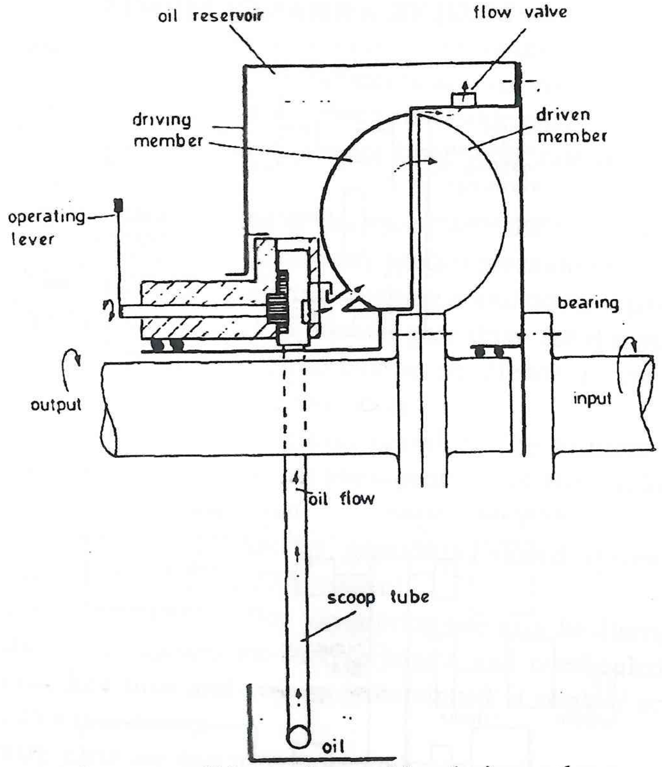


Fluid Couplings

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

FIG 134
FLUID COUPLING (VULCAN)



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 4-stroke engines are used camshafts have to be moved, etc.

Reverse Reduction Gear

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

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 large powers the friction clutches used in the system can become excessively large, great heat generation during engagement may require a cooling system, the whole becomes more expensive and it may be cheaper to use direct reversing engines – however it would for reasons previously outlined be prudent to use a c.p. propeller.

Two systems of reverse reduction gear are shown in Figs. 135. and 136. In Fig. 135. 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

FIG 135
FRICTION CLUTCH

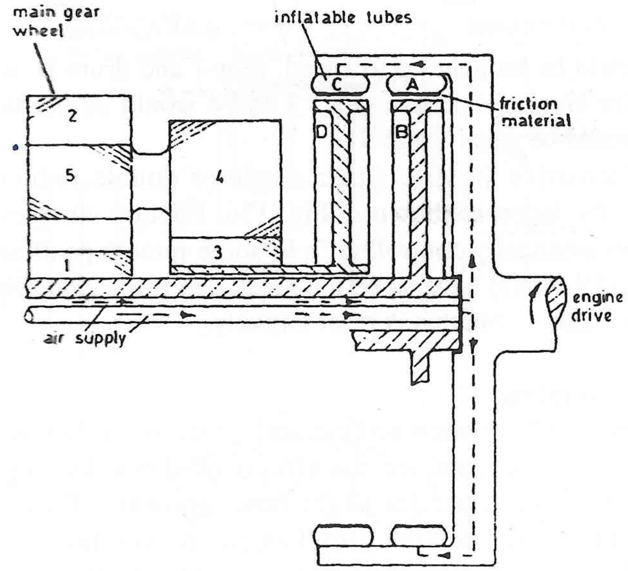
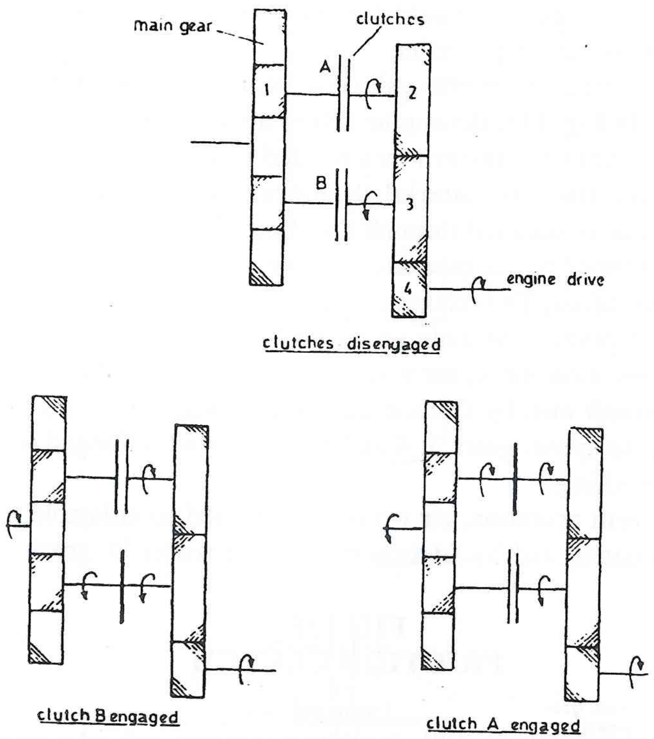


FIG 136
REVERSIBLE REDUCTION GEAR



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 Fig. 136. 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.

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

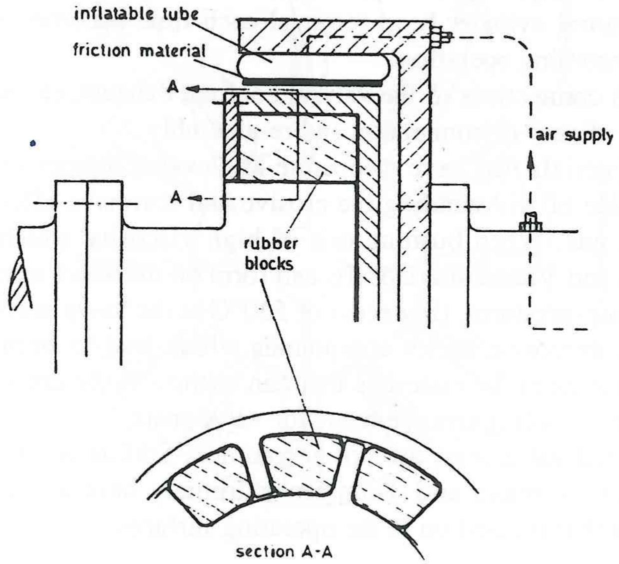
multi-tooth type to be found in turbine installations or employ diaphragms or rubber blocks. Those types that use rubber or synthetic rubber, such as Nitrile, give electrical insulation between driving and driven members, but all types will minimise vibration and reduce noise level.

Fig. 137. shows a combination of flexible couplings and pneumatically operated friction clutch, the arrangement 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.

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

FIG 137
FLEXIBLE CLUTCH COUPLING



components of the coupling. The makers claim that this effective damping is achieved without problems of wear because of the absence of friction. Fig. 138.

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.

EXHAUST VALVES

Most 4-stroke medium speed diesels incorporate two exhaust valves per cylinder and if we consider a moderate sized installation consisting of two 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:

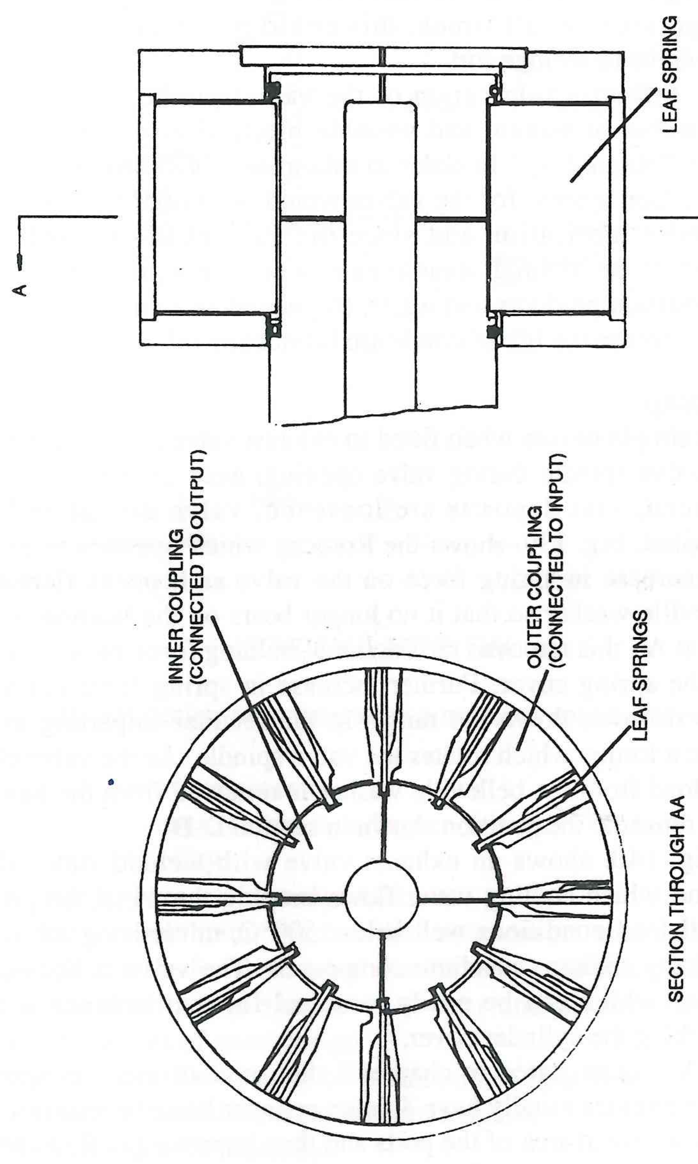
1. Separately caged exhaust valves are preferred even though they increase first cost. If they are made integral with the cylinder head and a fuel of poorer quality than normal has to be burnt, increased frequency of replacement and overhaul of the valves necessitating cylinder head removal each time becomes a tedious time consuming operation.

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.

Stellited valve seats are not uncommon, Stellite is a mixture of Cobalt, Chromium and Tungsten extremely hard and corrosion resistant that is fused on to the operating surfaces.

FIG 138
GEISLINGER TORSIONAL VIBRATION DAMPING COUPLING



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, this could prove complicated and expensive to arrange for.

4. 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, this would also increase the life of crankcase lubricating oil.

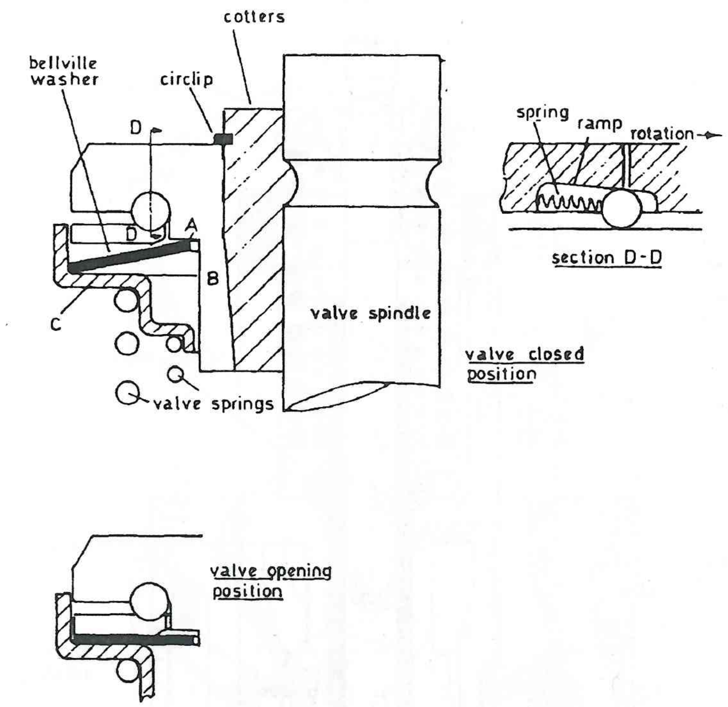
Rotocap

This simple device when fitted to exhaust valves causes rotation of the valve spindle during valve opening, wear of the valve seat is reduced, seat deposits are loosened, valve operation life is extended. Fig. 139. shows the Rotocap which operates as 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.

Fig. 140. 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 4-stroke engines usually have 4 valve cylinder heads to maximise the cross-sectional area of the ports and thus improve gas flow through the engine. The gas flow of a typical 4 valve cylinder head is shown in Fig. 141.

FIG 139
ROTOCAP

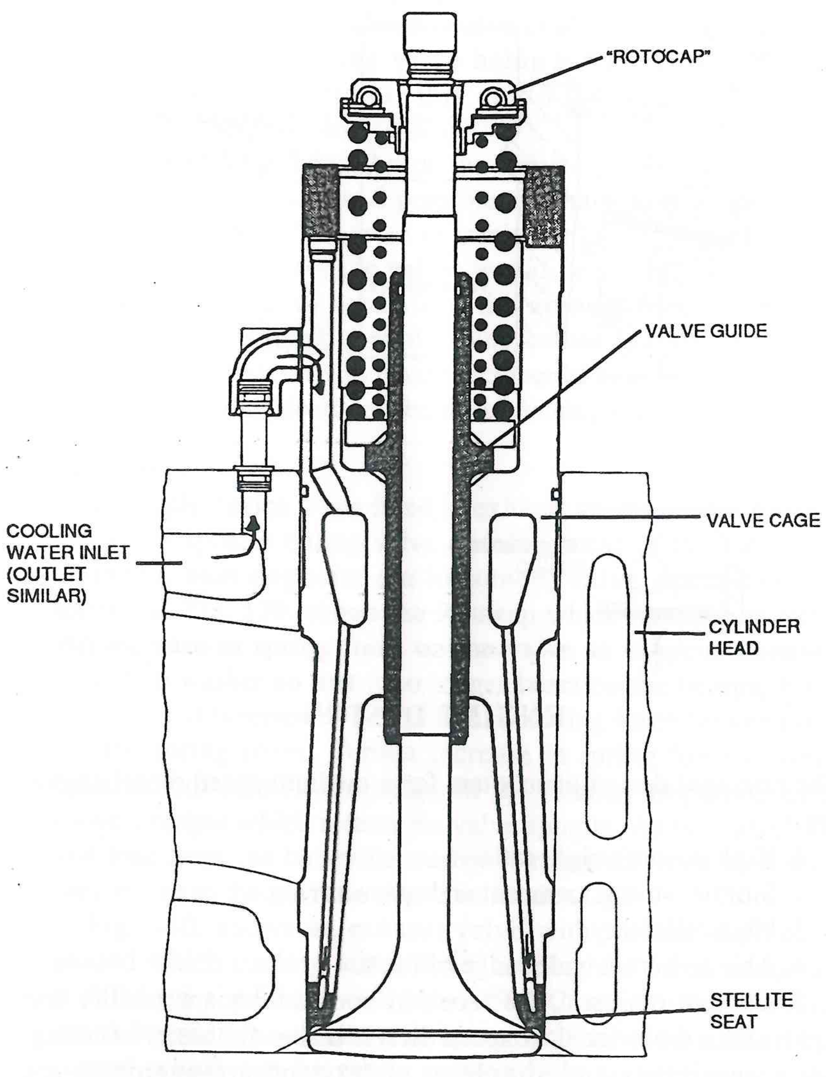


ENGINE DESIGN

The principal design parameters for a medium speed diesel engine are:

1. High power/weight ratio.
2. Simple, strong, compact and space saving.
3. High reliability.
4. Able to burn a wide range of fuels.
5. Easy to maintain, the fact that components are smaller and lighter than those for slow speed diesels makes for easier handling, but accessibility and simple to understand arrangements are inherent features of good design.
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.

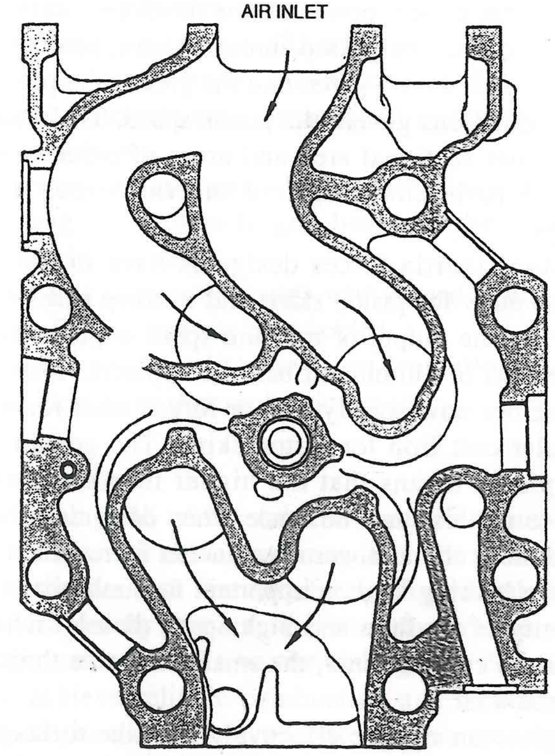
FIG 140
EXHAUST VALVE



Types of Engine

Either 2- or 4-stroke cycle single acting turbocharged with 'in line' or 'V' cylinder configuration. The main choice is, certainly at present, for the 4-stroke engine and there are various reasons for this.

FIG 141
GAS FLOW OF TYPICAL 4 VALVE CYLINDER HEAD



1. They are capable of operating satisfactorily on the same heavy oils as slow speed 2-stroke engines.

2. Effective scavenging is relatively easy to achieve in slow speed 2-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 4-stroke configuration. With large inlet and exhaust valve overlap effective scavenging can be accomplished. Scavenging is further improved by utilising high turbocharger pressure ratios. Pressure ratios of 3·5 to 4·0 are not now uncommon with 4·5 to 5·0 being developed. Good scavenging and high turbocharger pressure ratio results in engines producing high turbocharger pressure ratio results in engines producing high BMEP figures.

3. Higher mean piston speed. Mean piston speed is simply twice the stroke times the revs/second. For medium speed diesels it would be approximately 9 to 10 m/s and for slow running diesels 7 to 9 m/s would be about average.

In order that greater power can be developed in the cylinder the working fluid must be passed through faster, hence the higher the mean piston speed for a given unit the greater the power. However, practical limitations govern the piston speed. The relation between cylinder cross sectional area and areas of exhaust and air inlet, method of turbocharging 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, the smaller the side thrust the less the friction and wear.

4. Engine can operate effectively with the turbocharger out of commission, this would present a considerable problem with some 2-stroke engines of the medium speed type.

5. Turbocharger size and power can be reduced.

6. It is also claimed that the fuel consumption would be reduced.

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
- 4-stroke turbocharged 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/kW hr

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

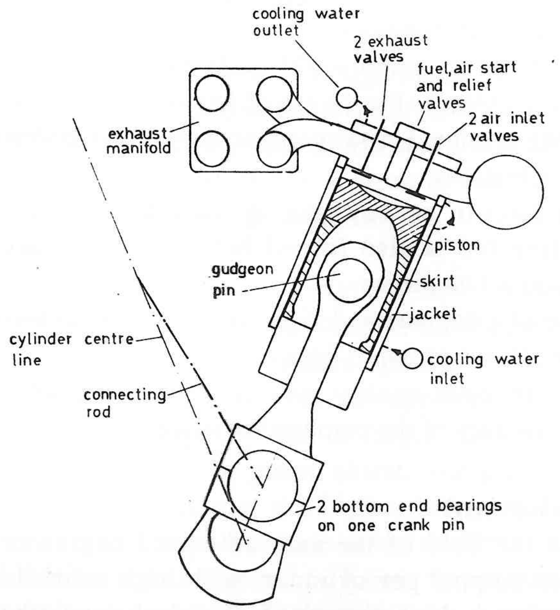
The trend in the field of the medium speed engine is towards higher power outputs per cylinder, with high reliability, when operating on cheaper high viscosity fuels. Much development work is being carried out by manufacturers to improve the combustion process. This work focuses on the timing and duration 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 1250 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 crank pin 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

FIG 142
V-TYPE ENGINE

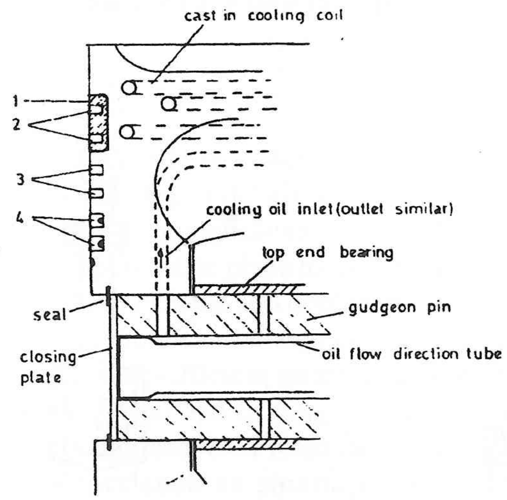


engine sump. Oil would also be taken from the main feeder to lubricate camshaft gear drive, camshaft bearings, pump bearings, etc.

Fig. 143. 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.

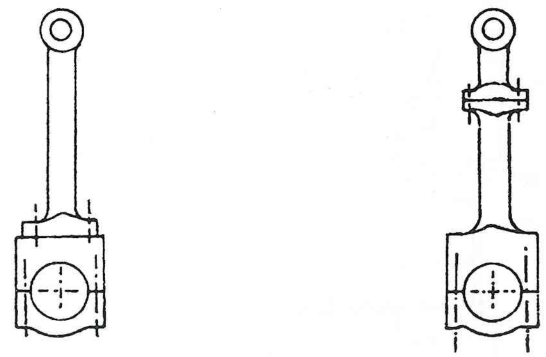
Due to complex vibration problems that can arise in medium speed engines of the 'V' type it would appear important to have a very strong and compact arrangement of bedplate etc. Excessive vibration of the structure can lead to increased cylinder liner wear and considerable amounts of lubricating oil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

FIG 143
PISTON COOLING

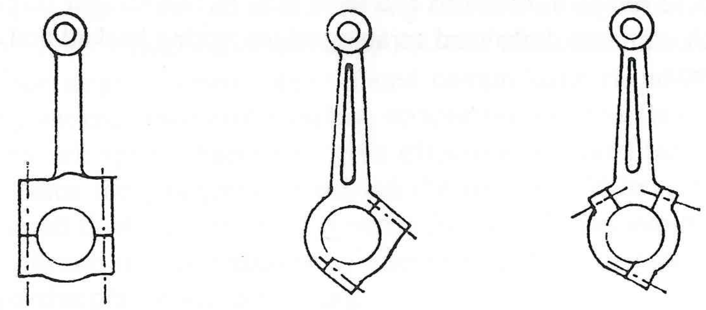


Alkaline lubricating oil of the type used in these engines is expensive and because the engines are mainly trunk type consumption rates can be high. Positioning, and type, of oil scraper ring is important. With some engines they have been moved from a position below the gudgeon pin to above since considerable 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.

FIG 144
VARIATIONS OF CONNECTING ROD DESIGN



"MARINE TYPE" CONNECTING ROD



CONNECTING RODS MAY BE ROUND OR H SECTION

CHAPTER 9

WASTE HEAT PLANT

General Details

Reference should be made to chapter 1 for general comments relating to heat balance. Fig. 3 details an approximate heat balance for an IC engine showing significant losses to the exhaust and cooling. Every attempt must be made to utilise energy in waste heat and recovery 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

Obviously most boilers and heaters have arrangements for burning oil fuel during low engine power conditions. It is therefore appropriate to repeat some very general remarks on combustion with details of typical equipment in use.

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 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

If these 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 affect combustion, it could lead to the burners being extinguished and a dangerous situation arising. It could also produce panting 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 it should be noted that clear uptake gases can be achieved while supplying excess 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.

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 Fig. 145. 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. Fig. 146a. The water wall tubes 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 Fig. 146b. In some Sunrod designs the firetube is also water-cooled. This design is manufactured in sizes ranging from 700 kg/hr to 35000 kg/hr with pressures up to 18 bar. The boiler is usually fitted with automatic start up/shut down and combustion control.

Due to the absence of furnace refractory lining this type of boiler is extremely robust and easy to operate. Cleaning the boiler

FIG 145
SUNROD MARINE BOILER

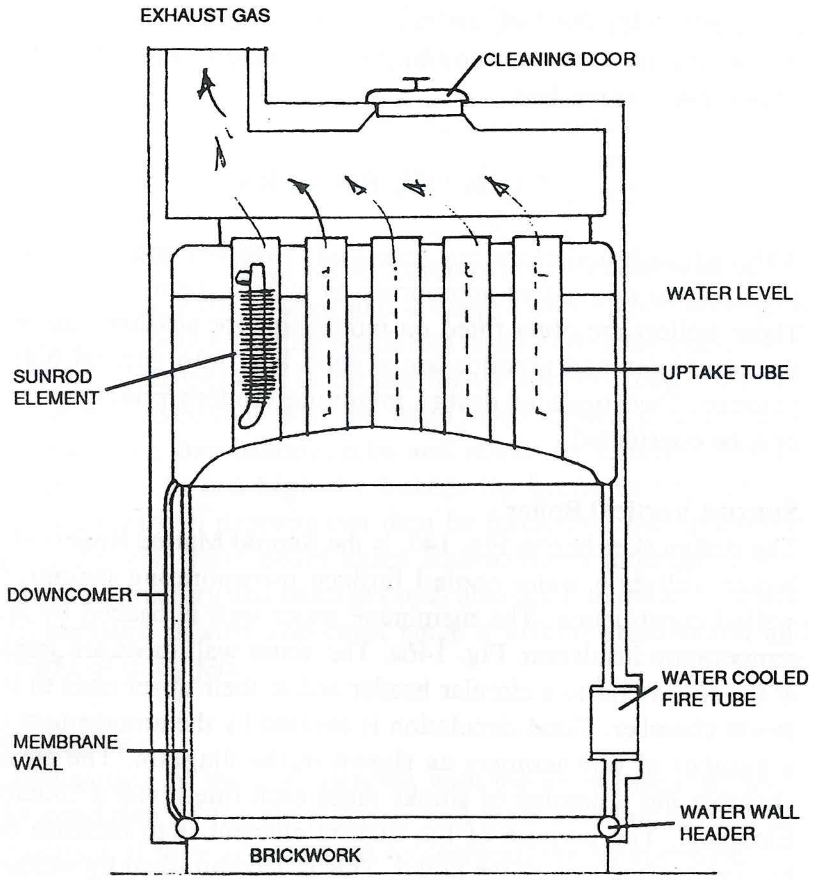
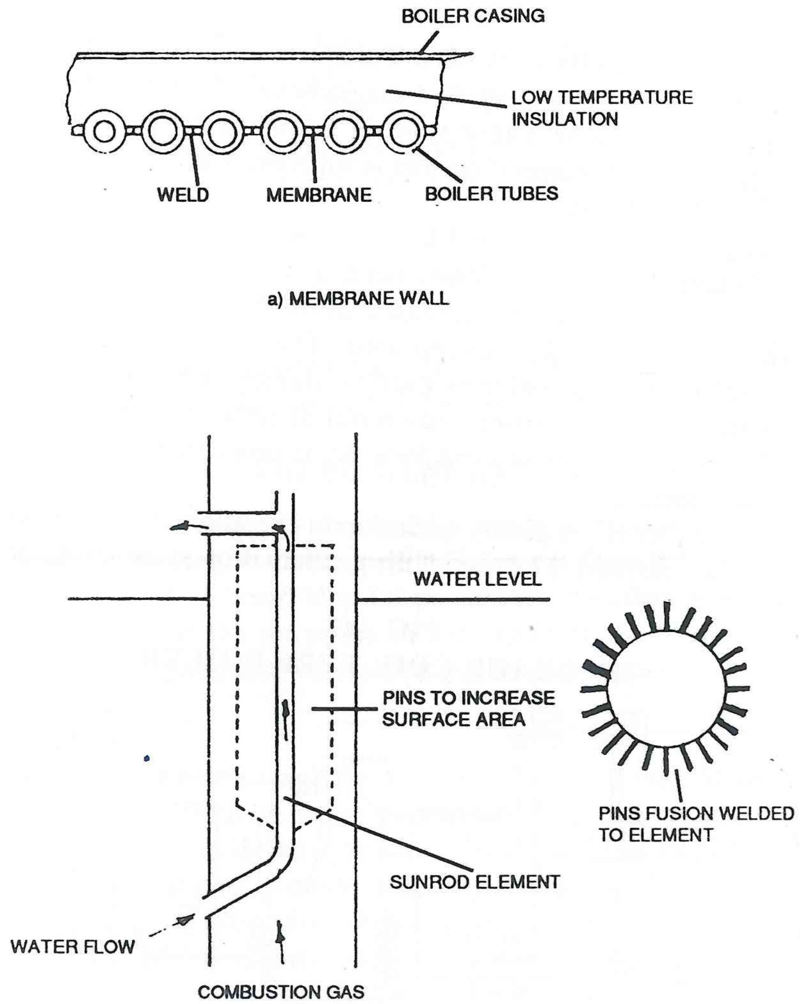


FIG 146
SUNROD BOILER DETAIL



a) MEMBRANE WALL

b) SUNROD ELEMENT

is also relatively easy and is accomplished, 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 pre-set 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)

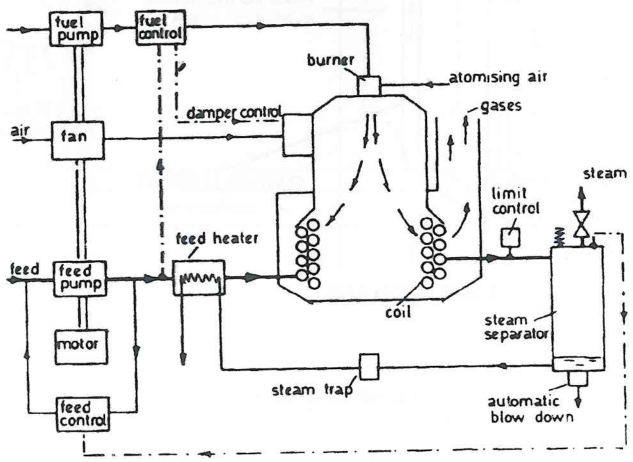
Fig. 147. shows in a simplified diagrammatic form a coiled-tube boiler of the Stone-Vapor type. It is compact, space saving, designed for u.m.s. operation, and is supplied ready for connecting to the ships services.

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

FIG 147
PACKAGE COIL TYPE BOILER



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 3000 kg/h
- Thermal efficiency 80%
- Full steam output in about 3 to 4 mins.

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.)

TURBO GENERATORS

Such turbines are fairly standard l.p. steam practice and reference, where necessary, could be made to Volume 9. 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 standard modern practice.

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 incorporates 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 Fig. 148. 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 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 Fig. 148. and the construction should be clear. Energy pulsations and sound waves are dissipated by repeated throttling and expansion.

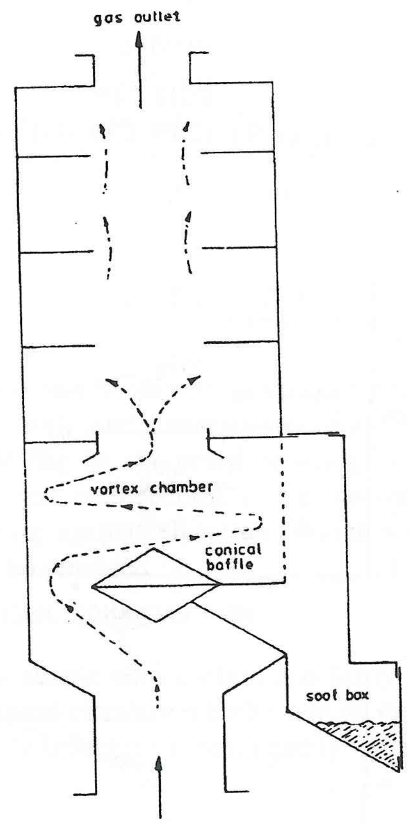
GAS ANALYSIS

A number of factors have been stated which affect the design and operation of the plant and some salient points will now be briefly considered.

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

FIG 148
SILENCER AND SPARK ARRESTER

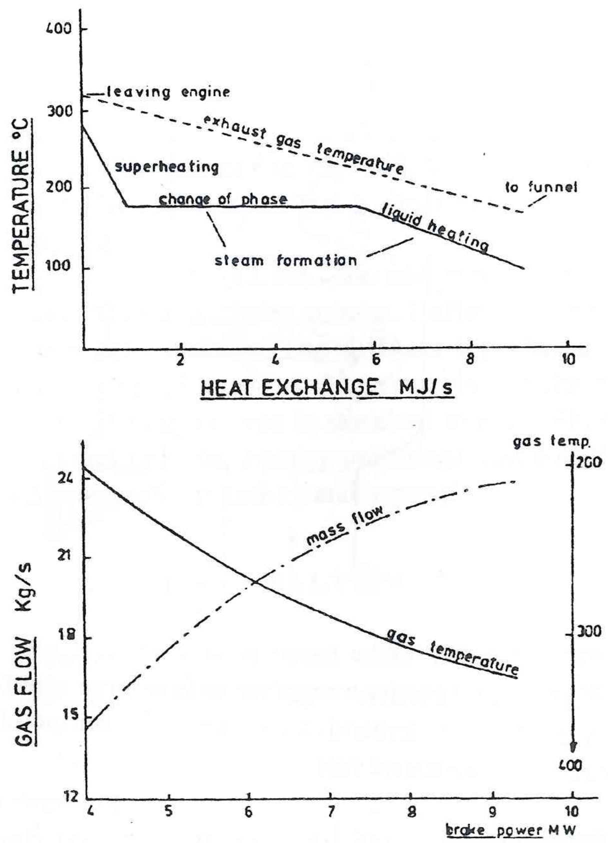


certain auxiliary functions together with reserve steam capacity to meet variations in demand. Low feed inlet temperatures reduce pressure and evaporative rate.

Temperature

A minimum temperature differential obviously applies for heat transfer. Temperature difference, fouling, gas velocity, gas distribution, metal surface resistance, etc., are all important factors. Reduction in service engine revolutions will cause reduced gas mass 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. Fig. 149. illustrates: (a) typical heat transfer diagram, and (b) gas temperature/mass-power curves. A common temperature differential is about 40°C, *i.e.* water inlet 120°C and gas exit 160°C.

FIG 149
EXHAUST GAS CONDITIONS



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 been fitted to auxiliary oil engine exhaust systems, operating mainly as economisers, 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 change-over 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

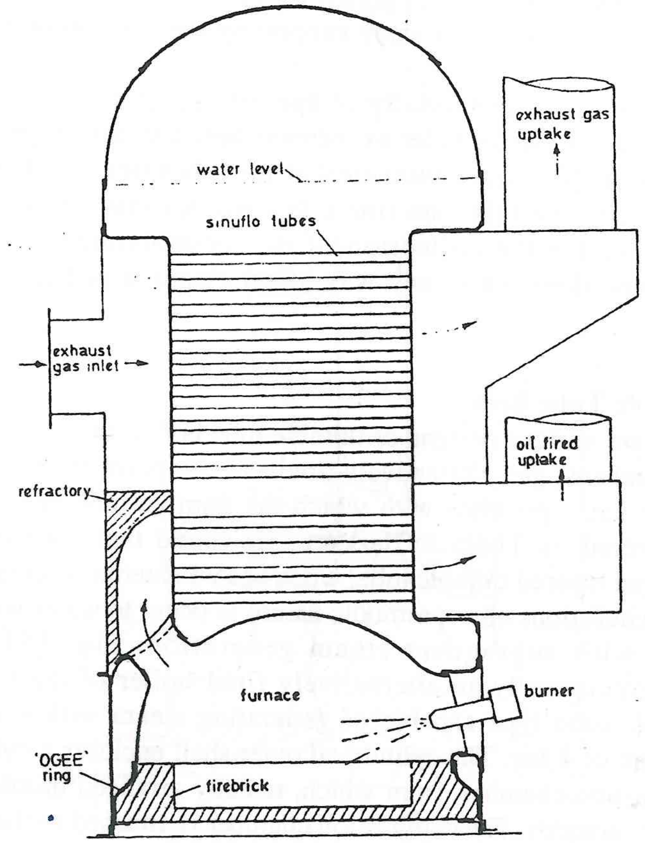
Such boilers are arranged for simultaneous operation on waste heat and oil fuel. The oil fuel section is usually only single pass. Early designs utilised Scotch boilers, with, say, a three furnace boiler, it may mean retaining the centre or the wing furnaces for oil fuel firing. The gas unit would often have a lower tube bank in place of the furnace, with access to the chamber from the boiler back, so 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 boiler 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:

Single pass composite, *i.e.* one pass for the exhaust gases and two uptakes, one for the oil fired system and one for exhaust system. Double pass composite, *i.e.* two passes for the exhaust

FIG 150
DIAGRAMMATIC ARRANGEMENT OF A SINGLE PASS COMPOSITE COCHRAN BOILER



gases and two uptakes, one for the oil fired system and one for the exhaust system. (Double pass exhaust gas, no oil fired furnace and a single uptake, is available as a simple type. Or, double pass alternatively fired, *i.e.* 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, the 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

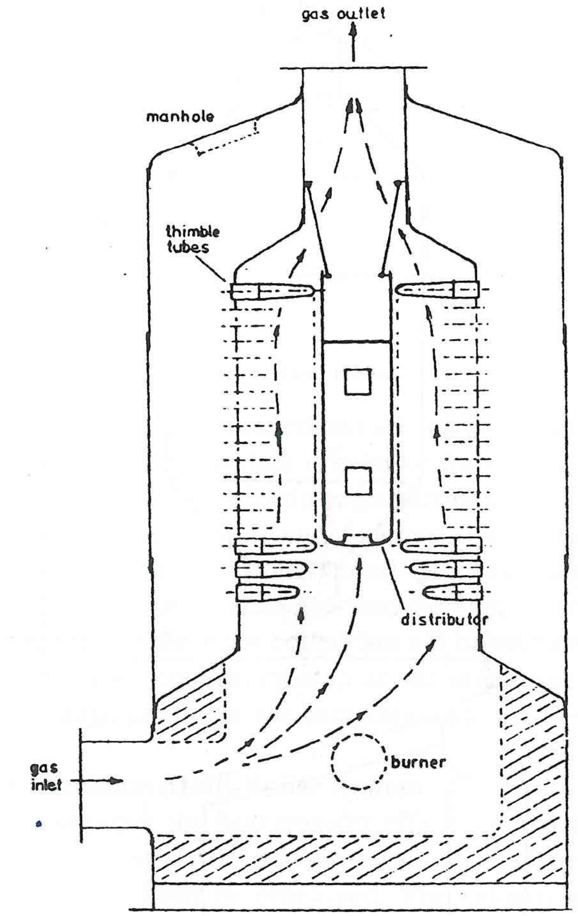
There are various designs of thimble tube boiler, these include: oil fired, exhaust gas, alternatively fired and composite types.

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. Fig. 151 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, whilst cheap, are not flexible and have relatively small steam generating capacity. The systems now considered are based on multi-boiler installations.

FIG 151
ALTERNATIVELY FIRED THIMBLE-TUBE BOILER



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 whilst 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. In port

FIG 152
NATURAL CIRCULATION/WASTE HEAT PLANT AND W.T. BOILER

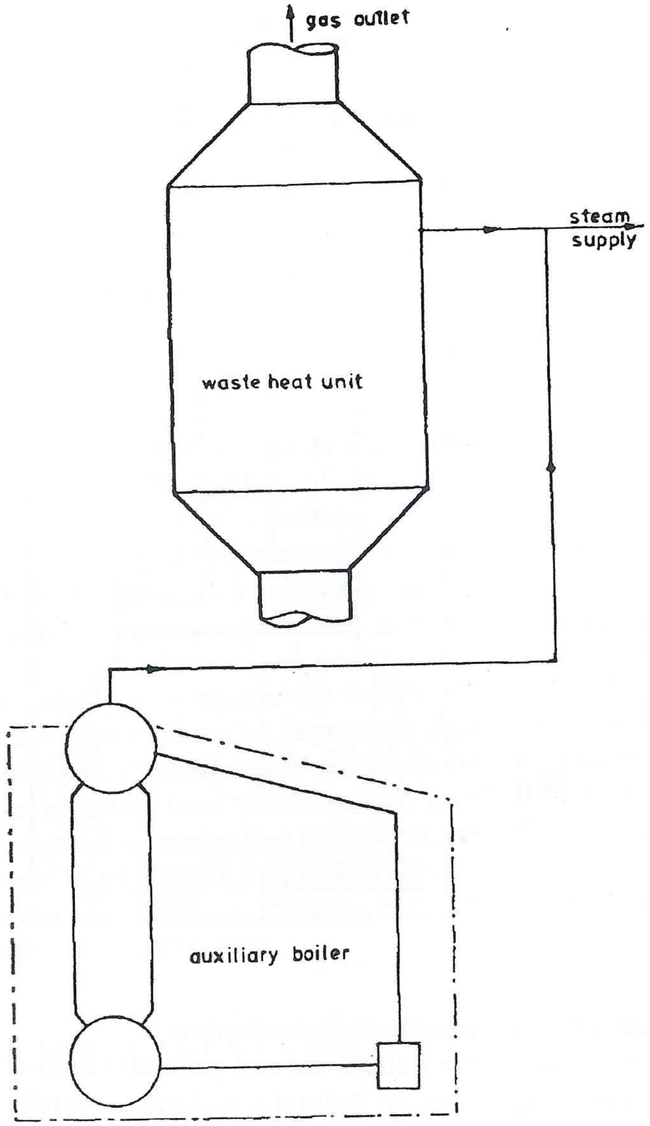
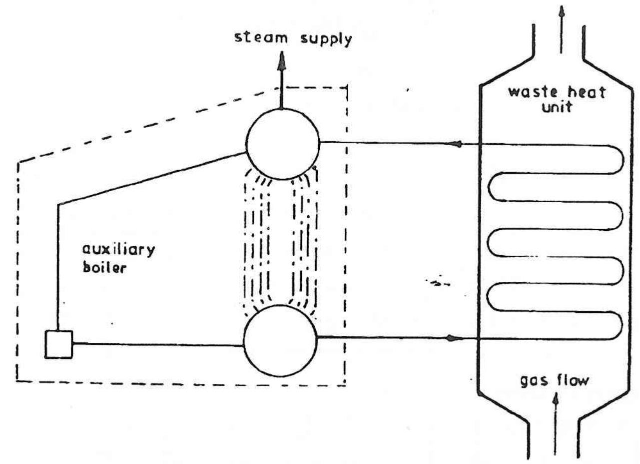


FIG 153
NATURAL CIRCULATION

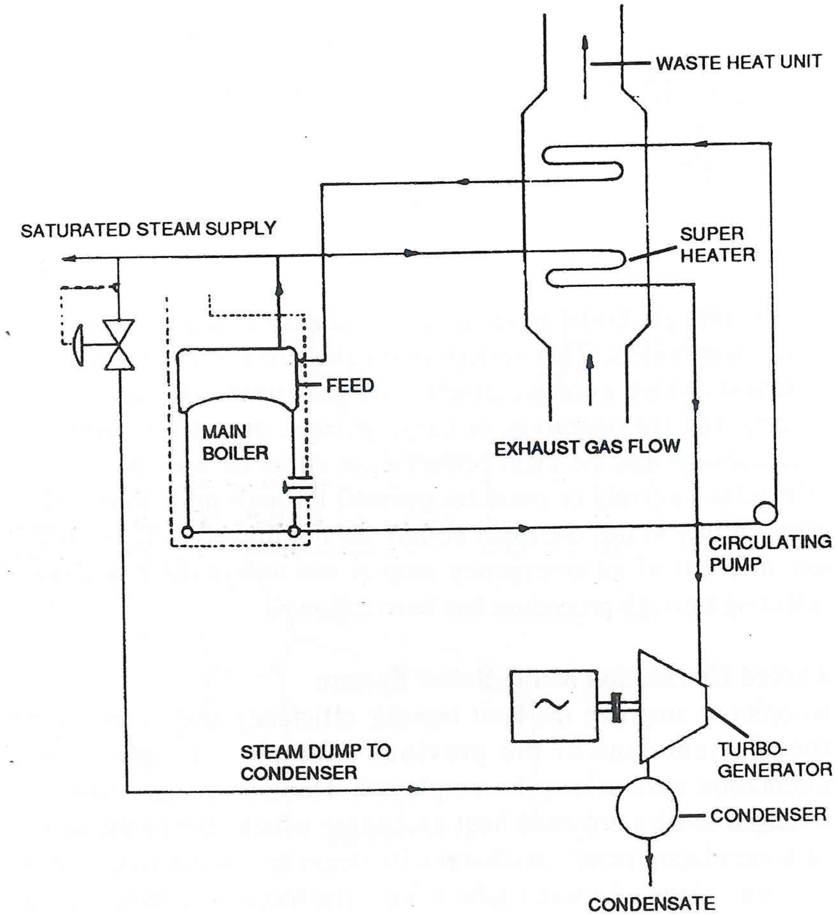


the exhaust gas boiler is secured and all steam supplied by the oil fired main boilers. This system is suitable for use on vessels such as tankers where a comparatively large port steaming capacity may be required for operation of cargo pumps, but suffers from the disadvantage that the main boilers must either be warmed through at regular intervals or must be warmed through prior to reaching port. Further to this the main boilers are not immediately ready for use in event of an emergency stop at sea unless the continuous warming through procedure has been followed.

Forced Circulation Multi-Boiler System

In order to improve the heat transfer efficiency and to overcome the shortcomings of the previous example a simple forced circulation system may be employed. The exhaust gas boiler is arranged to be a drowned heat exchanger which, due to the action of a circulating pump, discharges its steam and water emulsion to the steam drum of a water-tube boiler. The forced circulation pump draws from near the bottom of the main boiler water drum and circulates water at almost ten times the steam production rate so giving good heat transfer. The steam/water emulsion on being discharged into the water space of the main boiler drum separates out exactly in the same way as if the boiler were being oil fired. This arrangement ensures that the main boiler is always warm and

FIG 154
FORCED CIRCULATION WASTE HEAT PLANT AND MAIN BOILER



capable of being immediately fired by manual operation or supplementary pilot operated automatic fuel burning equipment. Feed passes to the main boiler and becomes neutralised by chemical water treatment. Surface scaling is thus largely precluded and settled out impurities can be removed at the main boiler blow-down. If feed flow only is passed through an economiser type unit parallel flow reduces risks of vapour locking. Unsteady feed flow at normal gas conditions can result in water flash over to steam and rapid metal temperature variations. Steam, hot water and cold water conditions can cause thermal shock and water hammer.

Contra flow designs are generally more efficient from a heat transfer viewpoint giving gas temperatures nearer steam temperature and are certainly preferred for economisers if circulation rate is a multiple of feed flow. The generation section is normally parallel flow and the superheat section contra flow. Output control could be arranged by output valves at two different levels so varying the effective heat transfer surface utilised. In addition a circulating pump by-pass arrangement gives an effective control method.

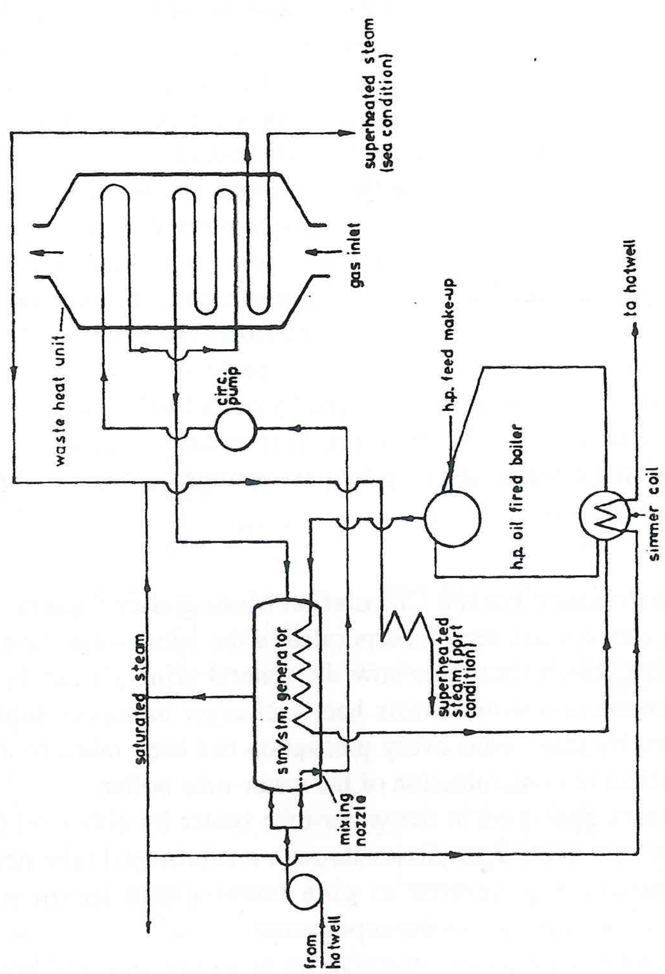
Dual Pressure Forced Circulation Multi-Boiler System

This concept has been incorporated in the latest waste heat circuits and the sketch illustrates how the general principle can be applied in conjunction with a waste heat exchanger to supply superheated steam. By this means every precaution has been taken to minimise the effect of contamination of the water-tube boiler.

Steam generated in the water-tube boiler by either oil firing or waste heat exchanger passes through a submerged tube nest in the steam/steam generator to give lower grade steam which is subsequently passed to the superheater.

A water-tube boiler, steam/steam generator and feed heater may be designed as a packaged unit with the feed heater incorporated in the steam/steam generator. The high pressure high temperature system at say 10 bar will supply a turbo generator for all electrical services while the low pressure system at say 2½ bar would provide all heating services. Obviously the dual system is more costly. Numerous designs are possible including separate lp and hp boilers, either natural or forced circulation, indirect systems with single or double feed heating, etc.

FIG 155
DUAL PRESSURE SYSTEM



WATER/WATER HEAT EXCHANGERS

Evaporators

The basic information given on evaporators in Volume 8 should *first* be considered. In motor ship practice efficient single effect units incorporating flexible elements and controlled water level are in service. Evaporators utilising jacket cooling water as the heating medium producing an output of 20-25 tonnes per day are common. Flash evaporators have increasingly been fitted on large vessels utilising multi-stage units. Also multiple effect evaporators of conventional form are used. The steam circuit of many modern motor ships has developed in complexity to approach successful steam ship practice.

FEED HEATING

The advantages of pre-heating feed water are obvious. Three methods will be considered, namely: economisers, mixture and indirect. Economiser types have been included in previous discussion and sketches. It is sufficient to repeat that such systems require a careful design to cope with fluctuations of steam demand and that particular attention is necessary to ensure protection against corrosive attack. Mixture systems employ parallel feeding with circulating pump and feed pump to the economiser inlet. Such circuits require careful matching of the two pumps and control has to be very effective to prevent cold water surges leading to reducing metal temperatures and causing corrosion. Indirect systems require a water/water exchanger feed heater.

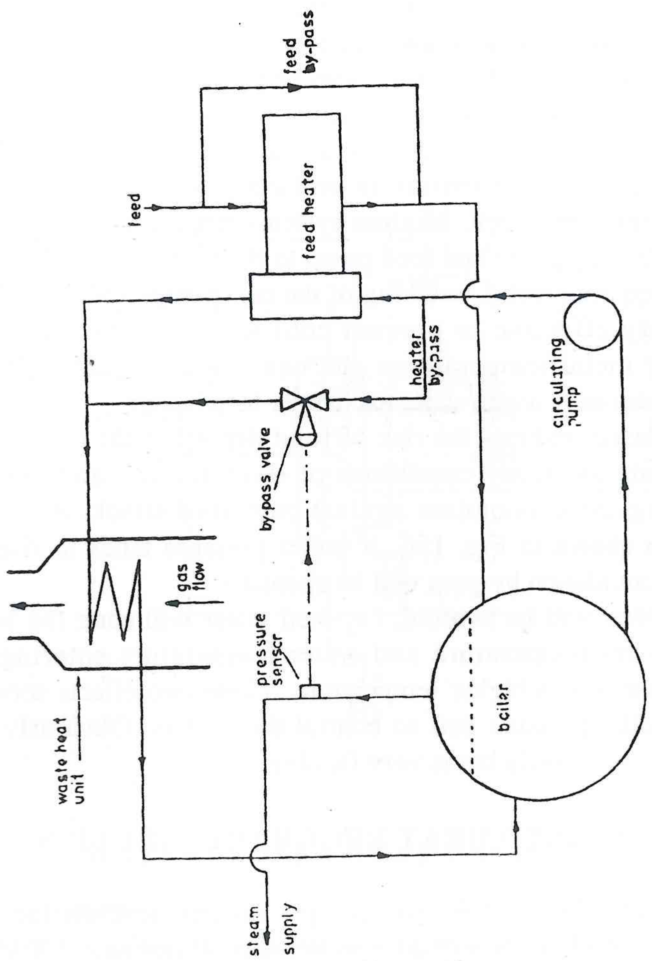
This design reduces the risk of solid deposit in the economiser and maintains steady conditions of economiser water flow so protecting the economiser against corrosive attack. A typical system is shown in Fig. 156. If boiler pressure tends to rise too high the circulation by-pass will be opened.

The effect will be twofold, *i.e.* feed water will enter the boiler at the lower temperature and water temperature entering the economiser is at a higher temperature. These two effects serve to reduce boiler pressure and so control the system. Obviously this system is more costly but is very flexible.

COMBINED HEAT RECOVERY CIRCUITS

The low grade heat of engine coolant systems restricts the heat recovery in such secondary circuits to temperatures near 7-8°C. As such it is normally restricted to use with distillation plants. Combined or compound units involving combination between engine coolant and exhaust gas systems are complicated by the need to prevent contamination and utilise the large volume of low temperature coolant in circulation. Jacket water coolant temperatures have increased in recent motorship practice but even if the engine design can be modified to suit even higher temperatures there is always a problem of high radiation heat loss from jackets to confined engine rooms.

FIG 156
INDIRECT FEED HEATING



CHAPTER 10

MISCELLANEOUS

CRANKCASE EXPLOSIONS

Introduction

The student should first refer to Volume 8 for a consideration of spontaneous ignition temperatures and also limits of inflammability in air. Crankcase explosions have occurred steadily over the years with perhaps that of the *Reino-del-Pacifico* in 1947 the most serious of all. In fact crankcase explosions have occurred in all types of enclosed crankcase engines, including steam engines. Explosions occur in both trunk piston types and in types with a scraper gland seal on the piston rod. Much research has been done in this field but the difficulties of full experimentation utilising actual engines under normal operating conditions is almost impossible to attempt. The following is a simplified presentation based on the mechanics of cause of explosion, appropriate DoT regulations and recommendations and descriptive details of preventative and protective devices utilised.

Mechanics of Explosion

1. A hot spot is an essential source of such explosions in crankcases as it provides the necessary ignition temperature, heat for oil vapourisation and possibly ignition spark. Normal crankcase oil spray particles are in general too large to be easily explosive (average 200 microns). Vapourised lubricating oil from the hot source occurs at 400°C, in some cases lower, with a particle size explosive with the correct air ratio (average 6 microns). Vapour can condense on colder regions, a condensed mist with fine particle size readily causes explosion in the presence of an ignition source. A lower limit of flammability of about 50 mg/l is often found in practice. Experiment indicates two separate temperature regions in which ignition can take place, *i.e.* 270°C-350°C and above 400°C.

2. Initial flame speed after mist ignition is about 0·3 m/s but unless the associated pressure is relieved this will increase to about

300 m/s with corresponding pressure rise. In a long crankcase, flame speeds of 3 km/s are possible giving detonation and maximum damage. The pressure rise varies with conditions but without detonation does not normally exceed 7 bar and may often be in the range 1-3 bar.

3. A primary explosion occurs and the resulting damage may allow air into a partial vacuum. A secondary explosion can now take place, which is often more violent than the first followed by similar sequence until equilibrium.

4. The pressure generated, as considered over a short but finite time, is not too great but instantaneously is very high. The associated flame is also dangerous. The gas path cannot ordinarily be deflected quickly due to the high momentum and energy.

5. Devices of protection must allow gradual gas path deflection, give instant relief followed by non return action to prevent air inflow and be arranged to contain flame and direct products away from personnel.

6. Delayed ignition is sometimes possible. An engine when running with a hot spot may heat up through the low temperature ignition region without producing flame because of the length of ignition delay period at low temperatures. Vapourised mist can therefore be present at 350-400°C. If the engine is stopped the cooling may induce a dangerous state and explosion. Likewise air ingress may dilute a previously too rich mixture into one of dangerous potential.

7. Direct detection of overheating by thermometry offers the greatest protection but the difficulties of complete surveillance of all parts is prohibitive.

8. A properly designed crankcase inspection door preferably bolted in place, suitably dished and curved with say a 3 mm thickness of sheet steel construction should withstand static pressures up to 12 bar although distorted.

9. There are many arguments for and against vapour extraction by exhauster fans. There is no access of free air to the crankcase and the fan tends to produce a slight vacuum in the crankcase. On balance most opinion is that the use of such fans can reduce risk of explosion. The danger of fresh air drawn into an existing over rich heated state is obvious. On the practical aspect leakage of oil is reduced.

Crankcase Safety Arrangements

The following are based on specific DoT rules:

1. Means should be adapted to prevent danger from the result of explosion.

2. Crankcases and inspection doors should be of robust construction. Attachment of the doors to the crankcase (or entablature) should be substantial.

3. One or more non-return pressure relief valves should be fitted to the crankcase of each cylinder and to any associated gearcase.

4. Such valves should be arranged or their outlets so guarded that personnel are protected from flame discharge with the explosion.

5. The total clear area through the relief valves should not normally be less than 9·13 cm²/m³ of gross crankcase volume.

6. Engines not exceeding 300 mm cylinder bore with strongly constructed crankcases and doors may have relief valve or valves at the ends only. Similarly constructed engines not exceeding 150 mm cylinder bore need not be fitted with relief valves.

7. Lubricating oil drain pipes from engine sump to drain tank should extend to well below the working oil level in the tank.

8. Drain or vent pipes in multiple engine installations are to be so arranged so that the flame of an explosion cannot pass from one engine to another.

9. In large engines having more than six cylinders it is recommended that a diaphragm should be fitted at near mid length to prevent the passage of flame.

10. Consideration should be given to means of detection of over-heating and injection of inert gas.

The above should be self explanatory in view of the previous comments made.

Preventative and Protection Devices

In general three aspects are worthy of consideration, *i.e.* relief of explosion, flame protection and explosive mist detection.

Crankcase Explosion Door

A design is shown in Fig. 157. The sketch illustrates a combined valve and flame trap unit with the inspection door insertion in the middle. The internal section supports the steel gauze element and the spider guide and retains the spindle. The external combined aluminium valve and deflector has a synthetic rubber seal. Pressure setting on such doors is often 1/15 bar (above atmospheric pressure). Relief area and allowable pressure rise vary with the licensing insurance authority but a metric ratio of 1:90 should not normally be exceeded based on gross crankcase volume and this should not allow explosion pressures to exceed about 3 bar.

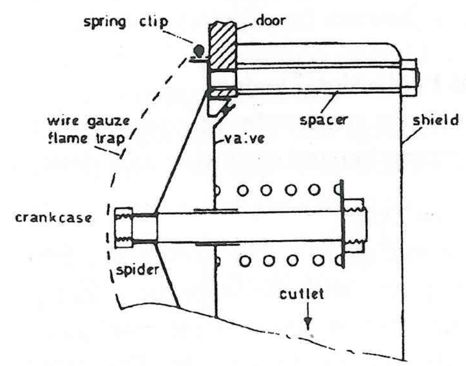
Flame trap
Such devices are advisable to protect personnel. The vented gases can quickly be reduced in temperature by gauze flame traps from say 1500°C to 250°C in 0·5 m. Coating on the gauzes, greases or engine lubricating oil, greatly increases their effectiveness. The best location of the trap is inside the relief valve when it gives a more even distribution of gas flow across its area and liberal wetting with lubricating oil is easier to arrange. A separate oil supply for this action may be necessary. The explosion door in Fig. 157. has an internal mesh flame trap fitted.

Flame traps effectively reduce the explosion pressure and prevent two stage combustion. Gas-vapour release by the operation of an oil wetted flame trap is not usually ignitable. Typical gauze mild steel wire size is 0·3 mm with 40% excess clear area over the valve area.

Crankcase Oil Mist Detector

If condensed oil mists are the sole explosive medium then photo-electric detection should give complete protection but if the

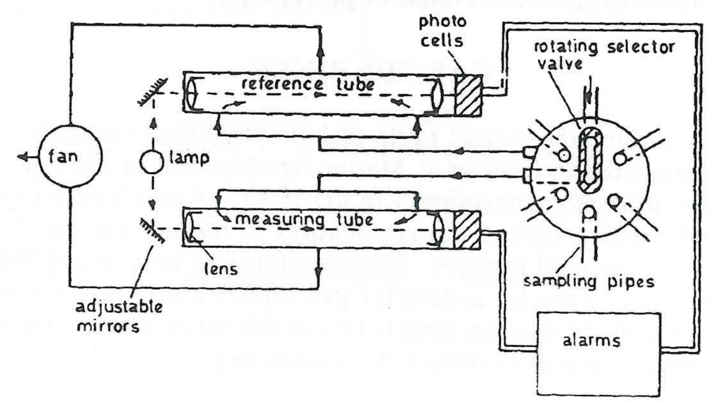
FIG 157
CRANKCASE EXPLOSION RELIEF DOOR



crankcase spray is explosive the mist detection will only indicate a potential source of ignition. The working of one design of detector should be fairly clear from Fig. 158. The photo cells are normally in a state of electric balance, *i.e.* measure and reference tube mist content in equilibrium. Out of balance current due to rise of crankcase mist density can be arranged to indicate on a galvanometer which can be connected to continuous chart recording and auto visual or audible alarms. The suction fan draws a large volume of slow moving oil-air vapour mixture in turn from various crankcase selection points. Oil mist near the lower critical density region has a very high optical density. Alarm is normally arranged to operate at $2\frac{1}{2}$% of the lower critical point, *i.e.* assuming 50 mg/l as lower explosive limit then warning at 1·25 mg/l.

Operation
The fan draws a sample of oil mist through the rotary valve from each crankcase sampling pipe in turn, then though the measuring tube and delivers it to atmosphere. An average sample is drawn from the rotary valve chamber through the reference tube and delivered to atmosphere at the same time. In the event of overheating in any part of the crankcase there will be a difference in optical density in the two tubes, hence less light will fall on the photo cell in the measuring tube. The photo cell outputs will be different and when the current difference reaches a pre-determined value an alarm signal is operated and the slow turning rotary valve

FIG 158
CRANKCASE OIL MIST DETECTOR



stops, indicating the location of the overheating.

Normal oil particles as spray are precipitated in the sampling tubes and drain back into the crankcase.

CO_2 Drenching System

30% by volume of this inert gas is a complete protection against crankcase explosion. This is particularly beneficial during the dangerous cooling period. Automatic injection can be arranged at, say 5% of critical lower mist density but in practice many engineers prefer manual operation. When the engine is opened up for inspection, and repair at hot source, it will of course be necessary to ensure proper venting before working personnel enter the crankcase.

EXHAUST GAS POWER TURBINE

In an effort to improve overall plant efficiency the turbocharger manufacturer ABB has developed a system which exploits surplus exhaust gas in a power turbine and is fed either, to the engine crankshaft or, to an auxiliary diesel or steam turbine generator. Fig. 159. The latter is only feasible if the demand for electricity is greater than the output of the power turbine. This development has been made possible by the improved turbocharger efficiencies achieved in recent years resulting in surplus energy being available in the exhaust gas.

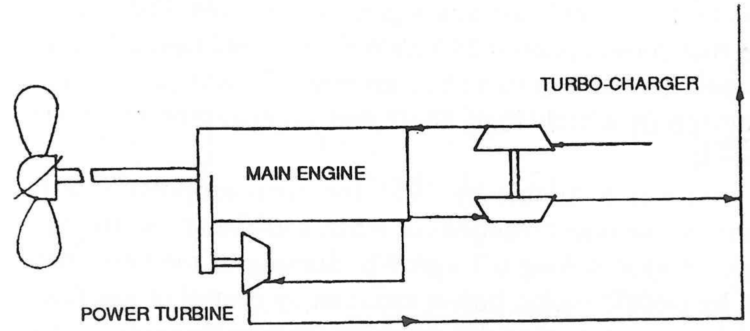
The power turbine can be brought in and out of service, as conditions require, by operating a flap in the exhaust line.

The turbine part of the power turbine is similar to those of turbochargers. The drive from the turbine is via epicyclic gearing and a clutch to the chosen mode of power input.

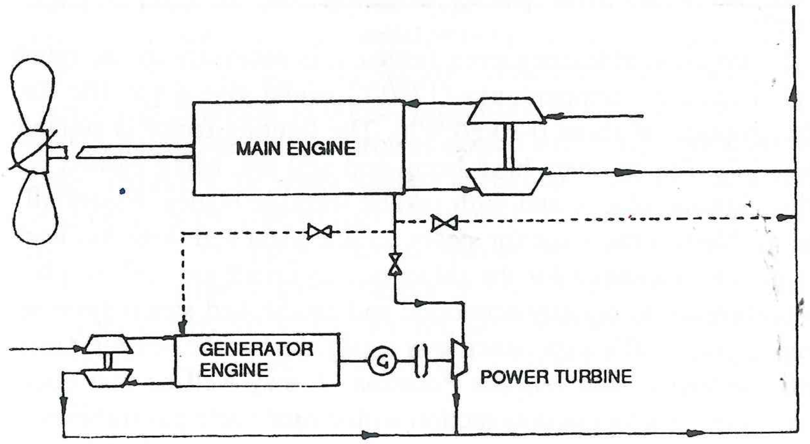
GAS TURBINES

The gas turbine theoretical cycle and simple circuit diagram have been considered in Chapter 1. Marine development of gas turbines stemmed from the aero industry in the 1940's. Apart from an early stage of rapid progress the application to marine use has been relatively slow until recently. Consideration can best be applied in two sections, namely, industrial gas turbines and aero-derived types. In general this can largely be considered as a 'marinisation' of equipment originally designed for other duty.

FIG 159
RECOVERY OF SURPLUS EXHAUST GAS ENERGY IN A POWER TURBINE



POWER TURBINE COUPLED TO MAIN ENGINE



POWER TURBINE COUPLED TO A GENERATOR ENGINE

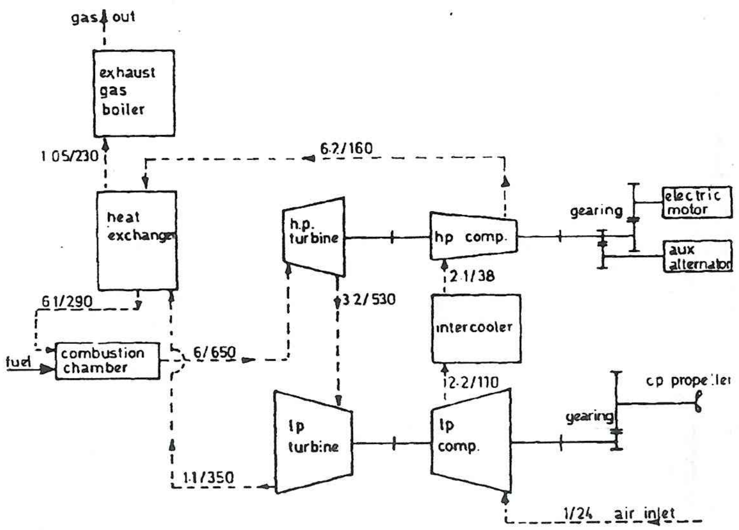
Industrial Gas Turbines

The simplest design is a single-shaft unit which has low volume and light weight (5 kg/kW at 20,000 kW). Fuel consumption (specific) may be about 0·36 kg/kWh on residual fuels. This consumption is not normally acceptable for direct propulsion and initial usage was as emergency generators in MN practice and the RN for small vessels or as boost units in larger warships. Compared to steam turbines (32% output, 58% condenser loss) the simple gas turbine (24% output, 73% exhaust loss) is less efficient but the addition of exhaust gas regeneration gives 31% output (specific fuel consumption 0·28 kg/kWh) and combined RN units 36% output. Normally a two-shaft arrangement was preferred in MN practice in which load shaft and compressor shaft are independent.

A design was available by 1955 for main propulsion with maximum turbine inlet conditions of 6 bar, 650°C and specific fuel consumption approaching 0·3 kg/kWh. Starting of the twin-shaft unit was by electric motor, power variation by control of gas flow, conventional gear reduction and propeller drive by hydraulic clutch with astern torque converter (more modern practice uses variable pitch propeller). Turbine and turbo-compressor design utilised standard theory and simple module construction utilising horizontally split casings, diffusers, etc., and easily accessible nozzles.

To improve efficiency even further it is necessary to use much higher inlet gas temperatures (1200°C would give a specific fuel consumption of about 0·2 kg/kWh). The limiting factor is suitable materials. Experiments have been, and still are, being carried out with ceramic blades and with cooled metallic blades. Essentially the problem is the same for steam turbine plant and there has been no marked incentive for the shipowner to install gas turbine plant in preference to equally economic and established steam systems. During the 1960's experience was established in the vessels *Auris, John Sergeant* and *William Paterson*. It may well be that direct gas cooled reactors in conjunction with closed cycle gas turbines in electric power generation may be an attractive possibility in Nuclear technology. G.E.C. produce a wide range (4000-50,000 kW) of industrial gas turbines now effectively marinised for marine propulsion. In addition to reliability, easy maintenance, low

FIG 160
OPEN CYCLE MARINE GAS TURBINE SYSTEM



volume, etc., the very easy application to electric drives and to automation make the units attractive. Geared drive usually utilises locked train helical gears or alternatively epicyclic gearing.

C.P. propeller development has also broadened the possibilities of various propulsion systems, including geared diesel – gas turbine systems. Marine gas turbines do run with a high noise level and they require to be water washed at regular intervals, the latter depending upon the type of fuel being used. The recently changing design of ships has meant that the owner, or operator, needs to analyse propulsion systems carefully for all economic factors, which vary greatly for VLCC, Ro-Ro, LNG, container vessels, etc. Gas turbines have been exclusively adopted for RN surface vessels.

Aero-Derived

Apart from RN units so derived from aero gas turbines the first British MN vessel so engined was the g.t.s. *Euroliner* in 1970. Turbo Power and Marine Systems Inc. twin gas turbines, 22, 500 kW each at 3600 rev/min drive separate screw shafts at 135

FIG 161
NITROGEN CHEMICAL ACTIVITY INCREASING WITH TEMPERATURE

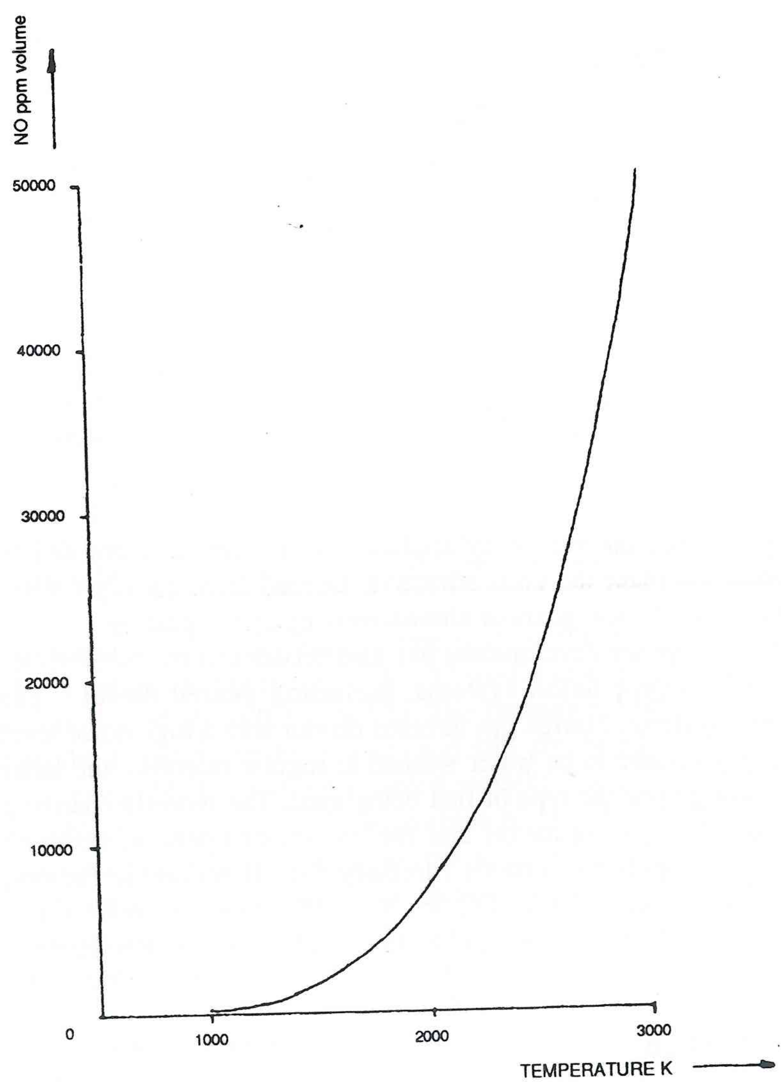
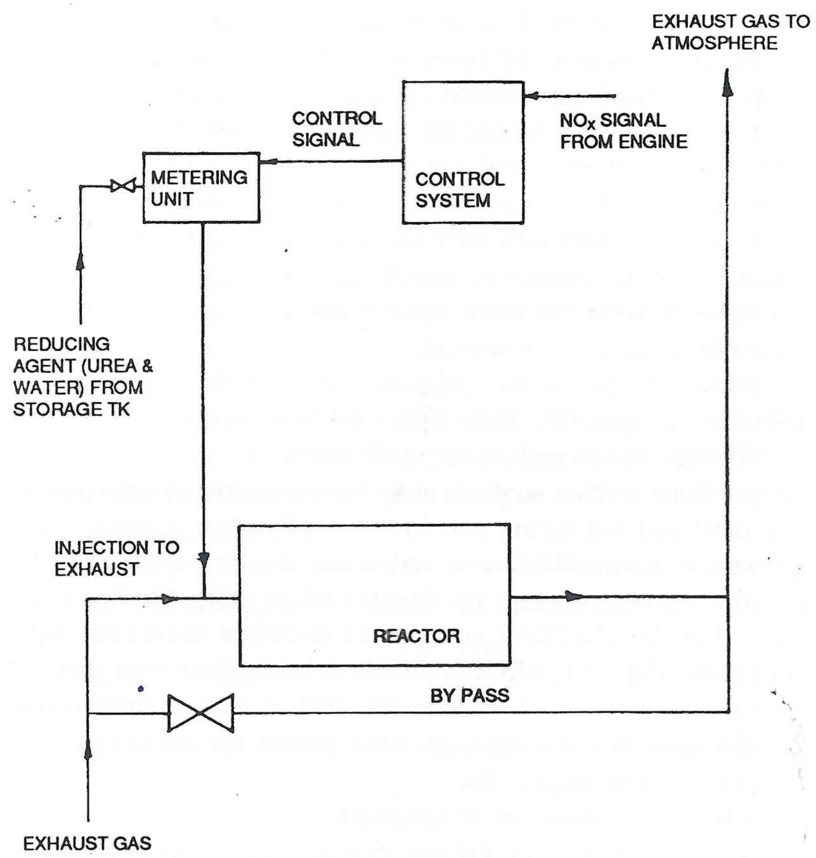


FIG 162
SELECTIVE CATALYTIC REDUCTION (SCR) APPARATUS



rev/min through double reduction locked train gears, with controllable pitch propellers. Main electrical alternators are driven from the gearbox.

Exhaust Gas Emissions

Small quantities of toxic substances such as oxides of sulphur and nitrogen occur in diesel engine exhaust gas. The oxides of sulphur, often referred to as SO_x are formed in quantities proportional to the sulphur content of the fuel. It is likely that the only practical method of reducing the level of SO_x in the exhaust is by consuming fuel of low sulphur content. The oxides of nitrogen, often referred to as NO_x, are the most significant emission from marine diesel engines and comprise nitrous oxide [NO] and nitrogen dioxide [NO_2]. Loaded diesels produce very low amounts of NO_2 directly, over 90% of NO_x being NO. Although the NO produced by diesel engines is considered harmless it oxidises in the atmosphere to form the more environmentally harmful NO_2. The NO_x are formed from two sources:

1. When nitrogen in the fuel combines with the oxygen in the combustion air [possibly 20% of the total NO_x present]
2. Nitrogen and oxygen in the combustion air.

In previous studies students may have considered nitrogen as being inert and not taking part in the combustion process. This, however, is a simplification and when dealing with exhaust emission, nitrogen cannot be thought of as inert. Studies have shown that the chemical activity of nitrogen increases with temperature. Fig. 161. Modern diesels achieve their high thermal efficiency as a result of burning the fuel at high temperatures. Thus, the more efficient the engine the greater the amount of NO_x the engine is likely to generate.

Low NO_x generation can be achieved:

1. By ensuring that the fuel/air mixture is as homogeneous as possible in order to keep the percentage of rich mixture in the combustion chamber small. This requires a nozzle with as many spray holes as possible.
2. By using a high air/fuel ratio.
3. By retarding the fuel injection.

Only the first of these options does not involve a fuel penalty. Raising the air/fuel ratio beyond what is necessary leads to increased pumping losses. Retarding the injection results in lower firing pressures. These measures, however, have the advantage that they do not require extra components to be added to the engine.

Other measures which do require extra components and added cost include.

1. Adding water to the fuel. This works in two ways, it helps to create a homogeneous fuel/air mixture and also reduces the temperature of combustion. The injection of water requires an increase of 20-30% in fuel pump capacity. Precautions must also be taken to maintain the oil/water emulsion in a stable condition and to prevent corrosion of the fuel system components.
2. The use of Selective Catalytic Reduction [SCR].

Selective Catalytic Reduction

SCR technology was developed for land based installations and is being developed for main applications and involves injecting small amounts of a single atom nitrogen based additive, such as ammonia [NH_3], into the exhaust. Because of the difficulties and dangers of handling ammonia on board ship a safer more easily handled ammonia compound called urea {2[NH_2]CO} is being used. The principle is to combine the nitrogen atoms of the NO_x and NH_3 compound to form a stable nitrogen [N_2] molecule which is the main constituent of air.

The principle catalytic reduction process of the ammonia compound is according to the following chemical reaction.

$$2NO + 2NH_3 + 1/2O_2 + CO_2 \xrightarrow{\text{catalysis}} 2N_2 + 3H_2O + CO_2$$

To accomplish this at the temperatures encountered in exhaust gas – 250-450°C a catalyst is required. The catalyst is an oxide of vanadium carried on a heat resistant honeycomb of ceramic. In order to minimise the pressure drop across the reactor the gas passages of the honeycomb core must be of sufficient cross sectional area.

The urea is mixed with water and metered into the exhaust gas upstream of the reactor at a rate dependent upon engine load. Fig. 162.

TEST QUESTIONS

(S denotes SCOTVEC questions)

CHAPTER 1– CLASS ONE

1. *(a)* With reference to fatigue of engineering components explain the influence of stress level and cyclical frequency on expected operating life.
   (b) Explain the influence of material defects on the safe operating life of an engineering component. S
   (c) State the factors which influence the possibility of fatigue cracking of a bedplate transverse girder and explain how the risk of such cracking can be minimised.

2. With reference to engine performance monitoring discuss the relative merits of electronic indicating equipment when compared with traditional indicating equipment.

3. As Chief Engineer Officer how would you ascertain if the main engine is operating in an overloaded condition?
   If the engine is overloading what steps would you take to ensure that the engine was brought within the correct operating range?

4. A set of indicator cards suggests that individual cylinder powers of the main engine are not balanced:
   What action would you take to rectify the problem?
   How would you ascertain the accuracy of the cards?

5. Explain, by referring to the theoretical considerations how the efficiency of an IC engine is dependent upon the compression ratio.
   Why does an actual engine power card only approximate to the ideal cycle?

CHAPTER 2– CLASS ONE

1. *(a)* State with reasons the main causes of normal and abnormal cylinder liner wear.
   (b) State the ideal properties of a cylinder oil for use with an engine burning residual fuel. S
   (c) State the possible consequences of operating an engine with a cylinder liner worn beyond normally acceptable limits.

2. *(a)* Describe briefly three methods of crankshaft construction indicating for which type of engine the method is most suitable.
   (b) State the nature of and reasons for the type of finish used at mating surfaces of a shrink fit. S
   (c) Explain:
   (i) Why slippage of a shrink fit can occur.
   (ii) How such slippage may be detected.
   (iii) How slippage may be rectified.

3. During recent months it has been necessary to frequently re-tighten some main engine holding down bolts as the steel chocks have become loose:
   (a) Explain possible reasons for this.
   (b) State with reasons why re-chocking using a different material might reduce the incidence. S
   (c) Explain the possible consequences if the situation is allowed to continue unchecked.

4. *(a)* State, with reasons, why engine air inlet and exhaust passageways should be as large as possible.
   (b) Explain how such passageways can become restricted even when initially correctly dimensioned. S
   (c) Explain the consequences of operating an engine with:
   (i) Restricted air inlet passageways.
   (ii) Restricted exhaust passageways.

5. *(a)* Describe, using sketches if necessary, a main engine chocking system using resin based compounds, explaining how such a system is installed. S

   (b) State the advantages and disadvantages of resin based materials for use as chocks when compared with iron or steel.

CHAPTER 3 – CLASS ONE

1. Bunkers have been taken in a port and a sample is sent to a laboratory for analysis. The vessel proceeds to sea before results of the analysis are obtained. The analysis indicates that the fuel is off specification in a number of respects but the fuel must be used as there is insufficient old oil supply available to enable the ship to reach the nearest port. Explain with reasons what action should be taken to minimise damage and enable safe operation of the engine if the following fuel properties were above or below specified levels: S
   (a) Viscosity.
   (b) Compatibility.
   (c) Sulphur.
   (d) Ignition quality.
   (e) Conradson carbon.
   (f) Vanadium and sodium.

2. *(a)* Describe using sketches a Variable Injection Timing fuel pump and explain how timing is varied whilst the engine is in operation. S
   (b) Explain why it is necessary to adjust the timing of fuel pumps individually and collectively.

3. With respect to residual fuel explain the effects of EACH of the following on engine components, performance and future maintenance, stating any steps which should be taken in order to minimise these effects: S
   (a) High Conradson carbon level.
   (b) Aluminium level of 120 ppm.
   (c) Low ignition quality.
   (d) 450 ppm vanadium plus 150 ppm sodium.

4. *(a)* With reference to 'slow steaming nozzles' as applied to main engine fuel injectors. State with reasons when and why they would be used. S

(*b*) State with reasons the engine adjustments required when changing to a fuel having a different ignition quality. Explain the consequences of not making such adjustments.
(*c*) State the procedures which should be adopted to ensure that main engine fuel injectors are maintained in good operative order indicating what routine checks should be made.

5. With respect to fuel oil:
(*a*) Explain the meaning of term 'ignition quality' and indicate the possible problems of burning fuels of different ignition quality. S
(*b*) State how an engine may be adjusted to deal with different fuels of different ignition qualities.
(*c*) State how fuel structure dictates ignition quality.

CHAPTER 4 – CLASS ONE

1. (*a*) Sudden bearing failure occurs with a turbocharger which has been operating normally until that point. Explain the possible causes if the turbocharger has: S
(i) Ball or roller bearings.
(ii) Sleeve bearings.
(b) State with reasons the measures to be adopted to ensure that future failure is minimised.

2. With respect to turbochargers indicate the nature of deposits likely to be found on EACH of the following and in each case state the possible consequences of operating with high levels of such deposits and explain how any associated problems might be minimised:
(*a*) Air inlet filters.
(*b*) Impeller and volute. S
(*c*) Air cooler.
(*d*) Turbine and nozzles.
(*e*) Cooling water spaces.

3. (*a*) State what is meant by the term *surge* when applied to turbochargers. S
(*b*) State why surging occurs and how it is detected.

(*c*) Explain how the possibility of surging may be minimised.
(*d*) State what action should be taken in the event of a turbocharger surging and explain why that action should not be delayed.

4. It is discovered that delivery of air from a turbocharger has fallen even though engine fuel control has not been changed. State the reasons:
(*a*) The causes of such reduced delivery. S
(*b*) The effects of this reduced air supply on the engine.
(*c*) The immediate action to be taken.
(*d*) How future incidents might be minimised.

5. At certain speed vibration occurs in a turbocharger.
(*a*) State with reasons the possible causes. S
(*b*) Explain how the cause can be detected and corrected.
(*c*) Explain how the risk of future incidents can be minimised.

CHAPTER 5 – CLASS ONE

1. (*a*) State the possible reasons for an engine failing to turn over on air despite the fact that there is a full charge of air in the starting air receiver and explain how the problem would be traced.
(*b*) Explain how the engine could be started and reversed manually in the event of failure of the control system. S
(*c*) Outline planned maintenance instructions which could be issued to minimise the risk of failure indicated in (*a*) and (*b*).

2. Describe the safety interlocks in the air start and reversing system of a main engine.
What maintenance do these devices require?
At what interval would they be tested?

3. As Chief Engineer Officer what standing orders would you issue to your engineering staff regarding preparing the main engines for manoeuvring?

4. Routine watchkeeping reveals that a cylinder air start valve is leaking.
   What are the dangers of continued operation of the engine?
   What steps would you take if the vessel was about to commence manoeuvring?

5. Describe the main engine shutdown devices. How and how often would you test them?
   The shut down system on the main engine fails, immobilising the engine. Checks reveal that all engine operating parameters are normal. What procedures would you, as Chief Engineer Officer, adopt to operate the unprotected engine to enable the vessel to reach port?

CHAPTER 6 – CLASS ONE

1. *(a)* Describe briefly the operation of an electrical or hydraulic main engine governor.
   (b) For the type described indicate how failure can occur and the action to be taken if immediate correction cannot be achieved and the engine must be operated. S

2. Complete failure of the UMS, bridge control and data logging systems has occurred resulting in the need for the main engine to be put on manual control and monitoring:
   (a) State with reasons six main items of data which require to be monitored and recorded manually.
   (b) Explain how a watchkeeping system should be arranged to provide for effective monitoring and control of the main engine. S
   (c) Explain how the staff will be organised to allow the engine to be manoeuvred safely and state the items of plant which will require attention during such manoeuvring.

3. Discuss the relative merits and demerits of hydraulic and electronic main engine governors.

4. Describe, with the aid of a block diagram, a bridge control system for main engine operation.

As Chief Engineer Officer what standing orders would you issue to your engineering staff when the vessel was operating under bridge control?

5. Describe a jacket cooling water system temperature controller. When operating under low load conditions for an extended period how can cylinder liner corrosion be minimised?

CHAPTER 7 – CLASS ONE

1. During a period of manoeuvring it is noticed that difficulty is being experienced in maintaining air receiver pressure:
   (a) State, with reasons, possible explanations.
   (b) Explain how the cause may be traced and rectified. S
   (c) State what immediate action should be taken to ensure that the engine movements required by the bridge are maintained.

2. *(a)* Explain why it is essential to ensure adequate cooling of air compressor cylinders, intercoolers and aftercoolers.
   (b) State, with reasons, the possible consequences of prolonged operation of the compressor if these areas are not adequately cooled. S

3. *(a)* With reference to air receivers explain:
   (i) Why regular internal and external inspection is advisable.
   (ii) Which internal areas of large receivers should receive particularly close attention. S
   (iii) How the internal condition of small receivers is checked.
   (b) Where significant corrosion is found during an internal inspection what factors would you take into account when revising the safe working pressure?

4. It has been found that during recent periods of manoeuvring a number of air start valve bursting discs or cones have failed:
   (a) Explain the possible reasons for this.
   (b) Indicate how the actual cause might be: S
   (i) Detected.
   (ii) Rectified.

5. (a) State why starting air compressor performance deteriorates in service and how such deterioration is detected. S
   (b) Explain the dangers associated with some compressor faults.

CHAPTER 8 – CLASS ONE

1. (a) Explain the advantages and problems of using aluminium in the construction of composite pistons for medium speed engines.
   (b) Briefly describe the removal, overhaul and replacement of a pair of pistons connected to a single crank of a vee-type engine, explaining any problems regarding the bottom end bearings.

2. (a) Explain the advantages of fitting highly rated medium speed engines with double exhaust and air inlet valves.
   (b) State the disadvantages of double valve arrangements. S
   (c) Explain the possible causes of persistent burning of exhaust valves if it is:
   (i) General to most cylinders.
   (ii) Specific to a single cylinder.

3. Explain the problems associated with medium speed diesel exhaust valves when operating with heavy fuel oil.
   How can these problems be minimised?
   (i) By design.
   (ii) By maintenance.

4. Describe a suitable maintenance schedule for one unit of a medium speed diesel engine operating on heavy oil?

5. Describe the torsional vibration of medium speed diesel engine crankshafts.
   Describe, with the aid of sketches, a coupling that will aid the damping of torsional vibration.

CHAPTER 9 – CLASS ONE

1. As Chief Engineer Officer, what standing orders would you issue your engineering staff to ensure that the auxiliary boiler was operated in a safe and efficient manner?

2. Describe a waste heat plant that is able to produce sufficient steam to a turbo-generator to supply the entire ship's electrical load at sea.
   Due to trading requirements the vessel is sailing at reduced speed.
   Describe the steps you would take to ensure the slowest ship's speed commensurate with supplying sufficient steam to the turbo-generator without allowing the boiler to fire or starting diesel generators.

3. Describe, with the aid of sketches, an auxiliary boiler suitable for use with a waste heat unit.
   Explain how the pressure of the steam plant is maintained when operating under low steam load conditions.

4. Sketch and describe a composite thimble tube boiler. Describe how the thimble tubes are fitted and discuss burning of tube ends and other possible defects.

5. You are Chief Engineer Officer of a motor vessel equiped with a steam plant incorporating a waste heat unit in the engine uptake. On passage it is reported to you that the uptake temperature is rising.
   (a) What would this information indicate and what steps would you take?
   (b) How could you prevent a reoccurrence?

CHAPTER 1 – CLASS TWO

1. (a) State the ideal cycle most appropriate to the actual operations undergone in the modern diesel engine.
   (b) Give reasons why the actual cycle is made approximate to the ideal heat exchange process. S
   (c) State how the combustion process in the actual cycle is made approximate to the ideal heat exchange process.

2. *(a)* State why bottom end bolts of 4-stroke engines are susceptible to failure.
   (b) Sketch a bottom end bolt of suitable design.
   (c) Explain how good design reduces possibility of failure. S
   (d) State how the possibility of failure is reduced by good maintenance.

3. *(a)* Explain why in large, slow speed engines, power balance between cylinders is desirable.
   (b) State why it is never achieved in practice.
   (c) Describe how power balance between cylinders of a medium speed engine is improved. S
   (d) Describe how power balance in a slow speed engine is improved.

4. *(a)* Give an example of each of the four types of 2-stroke engine indicator diagrams, explain how each is taken and the use to which it is put.
   (b) Illustrate two defects which can show up on a compression card. S
   (c) How is cylinder power balance checked on a higher speed engine?

5. *(a)* Explain how the power developed in an engine cylinder is determined:
   (i) From indicator cards. S
   (ii) By electronic means.
   (b) State which of these is the most representative and why.

CHAPTER 2 – CLASS TWO

1. *(a)* State TWO reasons why large crankshafts are of semi-built construction.
   (b) State SIX important details of crankshaft construction that will reduce the possibility of fatigue failure. S
   (c) List FOUR operational faults that may induce failure in a crankshaft.

2. *(a)* State the nature of the stresses to which crank webs of large diesel engines are subjected. S
   (b) Explain how they are designed and manufactured to resist these stresses.

3. *(a)* State the reason for fitting crosshead guides to engines and explain why 'ahead' and 'astern' faces are required with uni-directional engines. S
   (b) Describe how crosshead guide clearance is checked and adjusted.
   (c) List reasons for limiting such crosshead clearance.

4. *(a)* State why bedplates of large engines are fitted with chocks rather than directly on foundation plates.
   (b) Sketch an arrangement of lateral chocking showing the position relative to the engine. S
   (c) State why such an arrangement is employed.
   (d) State the factors that determine the spacing of the main chocks.

5. With reference to auxiliary diesel engine machinery:
   (a) (i) State why this may be mounted on resilient mountings.
   (ii) State why such mountings have great flexibility. S
   (b) State why limit stops are provided.
   (c) State how the external piping is connected.

CHAPTER 3 – CLASS TWO

1. *(a)* Describe, with the aid of sketches, a fuel pump capable of variable injection timing.
   (b) State why injection timing might need to be changed. S
   (c) State how injection timing is adjusted whilst the engine is running.

2. *(a)* Sketch and describe a fuel valve for a diesel engine.
   (b) State FOUR factors which indicate that fuel valve(s) require attention. S

3. *(a)* State the factors that influence:
   (i) Droplet size during fuel injection.
   (ii) Penetration. S
   (b) State TWO methods of improving air turbulence.

4. *(a)* Sketch and describe a jerk type fuel pump that is not helix controlled.
   (b) Explain how the pump may be timed. S
   (c) State TWO advantages of this type of pump.

5. *(a)* Sketch a main engine fuel pump of the scroll type.
   (b) Explain how the fuel quantity and timing are adjusted. S
   (c) To what defects is this type of pump subject and how is the pump adjusted to counter their effects?

CHAPTER 4 – CLASS TWO

1. *(a)* Describe with the aid of sketches:
   (i) A pulse turbocharger system.
   (ii) A constant pressure turbocharger system.
   (b) State the advantages and disadvantages of each system in Q.1.*(a)*. for use with marine propulsion engines. S
   (c) In the event of turbocharger failure with one of the systems in Q.1.*(a)*. state how the engine could be arranged to operate safely.

2. *(a)* Sketch a simple valve timing diagram for a naturally aspirated 4-stroke engine.
   (b) Sketch a simple valve timing diagram for a supercharged 4-stroke engine. S
   (c) Comment on the differences between the two above diagrams.

3. *(a)* Sketch and describe a turbocharger with a radial flow gas turbine showing the position of the bearings. S
   (b) State the advantages of radial flow gas turbines.

4. *(a)* State why turbochargers are used to supply air to an engine rather than expanding the gas further in the cylinder and then employing crank driven scavenge pumps.
   (b) Explain what measures should be adopted to ensure safe operation of the engine should all turbochargers be put out of action. S

   (c) State why a 2-stroke cycle engine relies upon a pressurised combustion air supply but a 4-stroke cycle engine does not.

5. *(a)* Explain why air coolers and water separators are fitted to large turbocharged engines. S
   (b) Sketch a water separator, explain how it operates and indicate its positioning in the engine.
   (c) What are the defects to which coolers and separators are susceptible?

CHAPTER 5 – CLASS TWO

1. *(a)* Sketch a starting air distributor used for a large reversible engine.
   (b) Explain how the engine may be started with the crankshaft in any rotational position. S
   (c) Explain how the engine is started on air in either direction.

2. *(a)* Sketch a pneumatically operated starting air valve.
   (b) Explain how the valve is operated. S
   (c) State what normal maintenance is essential and the possible consequence if it is neglected.

3. *(a)* Sketch and describe the reversing system for a large slow speed diesel engine. S
   (b) List the safety devices fitted to the air start system.

4. *(a)* Explain why it is necessary to have air start overlap.
   (b) Show how air start timing is affected by exhaust timing. S
   (c) State why the number of cylinders have to be taken into consideration.

5. *(a)* Sketch an engine air start system from the air receiver to the cylinder valves and describe how it operates. S
   (b) List the safety devices and interlocks incorporated in such a system and state the purpose of each.

CHAPTER 6 – CLASS TWO

1. With reference to a jacket water temperature control system:
   (*a*) Sketch and describe such a system:
   (*b*) (i) Explain how disturbances in the system may arise. S
   (ii) Describe how these disturbances may be catered for.

2. (*a*) Construct a block diagram, in flow chart form, to show the sequence of operations necessary for the starting of a diesel engine on bridge control. S
   (*b*) Identify the safety features incorporated in the system of Q.2.(*a*).

3. (*a*) Sketch a cylinder relief valve suitable for a large engine.
   (*b*) State with reasons why such a device is required.
   (*c*) If the relief valve lifts state the possible causes and indicate the rectifying action needed to prevent engine damage. S
   (*d*) State why the relief valve should be periodically overhauled even though it may never have lifted.

4. With reference to mechanical/hydraulic governors explain:
   (*a*) Why the flyweights are driven at a higher rotational speed than the engine.
   (*b*) How dead band effects are reduced. S
   (*c*) How hunting is reduced.
   (*d*) How the output torque is increased.

5. Sketch and describe a hydraulic governor with proportional and reset action.

CHAPTER 7 – CLASS TWO

1. (*a*) Sketch a jacket water cooling system.
   (*b*) State why chemical treatment of the jacket cooling water is necessary. S
   (*c*) Describe how the correct concentration of the chemicals in the jacket water cooling system may be determined.

2. (*a*) Explain how oil may become mixed with starting air and state the attendant dangers.
   (*b*) Describe how this contamination may be reduced or prevented. S
   (*c*) Outline the dangers of lubricating oil settling in air starting lines.
   (*d*) How may an air start explosion be initiated?

3. (*a*) Explain why air compression for starting air duties is carried out in stages and why those stages are apparently unequal. S
   (*b*) What is the purpose of an intercooler and explain why it is important that it is kept in a clean condition?
   (*c*) What is the significance of clearance volume to compressor efficiency?
   (*d*) What is bumping clearance and how is it measured?

4. (*a*) State why compressor suction and delivery valves should seat promptly.
   (*b*) Explain the effect on the compressor if the air is induced into the cylinder at a temperature higher than normal. S
   (*c*) What would be the effect of the suction valves having too much lift.
   (*d*) Explain why pressure relief devices are fitted to the water side of cooler casings.

5. (*a*) State why inhibitors are employed with engine cooling water even though distilled water is used for that purpose.
   (*b*) State the merits and demerits of the following inhibitors used in engine cooling water systems: S
   (i) Chromate.
   (ii) Nitrite-borate.
   (iii) Soluble oil.
   (*c*) Briefly explain how each inhibitor functions.

CHAPTER 8 – CLASS TWO

1. Describe, with the aid of sketches, an exhaust valve of a medium speed diesel engine suitable for use with heavy fuel oil. Explain the procedure adopted when overhauling this valve.

2. Describe, with the aid of sketches, a piston suitable for use in a medium speed engine. Why is aluminium being generally superceded for pistons on highly rated medium speed engines.

3. Describe with the aid of sketches a system for main propulsion in which two medium speed diesel engines are coupled to a single propeller.

4. Describe the advantages and disadvantages of medium speed diesel engines compared to large slow running engines.

5. Explain why lubricating oil consumption is greater in medium speed engines than in slow running diesels and the steps taken to minimise the consumption.

CHAPTER 9 – CLASS TWO

1. *(a)* Describe, with the aid of sketches, an arrangement for producing electricity using steam generated from waste heat.
   (b) State how electricity can be generated with the system in Q.1.*(a)* when the engine is not operating. S
   (c) State the circumstances which could lead to an emergency shut-down of the steam plant in Q.9.*(a)* and the use of diesel engines for electrical generation.

2. Describe the inspection of an auxiliary boiler.
   What precautions should be taken prior to entering the boiler?

3. Describe, with the aid of sketches a boiler which may be alternatively fired or heated with main engine exhaust gas in which the heating surfaces are common. Describe the change over arrangements and state any safety devices fitted to this gear.

4. What are the precautions that should be taken before and during the "flashing up" operation of an auxiliary boiler?
   State the checks carried out on the boiler when a fire is established.

5. What are the advantages and disadvantages of forced circulation and natural circulation multi-boiler installations?
   How can the steam pressure of the waste heat plant be controlled when operating on exhaust gas?

6. Describe the dangers of dirty uptake in the waste heat unit.
   Explain how these dangers are minimised.

SPECIMEN QUESTIONS – CLASS ONE

1. *(a)* Define the term *hot spot.*
   (b) State SIX specific areas in a diesel engine where hot spots have occurred. S
   (c) State other factors that may contribute to the occurrence of a crankcase explosion.

2. With reference to crankcase explosions state:
   (a) The conditions that may initiate an explosion.
   (b) What may cause a secondary explosion. S
   (c) How a crankcase explosion relief valve works.

3. *(a)* State the basic processes leading up to a crankcase explosion and explain how a secondary explosion can occur. S
   (b) List with reasons the precautions which can be taken to minimise the risk of a crankcase explosion occurring.

4. *(a)* Explain how a primary crankcase explosion is caused and how it may trigger a secondary explosion.
   (b) Indicate the possible benefits or dangers of the following features on the likely development of a crankcase explosion: S
   (i) Oil mist detector.
   (ii) Inert gas injection.
   (iii) Infra-red heat detectors.
   (iv) Bearing shells having a layer of bronze between the white metal and steel backing steel.

5. *(a)* Describe, using sketches if necessary, the procedure for complete inspection of a propulsion engine main bearing and journal.
   (b) State the possible bearing and pin defects which might be encountered. S
   (c) State what precautions should be taken before returning an engine to service following such bearing inspection and adjustment.

6. *(a)* Explain the reason for fitting crossheads and guides to large slow speed engines.
   (b) Explain:
   (i) Why guide clearance is limited. S
   (ii) How guide clearance is adjusted.
   (iii) How guide alignment is checked.

7. *(a)* During an inspection it is noticed that tie rods of certain main engine units have become slack, state with reasons the possible causes of this.
   (b) Explain how correct tension is restored and the risk of future slackness minimised. S
   (c) A tie rod has fractured and cannot be replaced immediately. State with reasons the course of action to be adopted in order to allow the engine to be operated without further damage.

8. *(a)* Explain the term *fuel ignition quality* and indicate how a fuel's chemical structure influences its value.
   (b) State, with reasons, the possible consequences of operating an engine on a fuel with a lower ignition quality than that for which it is timed. S
   (c) (i) Explain how an engine might be adjusted to burn fuel of different ignition quality.
   (ii) State what checks can be carried out in order to determine that the engine is operating correctly.

9. *(a)* Describe the phenomenon of surging as applied to turbochargers.
   (b) Explain why turbochargers are not designed to completely eliminate the possibility of surging. S
   (c) State with reasons the possible consequences of allowing a turbocharger to continue to operate whilst it is surging.

10. With reference to turbocharger systems state how deposit build-up might be detected on the following parts and explain the consequences on turbocharger and engine operation of excessive deposits: S
   (a) Suction filter.
   (b) Impeller.

   (c) Turbine nozzle and blades.
   (d) Air cooler.

11. Difficulty is experienced in starting an engine even though there is full air pressure in the air receivers and fuel temperature is correct. Explain how the cause of the problem can be: S
   (a) Detected.
   (b) Rectified.

12. With reference to piston ring and liner wear:
   (a) State, with reasons, the causes of abnormal forms of wear known as cloverleafing and scuffing (microseizure);
   (b) Explain how cylinder lubrication in terms of quantity and quality can influence wear; S
   (c) Describe the procedure for determining whether piston rings are suitable for use.

13. With reference to main engine holding down studs/bolts:
   (a) Explain the causes of persistent slackening.
   (b) State, with reasons, the likely consequences of such slackening. S
   (c) Describe how future incidents of slackening might be minimised.

14. *(a)* Inspection of an engine indicates an unexpected increase in cylinder liner wear rate, state with reasons the possible causes if:
   (i) The problem is confined to a single cylinder.
   (ii) The problem is common to all cylinders. S
   (b) Explain how cylinder wear rate may be kept within desired limits and indicate the instructions to be issued to ensure that engine room staff are aware as to how this can be achieved.

15. Cracks have been discovered between the crankpin and web on a main engine crankshaft:
   (a) Describe action to be taken in order to determine the extent of the cracking. S

(*b*) Explain the most likely reasons for the cracking.
(*c*) State, with reasons, the action to be taken in order that the ship may proceed to a port where thorough inspection facilities are available.

16. It is found that tie rods are persistently becoming slack:
(*a*) State, with reasons, the possible causes.
(*b*) State, with reasons, the likely effects on the engine if it is allowed to operate with slack tie rods. S
(*c*) Explain how this problem can be minimised.

17. As Chief Engineer Officer, explain the procedure to be adopted for the complete inspection of a main engine cylinder unit emphasising the areas of significant interest. S

18. (*a*) The water jacket on a turbocharger casing has fractured allowing water into the turbine side. State possible reasons for this.
(*b*) Explain how the engine may be kept operational and the restrictions now imposed upon the operating speed. S
(*c*) State how the fracture can be rectified and how future incidents can be minimised.

19. (*a*) State the conditions which could result in a fire in the tube space and/or uptakes of a waste heat boiler.
(*b*) State how such conditions can occur and how the risk of fire can be minimised. S
(*c*) State how such fires can be dealt with.

20. As Chief Engineer, explain the procedure to be adopted for the survey of an air compressor on behalf of a classification society. S

21. (*a*) Identify, with reasons, the causes and effects of misalignment in large, slow speed, engine crankshafts.
(*b*) Describe how the alignment is checked. S
(*c*) State how the measurements are recorded and checked for accuracy.

22. (*a*) Explain why side and end chocking arrangements are provided for large direct drive engines.
(*b*) State, with reasons, why non-metallic chocking is considered superior to metallic chocking. S
(*c*) State why top bracing is sometimes provided for large engines and explain how it is maintained in a functional condition.

23. (*a*) As Chief Engineer, describe how a complete inspection of a main engine turbocharger may be carried out indicating, with reasons, the areas requiring close attention. S
(*b*) Describe defects which may be found during inspection and their possible cause.

24. The main engine has recently suffered problems related to poor combustion and inspection indicates that a number of injector nozzles are badly worn:
(*a*) Explain the possible causes of the problem and how they may be detected. S
(*b*) State how future problems of a similar nature can be minimised.

25. With reference to fuel pumps operating on residual fuel:
(*a*) (i) State, with reasons, the defects to which they are prone.
(ii) Explain the effects of such defects on engine performance. S
(*b*) State, with reasons corrective action necessary to restore a defective fuel pump to normal operation.
(*c*) Suggest ways in which the incidence of these defects might be minimised.

SPECIMEN QUESTIONS – CLASS TWO

1. Describe the routine maintenance necessary on the following components in order to obtain optimum performance from a main engine turbocharger: S
(*a*) Lubricating oil for ball bearings.
(*b*) Air intake silencer/filter.

(c) Turbine blades.
(d) Diffuser ring.

2. (a) List the advantages of multi-stage air compression with intercooling compared with single stage compression.
(b) Explain the faults which may be encountered during overhaul of the H.P. stage and indicate how they may be rectified. S

3. (a) Outline the problems associated with air compressor cylinder lubrication indicating why it should be kept to a minimum.
(b) State why a restricted suction air filter might make the situation worse and lead to the possibility of detonation in the discharge line. S
(c) Explain why the compressor discharge line to the air receiver should be as smooth as possible with the minimum number of joints and connections.

4. (a) Explain the need for additives in engine jacket water cooling systems. S
(b) State what factors determine the choice of chemicals used.
(c) State why chromates are seldom used.

5. (a) Give a simple line sketch of a jacket water cooling system.
(b) Describe a control system capable of maintaining the jacket water temperature within close limits during wide changes in engine load. S

6. (a) Sketch an arrangement for securing turbocharger blades to the blade disc.
(b) How is blade vibration countered?
(c) What is the cause of excessive turbocharger rotor vibration? S
(d) Briefly describe an in-service cleaning routine for the gas side of a turbocharger.

7. (a) Describe with sketches a scroll type fuel pump.
(b) Explain how the quantity of fuel is metered and how the governor cut out functions. S

(c) State how this type of pump is set after overhaul.
(d) State the reasons that necessitate pump overhaul.

8 (a) Sketch a fuel injector.
(b) Explain how it operates and what determines the point at which injection occurs. S
(c) Describe the defects to which injectors are prone.
(d) How can injection be improved when a low speed engine is to operate at prolonged low load?

9. With reference to turbocharging:
(a) (i) Explain the terms pulse system and constant pressure system.
(ii) List the advantages of each.
(b) State how in a pulse system the exhaust from one cylinder may be prevented from interfering with the scavenging of another. S
(c) State why electrically driven blowers are usually fitted in addition to turbochargers.

10. (a) Show how combustion forces are transmitted to the cross members of the bedplate. S
(b) Describe TWO means by which the stresses within the cross members can be accommodated.

11. (a) Describe how crankshaft alignment is checked.
(b) Identify, with reasons the causes of crankshaft misalignment. S
(c) State how the measurements are recorded.

12. (a) Sketch a cross-section of a main engine structure comprising bedplate, frames and entablature showing the tie bolts in position.
(b) Explain why tie bolts need to be used in some large, slow speed engines. S
(c) Explain in detail how the tie bolts are tensioned.

13. Give reasons why, when compared to the other bearings of large slow speed engines, top end bearings: S

(*a*) Are more prone to failure.
(*b*) Have a greater diameter in proportion to pin length.

14. (*a*) State how engine cylinder power is checked and approximate power balance is achieved.
(*b*) Explain why the methods of checking may differ between slow and higher speed engines. S
(*c*) State why perfect cylinder power balance cannot be achieved.
(*d*) State the possible engine problems resulting from poor cylinder power balance.

15. (*a*) Describe with sketches the mono-box frame construction which is being used to replace the traditional A-frame arrangement for some crosshead engines. S
(*b*) State why this form of construction is considered to be more suitable than one using A-frames.

16. (*a*) State TWO reasons why large crankshafts are of semi-built or fully built construction.
(*b*) State SIX important details of crankshaft construction that will reduce the possibility of fatigue failure. S
(*c*) State FOUR operational faults that may induce fatigue failure.

17. (*a*) Define the cause of corrosive wear on cylinder liners and piston rings.
(*b*) Explain the part played by cylinder lubrication in neutralising this action. S
(*c*) State how the timing, quantity and distribution of cylinder oil is shown to be correct.

18. With reference to large fabricated bedplates give reasons to explain:
(*a*) Why defects are likely to occur in service and where they occur. S
(*b*) How these defects have been avoided in subsequent designs.

19. (*a*) Define the cause of cylinder liner and piston ring wear. S

(*b*) Describe how cylinder liner wear is measured and recorded.
(*c*) Explain the possible consequences of operating a main engine with excessive cylinder liner wear.

20. (*a*) Sketch a main engine holding down arrangement employing long studs and distance pieces.
(*b*) Explain why the arrangement sketched in Q.6.(*b*). may be employed in preference to short studs. S
(*c*) Describe, with the aid of sketches, how transverse movement of the bedplate is avoided.

21. (*a*) Briefly discuss the relative advantages and disadvantages of oil and water for cooling.
(*b*) Sketch a piston for a large two stroke crosshead engine indicating the coolant flow. S
(*c*) State the causes of piston cracking and burning, and how it can be avoided.

22. (*a*) Sketch the arrangement of a large 2-stroke engine cylinder liner in position in the cylinder block.
(*b*) Describe how jacket water sealing is accomplished between liner and cylinder block. S
(*c*) For the liner chosen illustrate the directions of cooling water flow, exhaust gas flow and combustion air flow.
(*d*) Explain how thermal expansion of the liner is accommodated.

23. (*a*) Give the reasons for progressive 'fall-off' of piston ring performance in service.
(*b*) State, with reasons, which ring clearances are critical. S
(*c*) State what effects face contouring, bevelling, ring cross section and material properties of rings and liners have on ring life.

24. (*a*) Sketch the arrangement for connecting a piston to the crosshead.
(*b*) State the type of piston coolant employed and show how the coolant is directed to and from the piston. S
(*c*) State the precautions to be exercised when lifting or overhauling the piston described.

25. (*a*) Explain the reasons for employing two air inlet and two exhaust valves for high powered trunk piston 4-stroke engines.
(*b*) State the problems relating to tappet setting with such valves.
(*c*) Sketch a caged valve as fitted to a trunk piston engine.

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