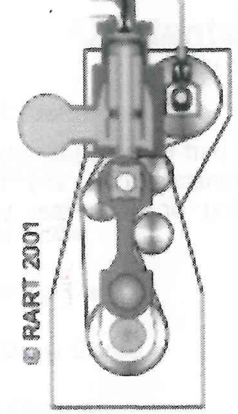

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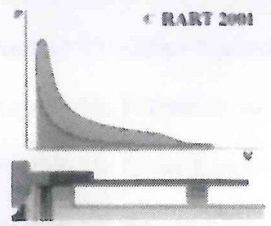

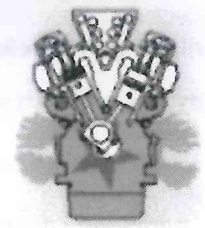

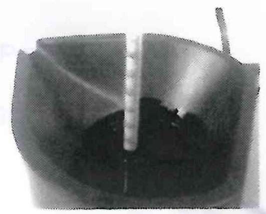

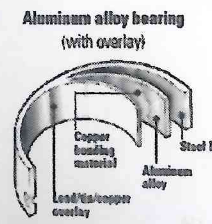

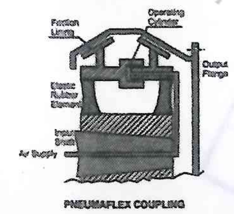

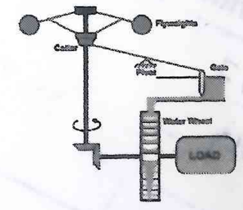

Indicator Cards & Power Assessment
Crankcase Explosions
Fuel Oil
Lubricating Oil and Bearings
Clutches and Couplings and Gearing
Governors

AUTHORS NOTE:

Although every care has been taken in the preparation of these study guides, no responsibility is taken for any error or omission.

For more information on the running and maintenance of marine diesel engines, visit

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or

http://www.marinediesels.info

Indicator Cards and Power Assessment

Contents

Chapter 1: The Ideal Diesel Cycle

The indicator diagram is a graph of Pressure against Volume as the piston moves up and down in the cylinder during a cycle of operations.

THE AIR STANDARD DIESEL CYCLE

In thermodynamics, reference is made to the Air Standard Diesel Cycle. It must be made clear that this does not represent what actually happens inside the cylinder of a working diesel engine, but was developed to be able to model engine behaviour and to analyse the effects of certain changes to the cycle without the complications of modelling an actual engine cycle.

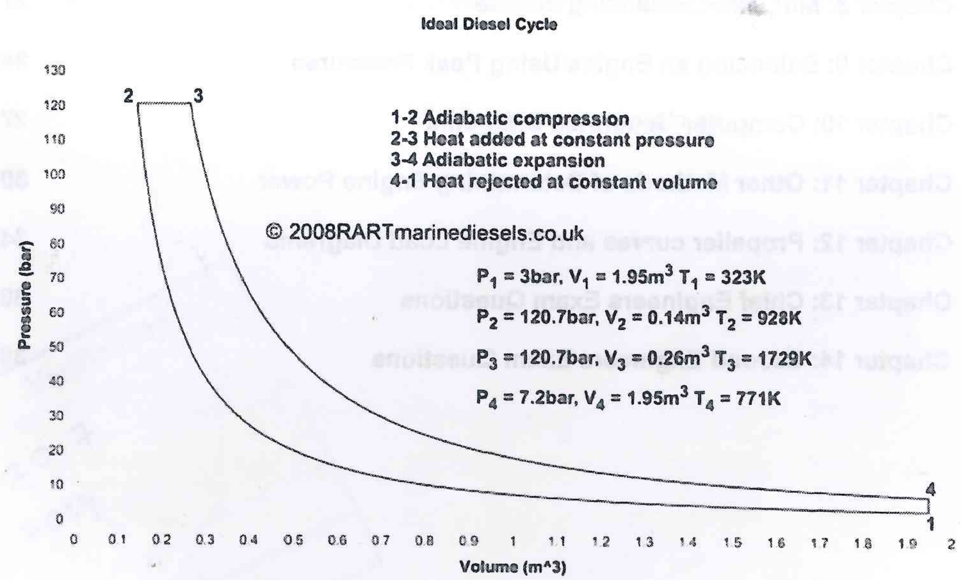
Like an indicator diagram it is a graph of Pressure against Volume, but it makes certain assumptions:

The mass of air within the cylinder remains the same: There are no inlet or exhaust valves.

Fuel is not injected and burnt, but heat energy is added at constant pressure.

At the end of the cycle the heat energy is rejected at constant volume.

The compression and expansion of the air is adiabatic (no heat energy is added or lost) and follows the law $PV^{\gamma} = C$ where γ for air is 1.4.



Enter Engine Speed in rpm	100
Enter Piston stroke in m	2.5
Enter Bore in m	0.96
Enter con Rod Length in m	2.88
Enter Compression Ratio	14
Enter Scavenge Pressure in Bar	2
Enter Scavenge Temp in °C	50
Enter index of comp & exp ' '	1.4
Enter angle of heat addition	25

Work under expansion curve	4332 72
Work under Constant Pressure line	1450 067
Work under Compression curve	2738 63
Work Done/rev (kJ)	3044 158
Power Developed (kW)	5073 596
Mean Pressure Bar	16 82266
Thermal efficiency %	59 98171

The graph above is of an ideal diesel cycle for a cylinder of bore 960mm and stroke of 2.5m. Note that the *x* axis is in m^3, not crank angle. However the axis could have also been the stroke of the piston in m, and the diagram would have had the same shape.

Because the diagram is a mathematical model, the temperature, pressure and volume at any point in the cycle can be calculated.

The area of the diagram using the units of the graph is equal to the work done in the cycle. To calculate this value:

The area between the horizontal line between points 2 and 3 and the *x* axis represents the work done during the heat addition at constant pressure. This is calculated using the formula:

$$P_2(V_3 - V_2)100 \text{ (kJ)}.$$

This area is added to the area between the expansion curve (points 3 to 4) and the *x* axis which is calculated using the formula:

$$\frac{100(P_3V_3) - 100(P_4V_4)}{\gamma - 1}$$

The area under the compression curve (points 1 to 2) and the *x* axis is now calculated using the formula

$$\frac{100(P_2V_2) - 100(P_1V_1)}{\gamma - 1}$$

and this value is subtracted from the sum of the other two areas. This will give the area enclosed by the diagram.

If the work done is divided by the swept volume of the cylinder, then the mean (or average) pressure of the diagram is given. Similarly multiplying the mean pressure by the swept volume will give the work done per cycle.

The Thermal efficiency of the cycle is given by the fomula:

$$1 - \frac{\text{heat rejected}}{\text{heat supplied}} = 1 - \frac{1}{\lambda}\left\{\frac{T4 - T1}{T3 - T2}\right\} \text{ or } 1 - \frac{1}{\gamma} \times \frac{1}{r^{\gamma - 1}}\left\{\frac{\rho^{\gamma} - 1}{\rho - 1}\right\}$$

where:

r = compression ratio V_1/V_2

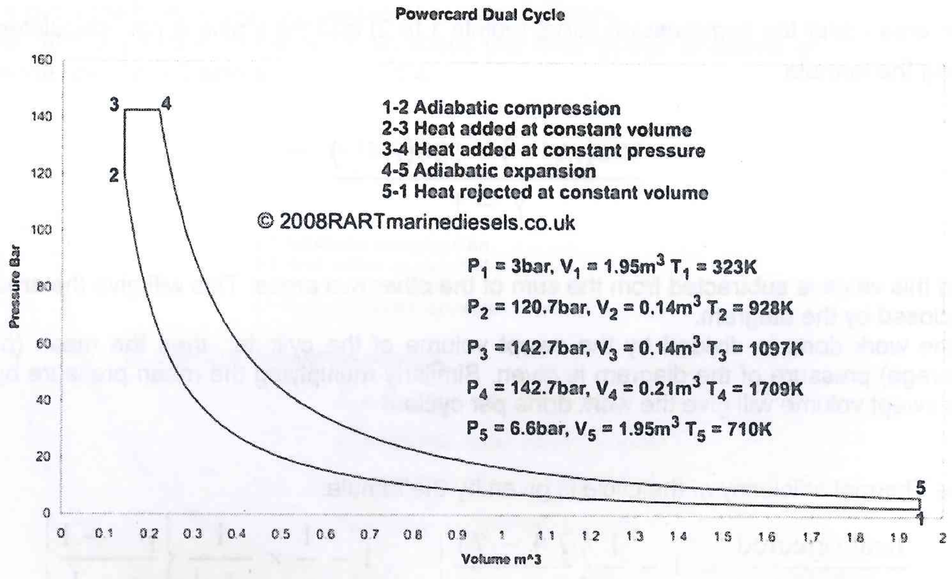
ρ = ratio of the burning period volumes V_3/V_2 (= T_3/T_2)

γ = adiabatic index of compression/expansion (1.4 for air)

Chapter 2: The Dual Combustion Cycle

The dual combustion cycle, like the ideal diesel cycle is a mathematical model, but comes closer to what actually happens in the cylinder of a diesel engine.

Like an indicator diagram it is a graph of Pressure against Volume, but it makes certain assumptions:
The mass of air within the cylinder remains the same: There are no inlet or exhaust valves.
Fuel is not injected and burnt, but heat energy is added at constant volume at the end of compression and then at constant pressure.
At the end of the cycle the heat energy is rejected at constant volume.
The compression and expansion of the air is adiabatic (no heat energy is added or lost) and follows the law $PV^{\gamma} = C$ where γ for air is 1.4.



Enter Engine Speed in rpm	100
Enter Piston stroke in m	2.5
Enter Bore in m	0.96
Enter con Rod Length in m	2.88
Enter Compression Ratio	14
Enter Scavenge Pressure in Bar	2
Enter Air Temp in °C	50
Enter index of compression	1.4
Enter angle of heat addition	20
Enter pressure rise at cv in bar	22

Work under expansion curve	4521.848
Work under Constant Pressure line	1108.212
Work under Compression curve	2738.63
Work Done/rev (kJ)	2891.43
Power Developed (kW)	4819.05
Mean Pressure (Bar)	15.97866
Thermal efficiency %	62.25838

The graph above is of a dual combustion cycle for a cylinder of bore 960mm and stroke of 2.5m. Note that the *x* axis is in m^3, not crank angle. However the axis could have also been the stroke of the piston in m, and the diagram would have had the same shape (because the cross sectional area of the cylinder is a constant).

Because the diagram is a mathematical model, the temperature, pressure and volume at any point in the cycle can be calculated.

The area of the diagram using the units of the graph is equal to the work done in the cycle. To calculate this value:

The area between the horizontal line between points 3 and 4 and the *x* axis represents the work done during the heat addition at constant pressure. This is calculated using the formula:

$$P_3(V_4 - V_3)100 \text{ (kJ)}.$$

This area is added to the area between the expansion curve (points 4 to 5) and the *x* axis which is calculated using the formula:

$$\frac{100(P_4V_4) - 100(P_5V_5)}{\gamma - 1}$$

The area under the compression curve (points 1 to 2) and the *x* axis is now calculated using the formula:

$$\frac{100(P_2V_2) - 100(P_1V_1)}{\gamma - 1}$$

and this value is subtracted from the sum of the other two areas. This will give the area enclosed by the diagram.

If the work done is divided by the swept volume of the cylinder, then the mean (or average) pressure of the diagram is given. Similarly multiplying the mean pressure by the swept volume will give the work done per cycle.

The Thermal efficiency of the cycle is given by the formula:

$$1 - \frac{\text{heat rejected}}{\text{heat supplied}}$$

$$= 1 - \frac{(T_5 - T_1)}{(T_3 - T_2) + \gamma(T_4 - T_3)} \quad \text{or} \quad 1 - \frac{1}{r^{\gamma - 1}}\left\{\frac{\alpha\rho^{\gamma} - 1}{(\alpha - 1) + \gamma\alpha(\rho - 1)}\right\}$$

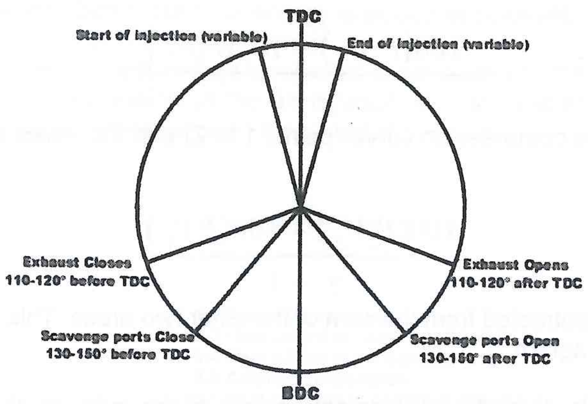
where:
r = compression ratio V_1/V_2
ρ = ratio of the burning period volumes V_4/V_3 (= T_4/T_3)
γ = adiabatic index of compression/expansion (1.4 for air)
α = ratio of pressure increase at constant volume (P_3/P_2)

Chapter 3: The Actual Diesel Cycle

The ideal cycles are theoretical only, and do not show what actually happens inside the cylinder of a large marine diesel engine.

For simplicity and because of its close relationship with the ideal cycles, the following describes a two stroke cycle:

TIMING DIAGRAM 1. ENGINE WITH EXHAUST PORTS OR WHERE THE EXHAUST VALVE CAM IS SYMMETRICAL ABOUT BDC



Starting at bottom dead centre, the exhaust valve is open and the scavenge ports are uncovered. Air is flowing into the cylinder, pushing out or "scavenging" the remaining exhaust gas.

As the piston moves up the cylinder, the scavenge ports are closed off by the piston. The angle at which this occurs varies but is about 140° BTDC. (Before Top Dead Centre).

The exhaust valve then closes. Again this angle is variable but is about 120° BTDC.

Compression now starts. However it is not adiabatic, where no heat is gained and lost to the liner, piston and cylinder head but polytropic (literally, many temperatures). The index of compression, '*n*' is variable but for this example is taken as 1.35.

Although the air inlet to the engine is maintained at about 45 -50°, The air inside the cylinder will have gained heat from the surroundings when the engine is at operating temperature, and at the start of compression is about 75°.

At about 6° BTDC, fuel injection will commence, and will continue through until about 10° ATDC (After Top Dead Centre), depending on engine load..

The fuel will heat up and start to burn just as the piston comes over TDC (Top Dead Centre). Pressure will rise in the cylinder quickly, reaching a maximum at between 12 and 15° ATDC. The piston is driven down by the rapidly expanding gases, converting the energy in the burning fuel into work.

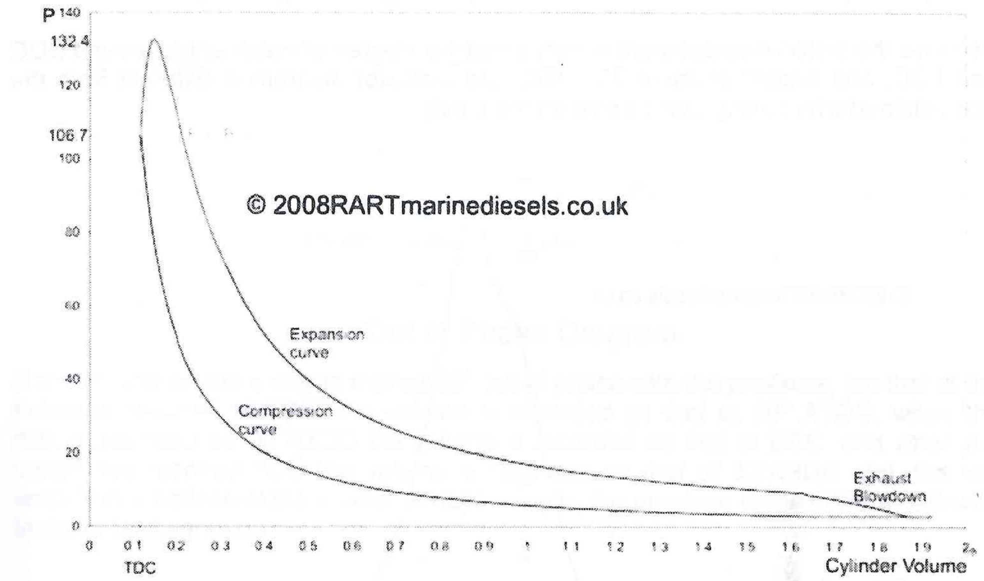
The length of time the fuel burns for again is variable. For this model it is until 33° ATDC when expansion again follows a polytropic curve with the index '*n*' here set at 1.23.

The pressure then drops in the cylinder as the piston moves down the cylinder until the exhaust valve opens (120°ATDC), when the pressure is about 9 bar.

During exhaust blowdown, pressure drops rapidly in the cylinder until the scavenge ports open (140° ATDC) and the pressure drops to scavenge pressure.

Note pressures quoted are 'absolute: i.e. gauge pressure + atmospheric pressure (1bar).

It is possible to model the cycle on a spreadsheet, and to calculate the Mean Indicated Pressure (MIP), Indicated Work and Indicated Power (IP).



Enter Engine Speed in rpm	100
Enter Piston stroke in m	2.5
Enter Bore in m	0 96
Enter con Rod Length in m	2.88
Enter Compression Ratio	14
Enter Scavenge Pressure in Bar	2
Enter Scavenge Temp in °C	50
Enter index of expansion 'n'	1.23
Constant for expansion	17
Enter index of compression 'n'	1.35
Constant of Compression	5 83353218

Work under compression curve (kJ)	2146.896
Work under scav line comp. stroke (kJ)	89.857
Work under fuel burn curve (kJ)	2021.921
Work under expansion curve (kJ)	2999.527
Work under exhaust blowdown (kJ)	151 272
Work under scav line exp. Stroke (kJ)	18.651
work done/rev (kJ)	2954.620
Mean indicated pressure (Bar)	18.328
Indicated power kW	4924.366

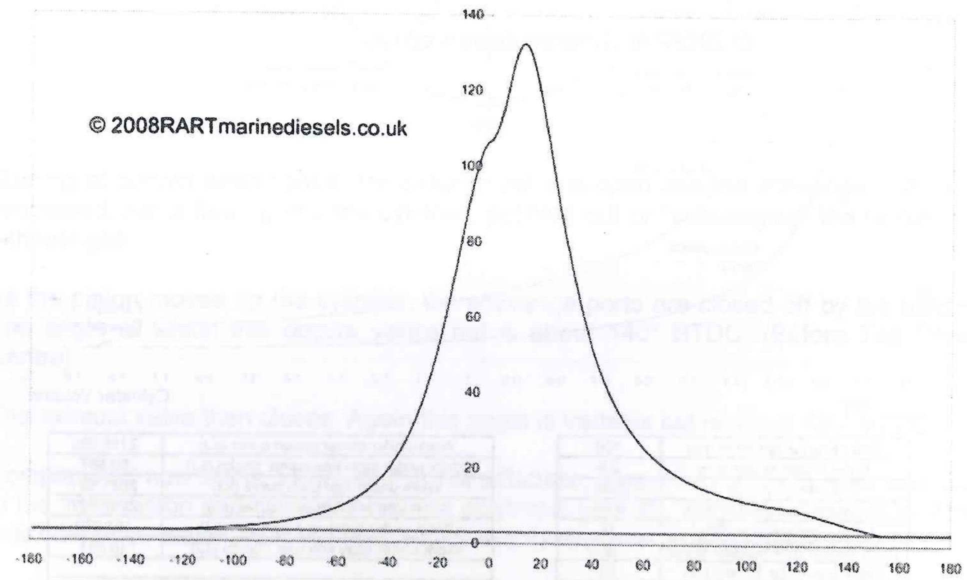
Actual Diesel Engine Cycle Indicator Diagram (Power card)

Because the diagram is a pV curve, the area enclosed by the diagram is indicative of the work done during the cycle. To calculate the area of the diagram, I subtracted the area under the compression line (i.e BDC - TDC) from the area under the expansion line (i.e TDC - BDC). The area under the polytropic compression and expansion curve are easily calculated because they follow the law $pV^n = C$. Although there is no set equation for the fuel burning curve or the exhaust blowdown curve, the Exel spreadsheet calculated equations which gave a good representation. By integrating the equation between the required values it was possible to calculate the area (and therefore the work done) under those curves. The results can be seen in the table on the diagram.

If the work done per cycle (revolution) is divided by the swept volume then the Mean Indicated Pressure is obtained. In this case it is just over 16.3 bar. When calculating the Indicated Power for a cylinder on an actual engine, it is the Mean Indicated Pressure that needs to be obtained from the indicator diagram. If the Mean Indicated Pressure is multiplied by the swept volume of the cylinder (a known quantity), then the work done/cycle is obtained.

As stated earlier, the air in the cylinder is about 75°C at start of compression. The temperature at the start of fuel injection can be calculated using PV/T = Constant. This gives a temperature of 593° at 6° BTDC and 611° at TDC. The temperature rises rapidly during combustion to about 1500°C depending on the type of fuel.

Because the piston displacement is very small per degree of crank angle around BDC and TDC, and largest at about 70°ATDC, the indicator diagram is different from the one obtained when using crank angle on the *x* axis.

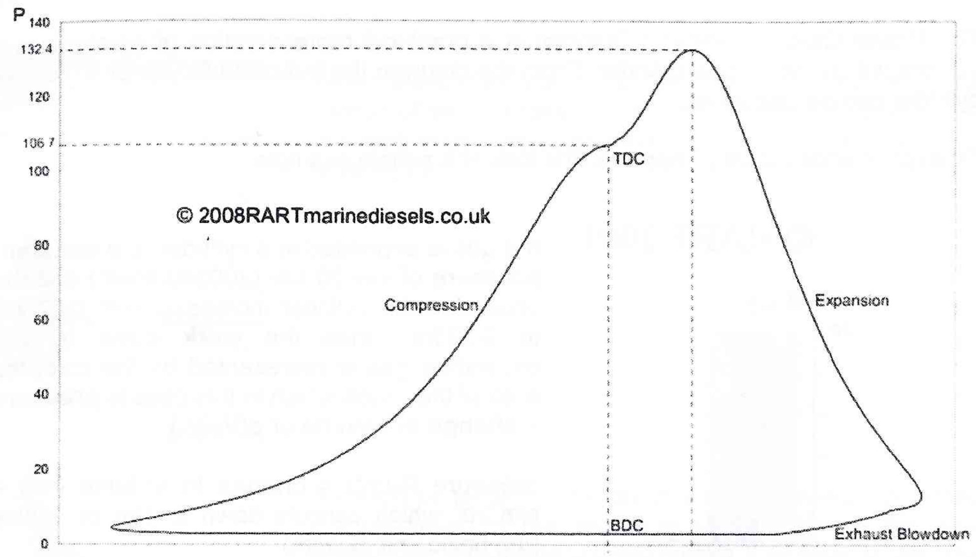


Crankangle/Pressure Diagram (2)

The crank angle/pressure diagram is often used by computer packages which calculate the MIP and IP by measuring the pressure in the engine cylinders using a pressure transducer, whilst at the same time measuring the crank angle using a pick up on the flywheel. However these diagrams tend to measure the crank angle from 0 - 360° and so look like the diagram above.

Because the computer program will have been programmed with the engine bore, stroke and conrod length, it can calculate the MIP and IP from the pressure recorded for each crank angle, which it measures every 1°.

Another diagram which can be drawn is the out of phase diagram. In this the crank angle is offset by 90°



The volume shown on the X axis is now 90° out of phase

Out of Phase Diagram

If the volume on the *x* axis is moved 90° out of phase with the pressure, (so that at the pressure reached at TDC, the volume is recorded as that at 90° ATDC, when the piston has reached 90°ATDC the volume is recorded as that at BDC, and when the piston has reached BDC the volume is recorded as that of 90°ABDC etc), this will amplify in a horizontal plane what is happening to the pressure in the cylinder allowing faults to be diagnosed.

Chapter 4: How to Take a Powercard

The Power Card or Indicator Diagram is a graphical representation of pressure and volume within the engine cylinder. From the diagram the Indicated Power or IP for the cylinder can be calculated.

To explain what actually happens, first look at a simple example:

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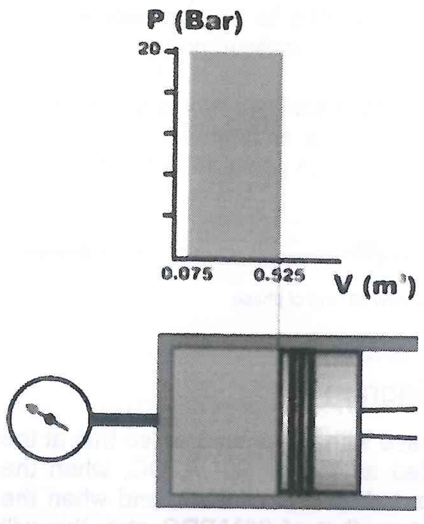


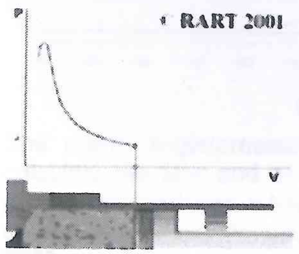
If a gas is expanded in a cylinder at a **constant pressure** of say 20 bar (2000000N/m^2) and the volume in the cylinder increases from 0.075m^3 to 0.525m^3, then the **work done** by the expanding gas is represented by the coloured area of the graph which in this case is **pressure × change in volume** or $p(V_2-V_1)$

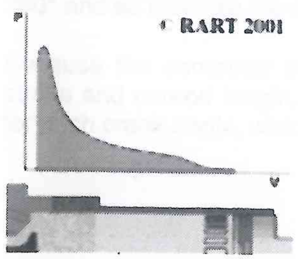
pressure (N/m^2) × **change in volume** (m^3) = Nm^3/m^2 which cancels down to Nm or Joules (the basic unit of work).

inserting values:

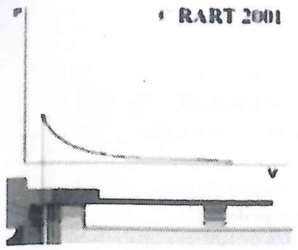
$20 \times 10^5(0.525 - 0.075) = 9 \times 10^5$ J or 900kJ

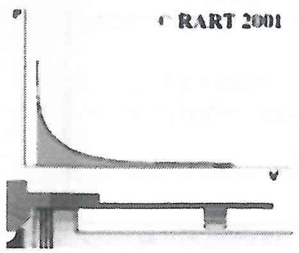
Similarly, if the piston was pushed up the cylinder and the pressure remained constant at 20 bar then the work **input** would be 900kJ (if the formula used is $p(V_2-V_1)$ then the answer would be $20 \times 10^5(0.075 - 0.525) = -9 \times 10^5$ J or -900kJ:- (the negative figure indicates work input)



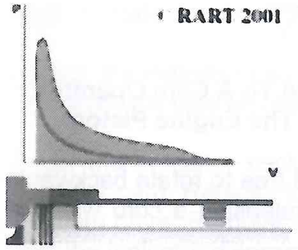


When a piston is moving downwards on the power stroke in a diesel engine, the pressure does not remain constant. However this does not change the principle that the work done (work output) is represented by the area under the expansion curve as shown on the diagram.



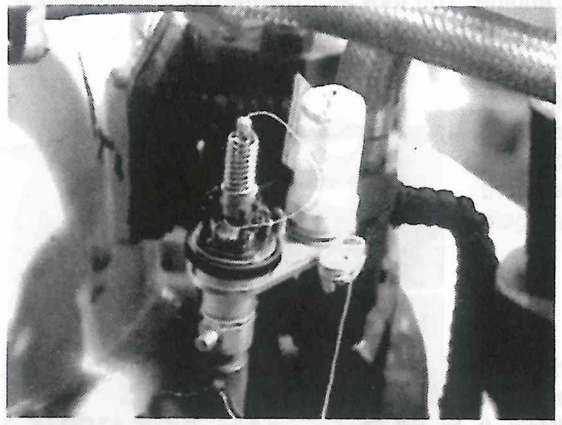


When the piston moves up the cylinder compressing the charge of air, this takes work. Again this is represented by the area under the compression curve, but this time it is a work input.



It should be evident therefore that the actual work obtained per revolution of the engine is represented by the area under the expansion curve minus the area under the compression curve.

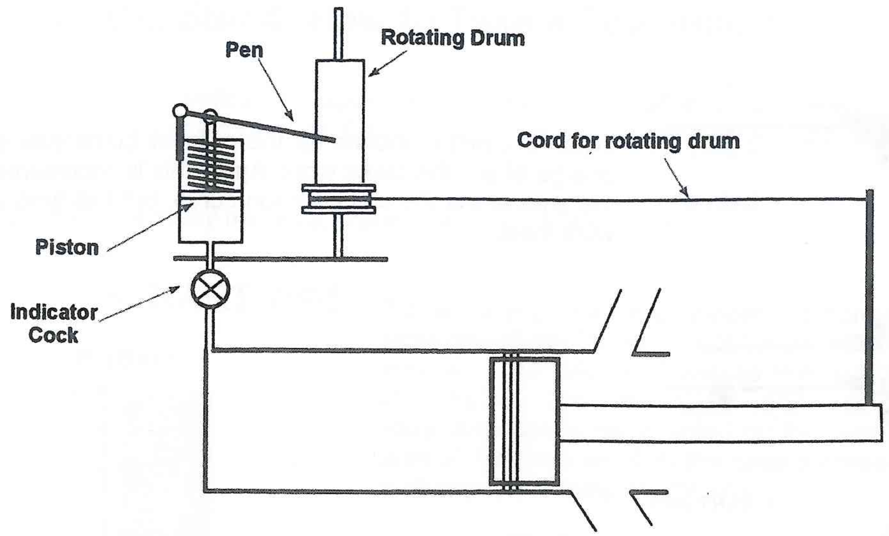
So that the work done per cycle (and thus the indicated power) can be calculated, a Pressure - Volume diagram must be taken from the engine. This diagram is known as a Power Card or Indicator Diagram.



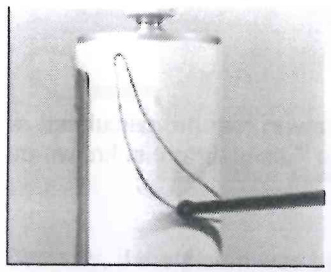
To measure the pressure in the cylinder at any time during the cycle, use is made of a tapping in the cylinder head, terminating in a valve known as an indicator cock. The indicator cock is fitted with a male screw thread on its outlet, and using this thread the piece of equipment shown, known as the engine indicator is mounted.

When the indicator cock is opened, the gas pressure in the cylinder acts on the under side of a piston moving it upwards against a spring.

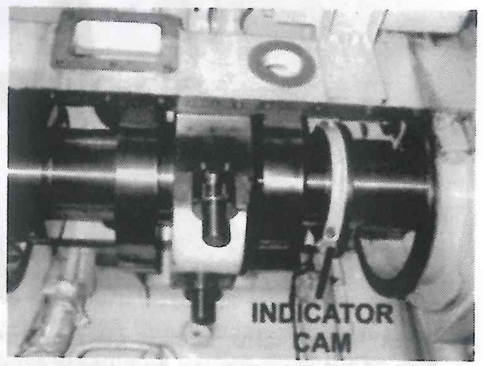
As the piston moves upwards, so a marker, connected through a parallel link motion will move up a drum, tracing a vertical line on a piece of paper wound around the drum. The height of the line is proportional to the maximum pressure in the cylinder. The manufacturer of the equipment gives the relationship between the pressure rise and the vertical distance the marker will move. This is known as the **spring constant.**



Indicator Card Principle: In Reality The Cord Is Attached To A Cam Operated Linkage Which Moves Up And Down In Synch With The Engine Piston



The drum is spring loaded and free to rotate backwards and forwards by pulling and releasing a cord which is wrapped round the drum.
The end of this cord is fixed to a cam operated linkage so that the drum rotates forwards and backwards as the piston moves up and down the cylinder.
This means that as the marker is moving up and down, the drum is rotating backwards and forwards in time with the engine. The indicator diagram or power card is traced out on the paper as shown.



Indicator cards should be taken under steady conditions when the weather is calm. The load on the engine should be between 85 and 100%. Liaise with the OOW that there is no intended change of course which could vary the load on the engine.

Run the engine at full load for 30 minutes before taking the cards, especially if it has been running at low load in the previous 24 hours. This is to burn off any carbon in the cylinder. Blow through the indicator cocks to prove they are clear and to remove any particles of carbon.

Ensure that the indicator equipment is in good condition and that the piston is free to move in the cylinder. Ensure that the correct spring is fitted.

Fit the pressure sensitive paper to the drum.

Before fitting the indicator equipment to each cylinder blow the indicator cock through.

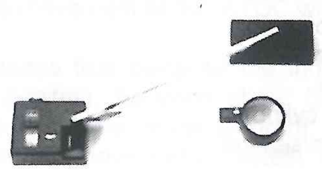
Fit the equipment to the indicator cock. Press the pen against the drum and pull the cord to rotate the drum by hand to draw the atmospheric line.

Unlock the indicator cam drive and drop it onto the cam. Attach the cord to the drive. The drum will now be rotating back and forward in synch with the piston position.

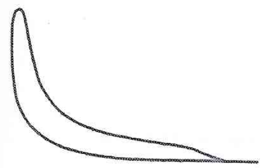
Open the indicator cock fully. Press the pen against the paper and trace out the indicator card. Release the pen and shut the indicator cock. Disconnect the cord from the cam drive. Remove the equipment from the indicator cock and remove the paper with the indicator diagram from the drum.

Record exhaust temperature, fuel rack setting, scavenge pressure and temperature.

Repeat for all the engine cylinders.



The next thing to do is measure the area of the diagram. This could be done mathematically using the mid-ordinate rule, or by using a piece of kit known as a planimeter (shown opposite). The outline of the powercard is traced using the planimeter and the area read off the scale. The length of the diagram is also measured.



If the total area of the diagram is divided by the length, the average or **mean height** of the diagram will be obtained. Looking at it another way, we now have an indicator diagram of a rectangular form with the same area and length as the original power card.

Because the **mean height** of the diagram can be converted into a pressure by multiplying by the **spring constant**, a pressure known as the **mean indicated pressure** or **MIP** can be obtained.

The length of the diagram is representative of the **swept volume** of the cylinder. This volume can be obtained by multiplying the cross sectional area of the cylinder by the stroke.

If the **MIP** is multiplied by the **swept volume**, the work done per cycle will be obtained. E.G.

Area of diagram = $840mm^2$
Length of diagram = 105mm
Mean height of diagram = 8mm
Spring constant = $200kN/m^2$ per mm.
MIP = $8 \times 200 = 1600N/m^2$
Diameter of cylinder = 960mm (radius 0.48m)
Stroke of piston = 2.5m
Work per cycle (or revolution) = $1600 \times \pi \times 0.48^2 \times 2.5 = 2895$ kNm or kJ

If this figure is now multiplied by the number of power strokes/sec, the power output of the cylinder will be obtained. (work done per second)

E.G. 90RPM = 1.5revs/sec.
$2895 \times 1.5 = 4343kW$

This process is repeated for each cylinder on the engine. The total indicated power for the engine can therefore be calculated, and individual power outputs for respective cylinders compared to each other.

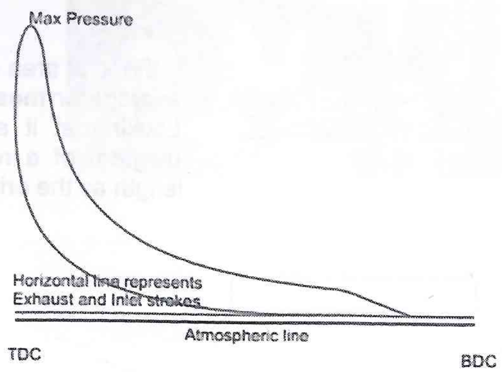
The formula to remember is

IP = MIPLAn
where IP = indicated power
MIP = mean indicated pressure
L = length of stroke
A = cross sectional area of cylinder
n = number of power strokes/second

NOTE: The number of power strokes/second is the same as the revs/second for a 2 stroke engine, and revs/second ÷ 2 for a 4 stroke engine.

The powercard for a 4 stroke engine is exactly the same as for a 2 stroke engine except that on the exhaust and inlet strokes, a horizontal line is traced.

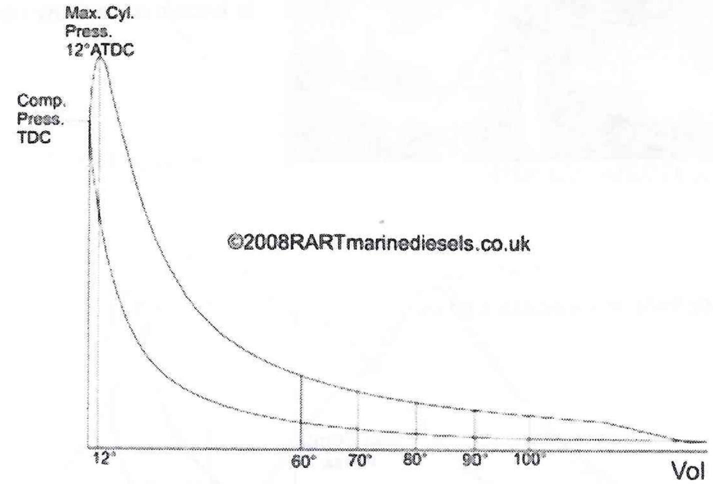
Although the pressure in the cylinder is different for the inlet and exhaust strokes, the difference is too small to be shown on the scale of the diagram.



POWER CARD - 4 STROKE ENGINE

Chapter 5: Out of Phase or Draw Cards

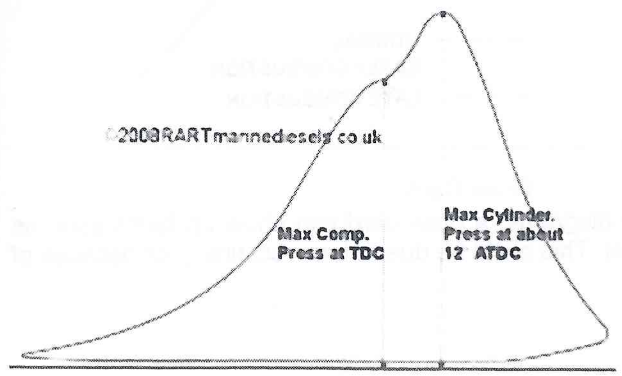
The conventional power card as used for measuring indicated power in the cylinder has its limitations. Because it is a pressure/volume diagram, the piston movement and therefore volume change is very small either side of Bottom Dead Centre and Top Dead Centre (i.e. each end of the diagram), and so what is happening at the end of compression and start of combustion is squeezed into a very small area.



Power Card, showing that relationship between volume and crank angle is not linear

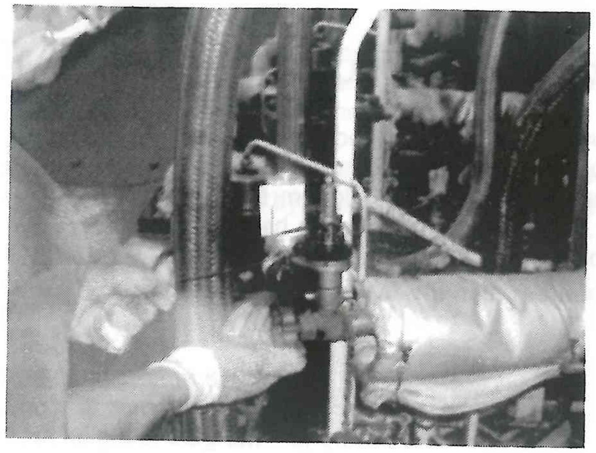
In the example shown, which is a powercard for an engine of 2.5m stroke with a conrod length of 2.88m, the piston displacement per degree of crank angle is greatest at about 70° ATDC where the displacement is almost 24mm/degree. Compare this with the movement at 10° ATDC where the piston displacement is only 5mm/degree.

This means that because the indicator drum movement is in direct relation to the piston position, between about 50 and 90° of crank angle represents the largest change in volume on the diagram, and it is here that the drum on the engine indicator equipment is moving at its fastest. Note also that the mid point on the diagram is not at 90° because on an engine of this stroke and conrod length, the mid stroke is reached at about 77°ATDC

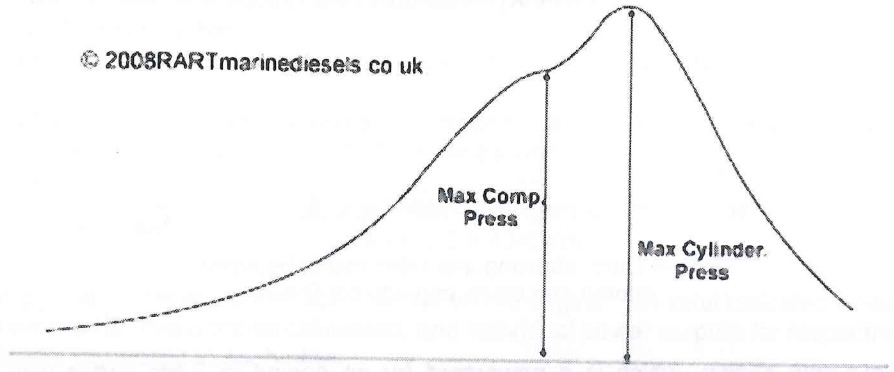


Out of Phase Card

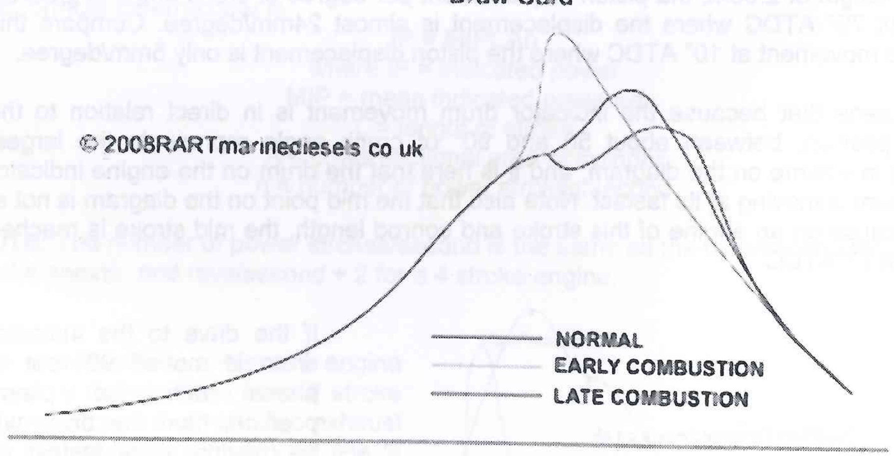
If the drive to the indicator drum is moved 90° out of phase with the piston position, then the drum will be moving at its fastest as the piston comes over TDC and BDC. This means that the pressure rise in the cylinder about TDC is shown over a wider horizontal axis and because of this, faults with timing and injection can be identified.



Instead of advancing the cam by 90°, conditions in the cylinder around TDC can be assessed by rotating the drum by pulling on the cord at the critical moment as the piston comes over TDC. This is known as a draw card.



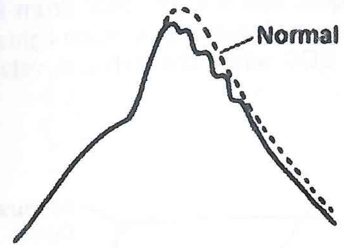
Draw Card



Draw Card

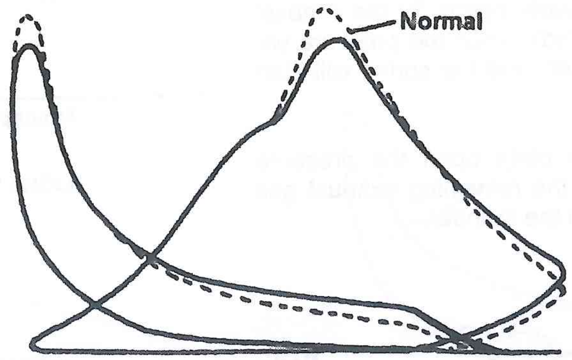
As can be seen from the above diagrams, a draw card can show up faults such as early or late combustion of the fuel. This could be due to incorrect timing or because of varying ignition delays.

Other faults which can be seen are a leaking or blocked fuel injector and afterburning.



LEAKING INJECTOR

CHOKED INJECTOR



AFTERBURNING

Not all engines are fitted with the equipment to take power cards. However, a draw card can be taken on any engine fitted with an indicator cock. A competent engineer, with practice, can take a draw card on an engine running at up to 450rpm.

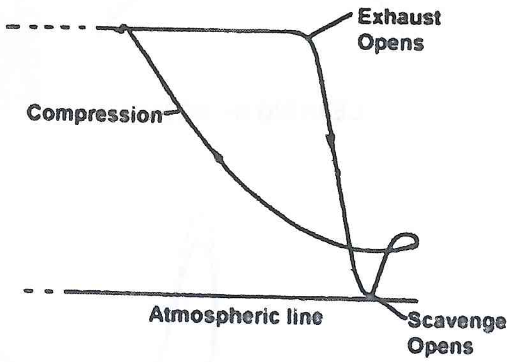
Chapter 6: The Light Spring Diagram

The light spring diagram is taken on a two stroke engine with the indicator drum in phase with the engine piston, but with the indicator spring replaced with a much lighter spring. It enables conditions to be examined around BDC when the exhaust valve opens and scavenging of the cylinder is taking place.

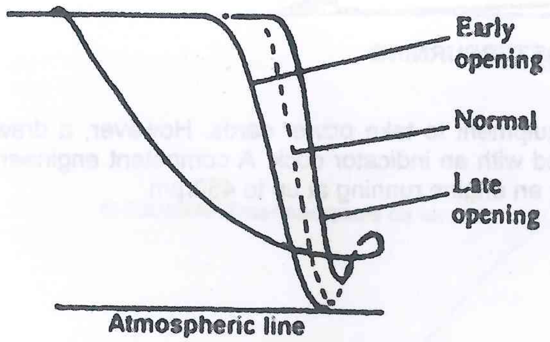
This means that as the piston comes up on compression, the spring will be fully compressed against the limiting stop at about 10 bar. The compression line will then stay horizontal as compression pressure increases in the cylinder.

During expansion the line remains horizontal until the exhaust valve opens (or the exhaust ports are uncovered), when the pressure will drop in the cylinder, and the spring will start to expand.

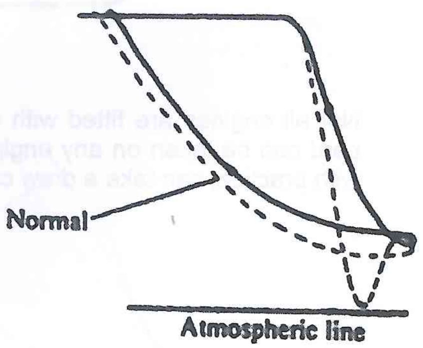
As the scavenge ports open the pressure rises slightly and the remaining exhaust gas is scavenged from the cylinder.



LIGHT SPRING DIAGRAM



EARLY OR LATE EXHAUST OPENING



CHOKED EXHAUST

Chapter 7: Engine Indicator Faults

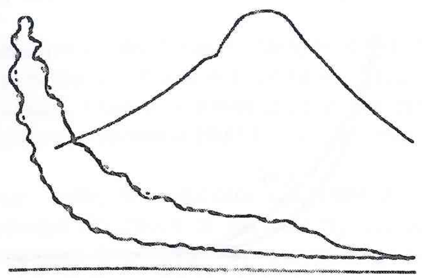


Fig. 1. Vibrations in drive.
Draw-diagram not affected.

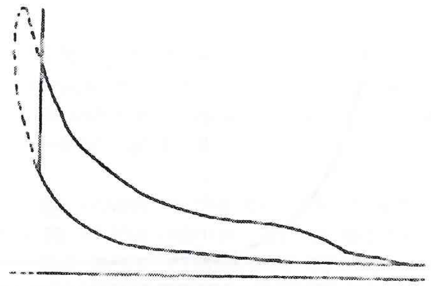


Fig. 2. Length of cord too long.
T.D.C.-part missing

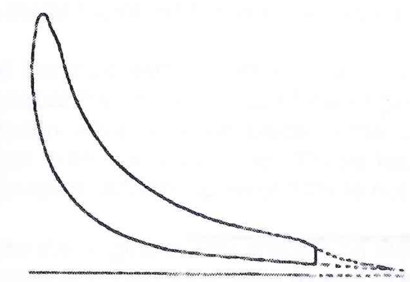


Fig. 3. Length of cord too Short.
B.D.C.-part missing

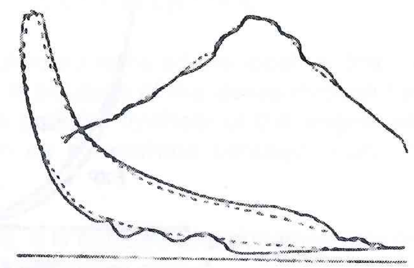


Fig. 4. Friction in indicator piston.
Draw-diagram also affected. This fault gives a too large working diagram area

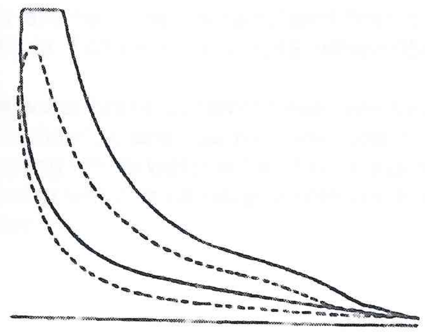


Fig. 5. Spring too weak. Indicator piston strikes top end of cylinder.

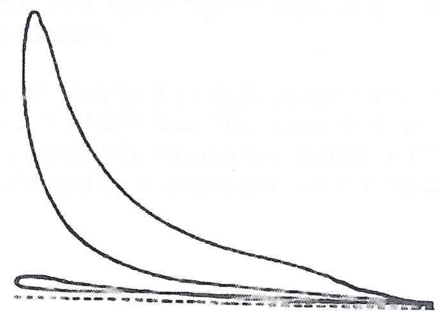
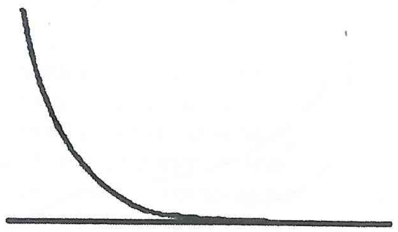
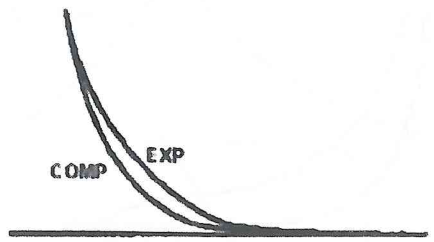


Fig. 6. Indicator cock leaking.
Atmospheric line untrue.

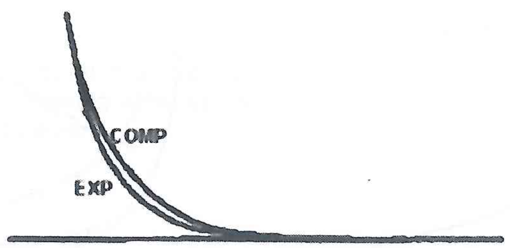
If there is doubt over the timing of the indicator drive cam with the piston position, then take an in phase indicator card with the fuel off the cylinder. The compression and expansion curves should overlap. If the diagram is positive in area then retard the indicator cam; If the diagram is negative in area, advance the indicator cam.



NORMAL: COMPRESSION AND EXPANSION CURVES OVERLAP



INDICATOR CAM TOO ADVANCED



INDICATOR CAM RETARDED

Chapter 8: MIP, MEP, Balancing Engine

Engine manufacturers quote Mean Effective Pressure when giving data about their engines. Although Mean Effective Pressure and Mean Indicated Pressure are similar they are not the same.

Sometimes reference is made to IMEP (Indicated Mean Effective Pressure) and BMEP (Brake Mean Effective Pressure). This really confuses the issue, but if it has the word Indicated, then it is measured in the cylinder, so is the same as MIP (Mean Indicated Pressure). Similarly BMEP is measured at the Brake or Flywheel.

When using an indicator diagram to calculate the power in the cylinder, the Mean Indicated Pressure is derived by dividing the area of the diagram by its length and multiplying the result by the spring constant. This is then multiplied by the swept volume of the cylinder (which is a constant for that particular engine) and the power strokes per second. So if indicator diagrams were taken for all the engine cylinders, because the speed and the swept volume are the same, to compare the indicated power for each cylinder, only the Mean Indicated Pressures need to be compared. To have the engine perfectly balanced, (i.e. Indicated Power the same for all cylinders), the Mean Indicated Pressure would have to be equal for all cylinders

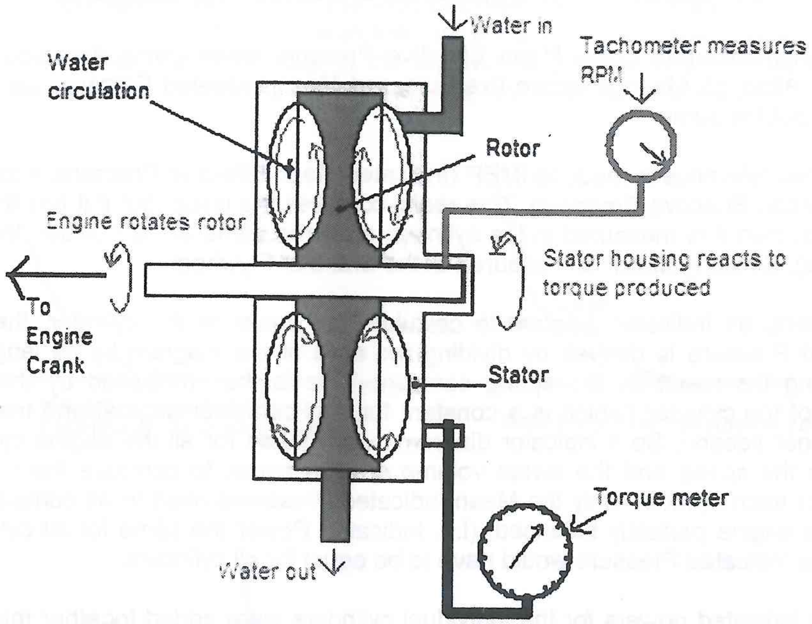
If all the indicated powers for the individual cylinders were added together this would not equal the power output of the engine. This is because of the losses through friction between what is taking place in the cylinders and the flywheel of the engine where power output is measured. These losses can be somewhere between 8 and 15%, although a ballpark figure of 10% is not far out.

When the engine is first erected on the testbed, it is coupled to a dynamometer (more commonly known as a water brake) to measure the power output.

The words Brake power comes from the type of dynamometer: often called a water brake because when put into operation it is trying to slow the engine down.

The engine power is calculated from the torque and speed figures according to the formula: Torque × rpm / 9549, where 9549 is a constant.

The water brake dynamometers are very popular, due to their high power capability, controllability, and relatively low cost compared to other types. They consist of a fluid coupling where water is the drive transmission between the engine driven rotor and the housing which is capable of rotation, but is restrained by a torque arm connected to a meter.



The schematic shows the most common type of water brake, the variable level type. Water is added until the engine is held at a steady rpm against the load. Water is then kept at that level and replaced by constant draining and refilling, which is needed to carry away the heat created by absorbing the energy (which in itself is a measure of power output of the engine). The housing attempts to rotate in response to the torque produced but is restrained by the scale or torque metering cell which measures the torque.

When the power for the full load operation has been measured, it is assumed that the power developed in each cylinder is equal.

Therefore the power developed by each cylinder is total power divided by the number of cylinders.

If this figure is now divided by the swept volume × power strokes/second, the Mean Effective Pressure (MEP) should be obtained.

For example a 10 cylinder two stroke engine of 960mm bore, 2.5 metre stroke has a swept volume per cylinder of:

$\pi \times 0.48^2 \times 2.5 = 1.81 m^3$

If the measured power at the flywheel is 45000kW when the engine is running at 100rpm (1.67rps), then it can be assumed that each cylinder is delivering 4,500kW.

Therefore MEP = 4500/(1.81 × 1.67) = 1492 kN/m^2 = 14.92 bar.

If the MIP was measured at 16.6 bar, this would show a 10% loss through the engine.

MEP/MIP = Mechanical Efficiency

Indicator cards should be taken at regular intervals as part of the planned maintenance for the engine. They should also be taken after any major overhaul of the engine or changes to the fuel injection equipment or adjustment of the camshaft to allow for elongation of the timing chain.

An evenly balanced engine is desirable because it ensures that the engine is operating at its maximum efficiency and will prevent overloading of individual cylinders which can lead to excessive bearing loading and crankshaft vibrations and stresses.

Cylinder power output is directly proportional to the Mean Indicated Pressure which is again proportional to the amount of fuel burnt in the cylinder, but before adjusting the fuel pumps, the draw cards (or crank angle diagram) should be examined to ensure that compression pressures and peak pressures are the same, and that there are no faults with the fuel injection. If compression pressure is down on an individual cylinder, it may point to poor ring sealing or leaking valves. If the peak pressure is too low or too high, then fuel timing should be checked.

Also to be taken into account are the exhaust temperatures and fuel pump rack setting.

Adjustments should be small, and noted down. After each adjustment, wait for the engine to settle down before retaking the cards.

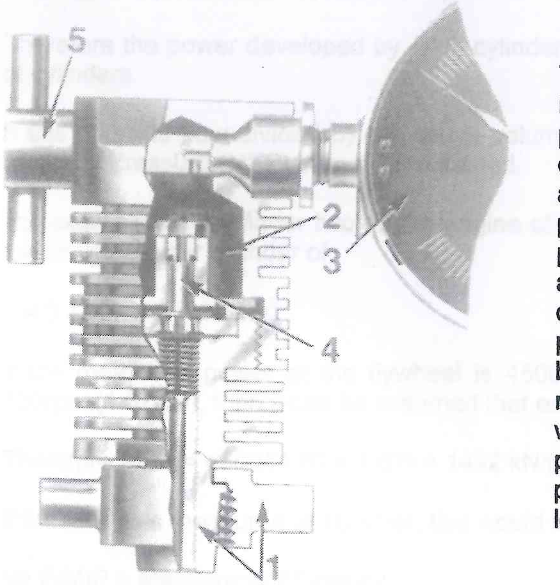
Chapter 9: Balancing an Engine Using Peak Pressures

Indicator power cards cannot be taken using mechanical equipment on a medium speed engine because the engine is running at too fast and vibrations set up in the drive will give an erratic irregular diagram. If a computer system is not fitted, then a degree of engine balancing can be achieved by looking at the compression pressures and the maximum cylinder pressure.

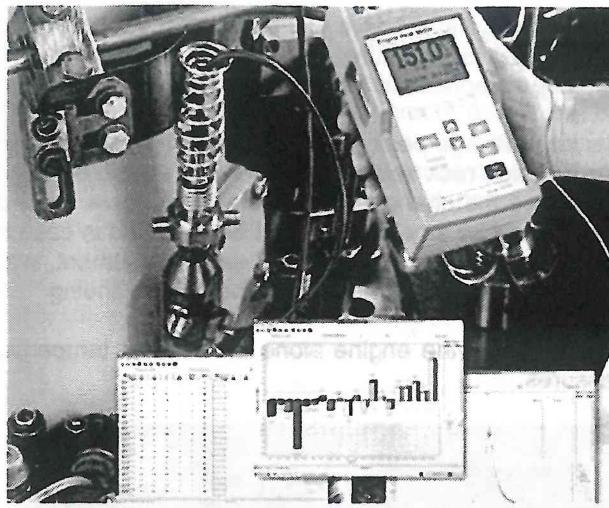
It should be understood that power developed in the cylinder is directly related to the Mean Indicated Pressure (MIP), and not the peak pressure, which is related not only to the quantity of fuel burnt but the injection timing. However if the timing of the fuel injection is correct, and combustion conditions correct within the cylinder, then balancing of the engine can be successfully carried out.

It may be possible to take a set of draw cards to ensure that injection timing and cylinder combustion is correct. However if this is not possible, then an assumption has to be made that conditions are correct and that injection equipment is in good condition. Obviously visible signs of poor or incorrect combustion such as smoke at the funnel or excessively variable exhaust temperatures should be investigated before a set of readings are taken. The turbocharger should be clean and operating efficiently, with the inlet air temperature and pressure at the normal parameters.

To measure pressure in the cylinder, a peak pressure indicator is used. This can be a gauge type indicator as shown below or a more sophisticated digital gauge which will measure max peak pressure, minimum peak pressure and average peak pressure over a number of cycles It will also measure the rate of pressure rise, which if too high can damage bearings. It is also possible to use a traditional mechanical indicator to trace a vertical line up the paper, the height of which is proportional to the max cylinder pressure.



The indicator is installed on the indicator valve using the wing nut & plug (1). When the indicator valve is opened, the check valve (4) will rise and fall with the engine cylinder pressure, trapping the maximum pressure in the pressure chamber (2), and this average maximum pressure can be read directly on the dual scale pressure gauge (3). The gauge may be conveniently positioned for ease of reading by loosening the gauge bolt wing nut (5). The pressure in the pressure chamber (2) and on the pressure gauge (3) is removed by loosening the wing nut (5).



Shown above: A Kistler Digital Peak Pressure Meter

Measuring functions:

pmax	maximum peak pressure
pmin	minimum peak pressure
pav	average peak pressure
Sdev	standard deviation of the peak pressure
dp/ca	maximum gradient of the pressure trace
r/min	speed
ppeak	instantaneous peak pressure of continuous measuring function

Before taking the maximum cylinder pressures, the compression pressures should be recorded. This is done by measuring the cylinder pressure with the fuel taken off the unit. To do this, the engine should be run at reduced load to avoid stressing the crankshaft. Poor compression will indicate worn rings or liner or a valve not seating correctly. It is not possible to balance an engine successfully if it has these faults.

Once the compression pressure on all the cylinders has been obtained, the engine load should be increased to at least 85%, although full load is preferable. (A generator, for safety reasons should not be loaded above 85% in case it trips off). The engine load should be steady whilst the engine peak pressures are recorded, so in the case of a main propulsion engine, liaise with the OOW to ensure there will be no change of course for the next 30 minutes or so.

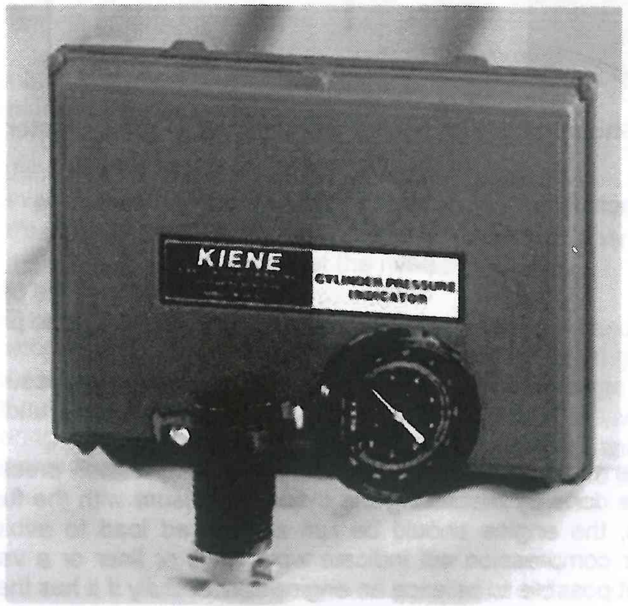
Take the peak pressures. At the same time record the cylinders fuel pump rack setting and the exhaust temperature. Also recorded are the inlet manifold or scavenge air pressure and temperature.

When attempting to balance the engine attention must be paid to the difference between maximum pressure and compression pressure for each cylinder, together with the exhaust temperature and fuel pump setting. It is not advisable just to increase the fuel to cylinders with lower maximum pressures. Some cylinders may need an increase in fuel, others a decrease.

For example: If a particular cylinder has a low peak pressure, but good compression pressure, and a low exhaust temperature, with the fuel rack a couple of points below the average, it is perfectly acceptable to increase the fuel rack on that cylinder.

However if a cylinder has average peak and compression pressures and high exhaust temperatures with an average rack setting, then either it should be decreased a small amount or left alone to see the changes when other units have been adjusted. In all cases any adjustment should be small and it must be recorded in case there is a need to return the engine to its original settings. After each adjustment, wait and see the effects, taking further peak pressures if necessary, before continuing.

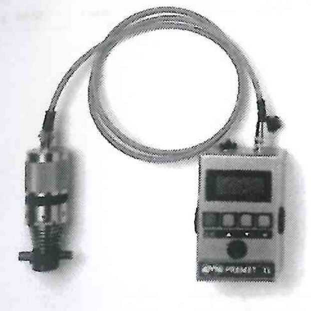
It is not acceptable to balance the engine alone on exhaust temperatures, fuel rack settings or peak pressures.

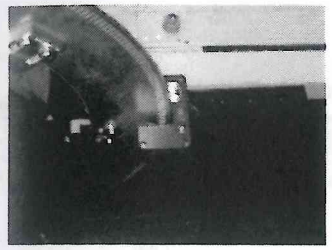


Chapter 10: Computer Generated Diagrams

Indicator diagrams taken using mechanical equipment are limited by the speed of the engine and the skill of the operator. Normally it is not possible to obtain an accurate powercard on engines running above 120rpm because of vibrations in the equipment. Draw cards are reliant on the operators skill. It is possible to take drawcards on engines running up to about 450 rpm.

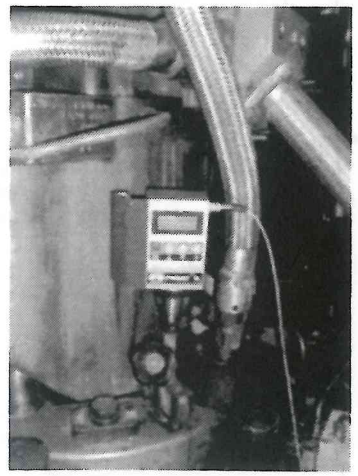
Modern methods allow the computer to generate indicator diagrams and calculate the power developed in the cylinder for engines running at any speed.

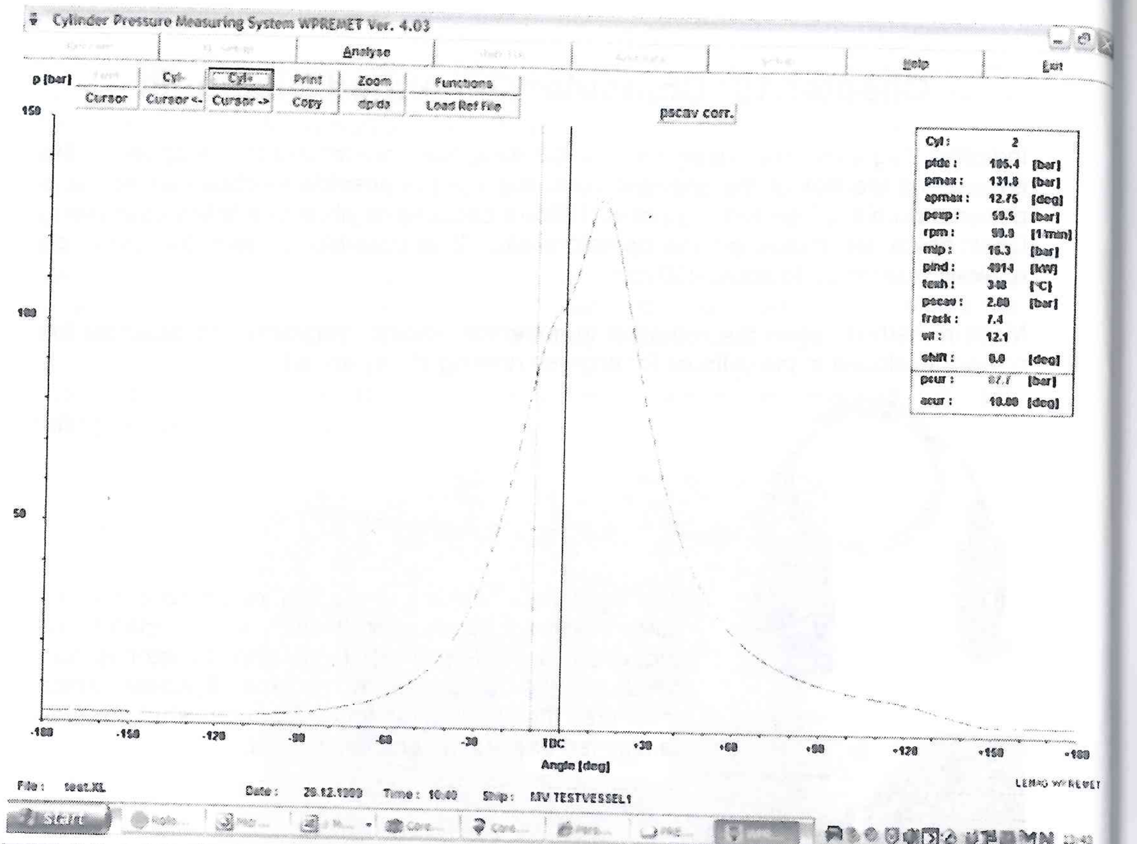


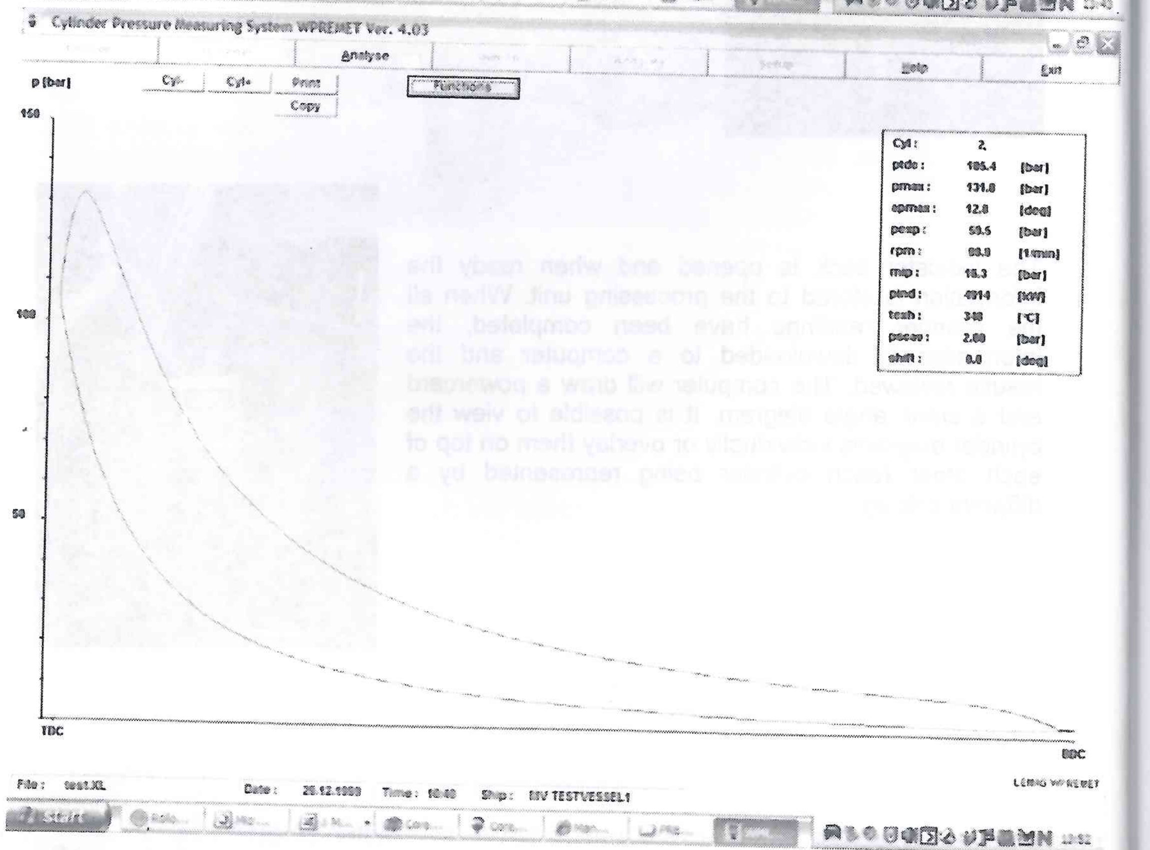


The equipment consists of a pressure transducer which screws onto the indicator cock which is connected to the processing unit. A crankshaft position detector is also fitted to the propshaft or engine flywheel which measures the crank angle to 1°. The crankshaft position is also fed back to the processing unit.

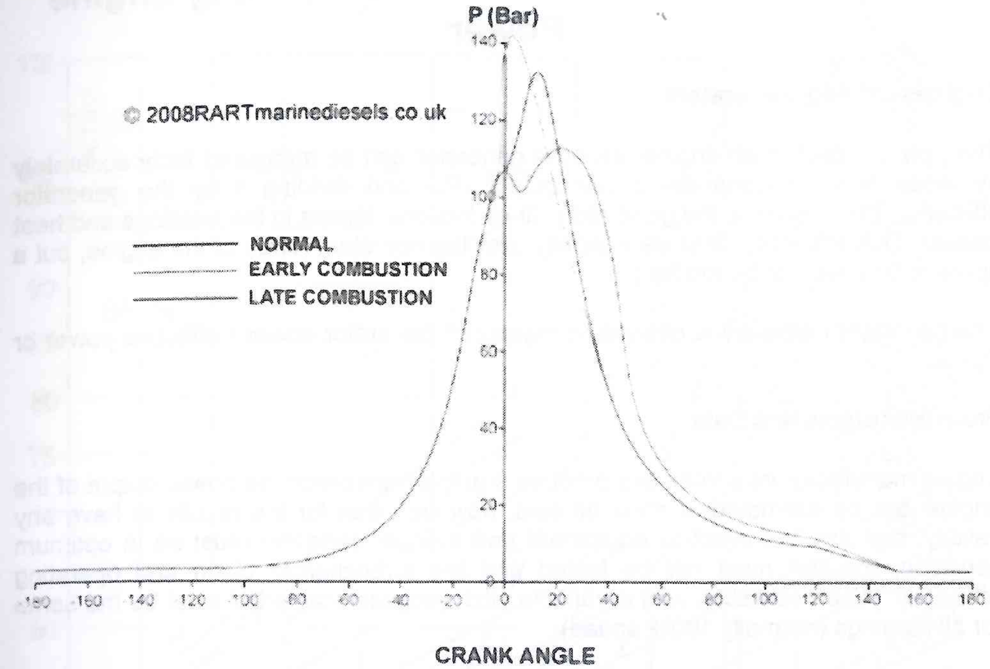
The indicator cock is opened and when ready the information is stored to the processing unit. When all the cylinder readings have been completed, the information is downloaded to a computer and the results reviewed. The computer will draw a powercard and a crank angle diagram. It is possible to view the cylinder diagrams individually or overlay them on top of each other (each cylinder being represented by a different colour).



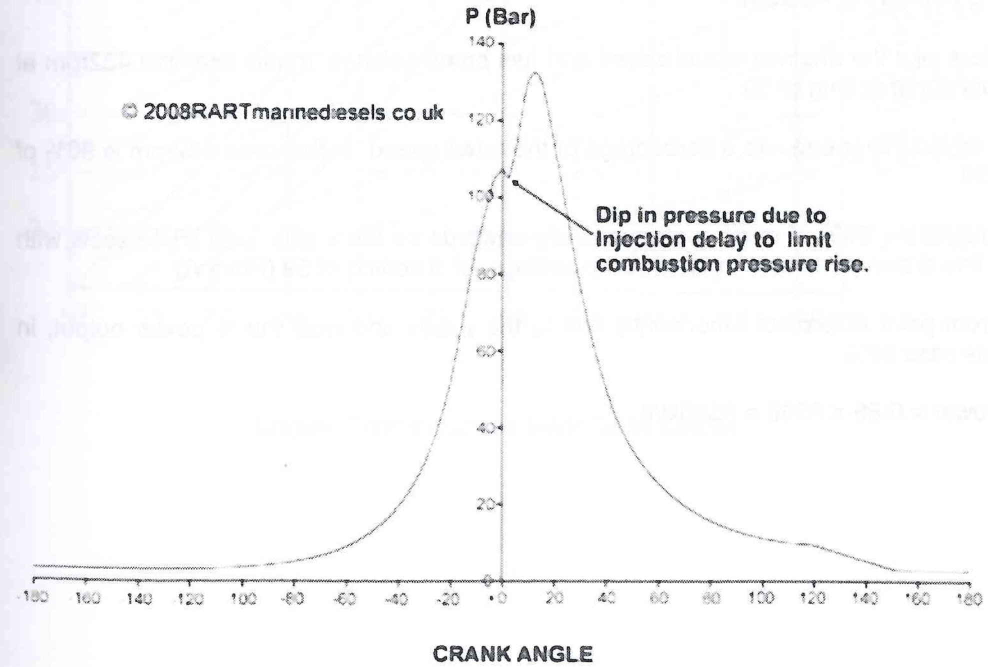




From the crank angle diagram, faults in the injection process can be determined:



It should be noted that some slow speed two strokes (MAN B&W MC engines) combustion is delayed until 2-3° after TDC to limit the pressure rise to 35 bar, to prevent overloading of the crosshead bearing. This will give a slight dip in the pressure after TDC, which should not be construed as a fault.



Chapter 11: Other Methods of Determining Engine Power

Engines Driving Generators

The power output of an engine driving a generator can be measured fairly accurately by measuring the generator power output (P_w) and dividing it by the generator efficiency. (the losses in the generator will be frictional losses in the bearings and heat losses). This efficiency does vary slightly over the operating range of the engine, but a figure of 0.92 will not be too far out.

The generator efficiency is defined as measured generator power / effective power or P_w / P_e

From Manufacturers Data

Engine manufacturers sometimes produce a graph from which the power output of the engine can be estimated. It must be said, however, that for the results to have any validity, that the fuel injection equipment and cylinder condition must be in optimum condition, the hull must not be fouled and the turbocharger clean and operating efficiently. When operating with a variable pitch propeller, the pitch must be the same for all readings (normally 100% ahead)

The type of fuel will have an effect and this must be taken into account.

The graph on the next page is for a MAN B&W medium speed engine.

To use the graph the engines rated speed and power output must be known.
e.g 6200kW at 450rpm.

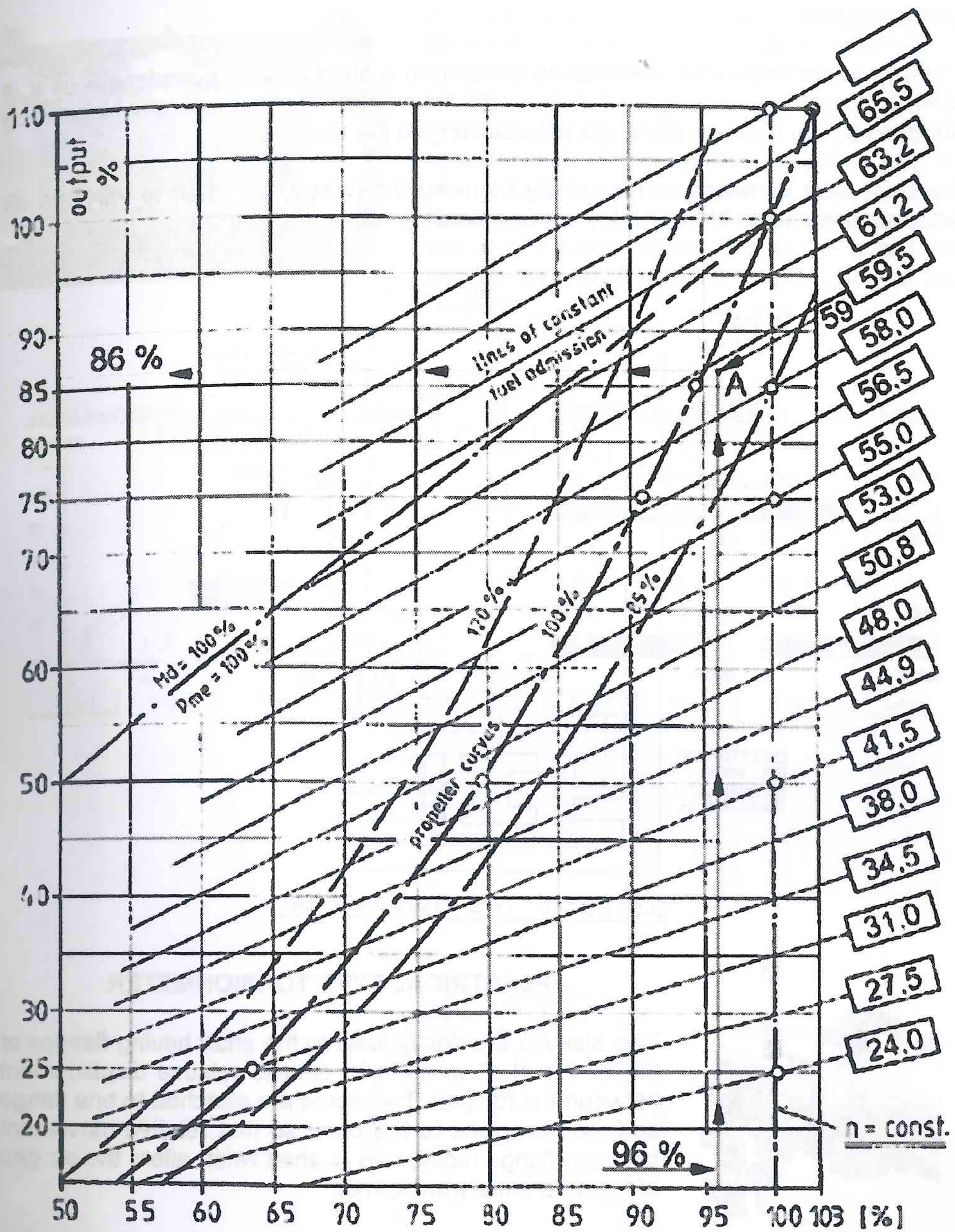
Next take the engines actual speed and fuel pump settings. In this example 432rpm at fuel pump setting of 59

Convert the speed into a percentage of the rated speed. In this case 432rpm is 96% of 450.

Project the 96% of rated speed vertically upwards on the *x* axis, until it intersects with a line drawn parallel to the fuel pump settings for a setting of 59 (Point A)

From point A, project a horizontal line to the *y* axis and read the % power output, in this case 86%

Power = 0.86 × 6200 = 5330kW

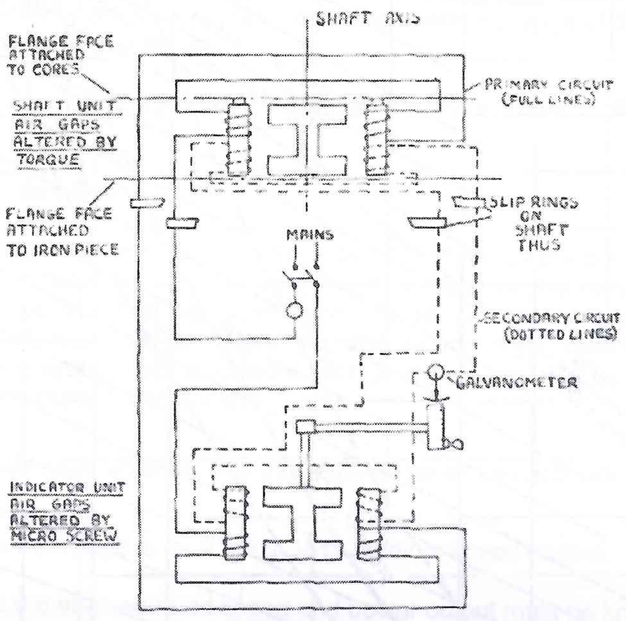


Engine Power Curves MAN B&W L58/64

Torsionmeters

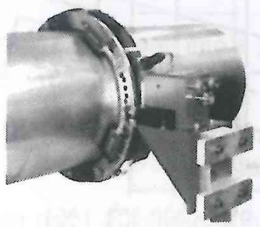
A torsion meter measures the twist (in radians) in a shaft over a given length as it is rotating. Experimental data will have established the shaft constant which when multiplied by the rpm and the angle of twist will give the shaft power.

The shaft twist is measured electrically by using the twist in the shaft to vary the air gap in a transformer. This will vary the resistance and thus the current.



ELECTRICAL TYPE TORSIONMETER

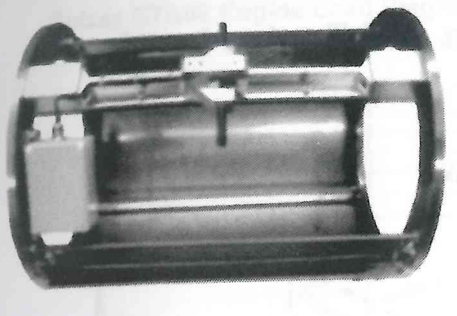
ELECTRICAL TYPE TORSIONMETER



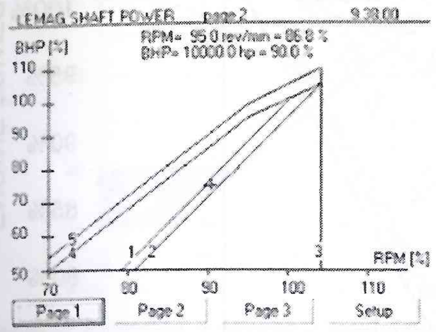
Two sleeves are rigidly fixed to the shaft having flanges at 90° to the shaft axis. Twist causes relative displacement between the flanges. Two cores are attached to one flange and the iron piece to the other so that relative movement between flange faces, due to shaft twist, alters the air gap of the differential transformer.

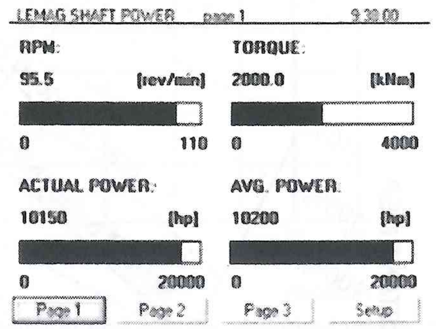
The primary circuit is wound to give the same polarity and the secondary circuits are in opposition. With no torque the air gaps are equal and the two secondary circuits are equal and opposite, but when torque is applied air gaps become unequal and a current flows in the secondary circuit which can be read on the galvanometer. An identical unit is fitted in the indicator box and by rotation of the handle the iron piece can be moved until the air gaps in the indicating unit are identical with those of the shaft unit. This restores the electrical equilibrium in the secondary circuit, as opposed equal currents the galvanometer reads zero, and the amount of movement at the indicating box dial is indicative of the angle of twist restoration required and hence gives the angle of twist for the length of shaft between the two flange faces in the shaft unit.

By application of the meter constant and rev /s the shaft MW is thus determined.

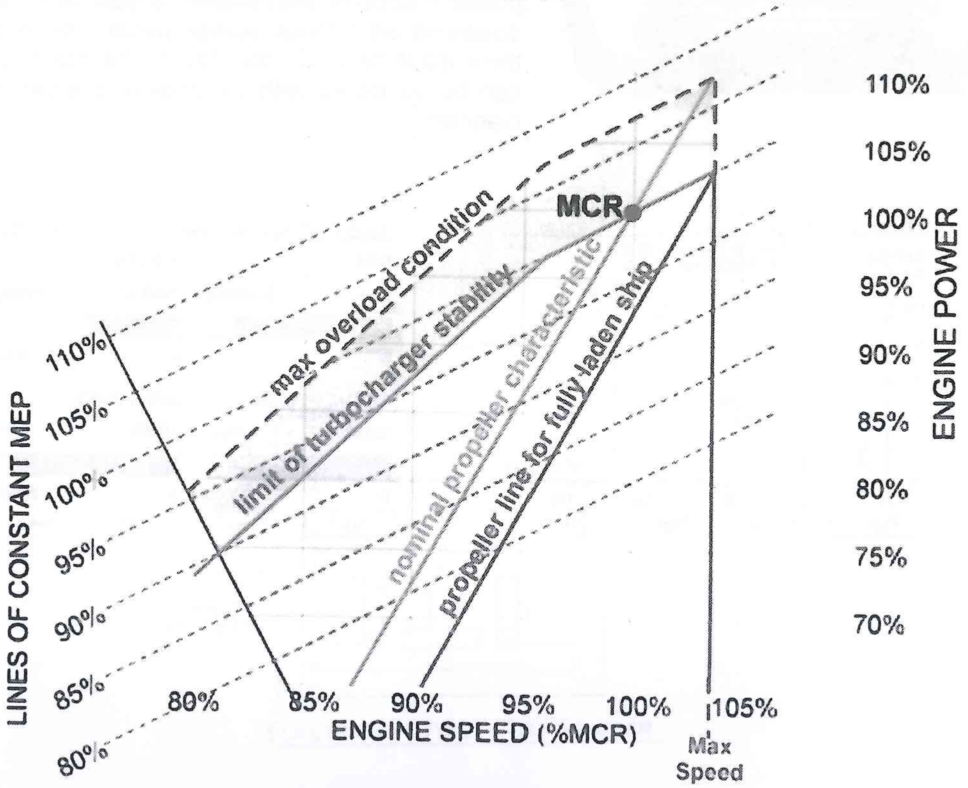


Another method is to use two position encoders between two clamp rings in 500mm apart on the propeller shaft. Using this method it is possible to measure the torque absolutely contact free and with quite high accuracy. Together with the speed, which is also measured contact free, the power output of the engine is calculated and displayed on a touch screen panel mounted in the control room. On the display the actual data can be compared with the original, pre-set load diagram.





Chapter 12: Propeller curves and Engine Load Diagrams



Normal operation

Transient load only (rough seas, acceleration)

Overload

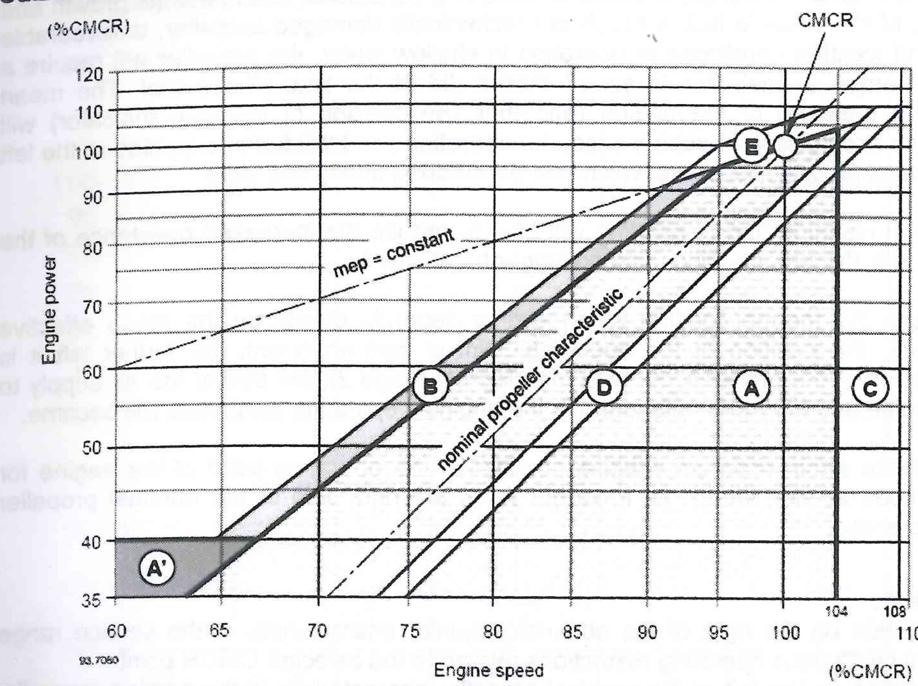
Note: Constant operation in Transient load area indicates fouled hull/propeller

ENGINE - PROPELLER CHARACTERISTICS

Engine builders supply an engine load diagram with their engine to allow the operator to establish whether the engine is being operated within limits.
The nominal propeller characteristic curve is plotted from information gained from running on the testbed where a load is put on the engine to simulate the propeller. This is sometimes referred to as the heavy running curve, which the engine would follow with a badly fouled hull or in heavy seas.

The propeller curve for a fully laden ship represents the engine load with a clean hull in calm weather. This is sometimes referred to as the light running curve. In normal operation the engine should be operating between these two propeller lines. If the ship is in ballast (i.e. light ship), then the operating point will move to the right of the propeller line.

Sulzer RTA96 Engine Load Diagram



There is a defined relationship between the propeller speed and the absorbed power in ships equipped with fixed pitch propellers.

With a given propeller this relationship mainly depends on its rotational speed.
The following formula provides us with an approximation which is adequate for the general consideration of conventional vessels:

$$\frac{P_1}{P_2} = \left(\frac{n_1}{n_2}\right)^3$$

Its graph is called the propeller characteristic.
If an engine is in good condition and properly supplied with air (i.e. turbocharger(s) in good order and the air and exhaust lines have low additional resistance) and the effective strokes of the fuel pump plungers are properly adjusted, the mean effective pressure developed under service condition according to the specific reading of the load indicator corresponds approximately with the mean effective pressure established for this particular position on the test bed.

In the diagram, the propeller characteristic line through the point of CMCR (Contract Maximum Continuous Rating), i.e. nominal power at nominal engine speed (100% power at 100% engine speed) is called the nominal propeller characteristic. Engines which are to be employed for the propulsion of vessels with fixed propellers are loaded on the test bed according to this propeller characteristic. However, the power requirement of a new ship with a smooth and clean hull should be less and correspond to the range D.

With increasing resistance, changes in wake flow conditions, due to marine growth and ageing of the vessel's hull, a rough or mechanically damaged propeller, unfavourable sea and weather conditions or operation in shallow water, the propeller will require a higher torque to maintain its speed than it did at the time of sea trial. The mean effective pressure of the engine (and thus the position of the load indicator) will increase accordingly. In such a case, the operating point will then be located to the left of the original propeller curve which was established during sea trials.

Although cleaning and re-painting will help to reduce the increased resistance of the ship's hull, the new condition can no longer be attained.

Whereas the thermal loading of an engine depends chiefly on the mean effective pressure, the position of the operating point is also important; the farther left it is situated from the propeller curve in the diagram (page 2), the poorer the air supply to the engine and the more unfavourable the engine's operating conditions will become.

In order to attain optimum working conditions, the operating point of the engine for continuous service should lie in **range A** on the right side of the nominal propeller characteristic.

Range A:
The portion on the right of the nominal propeller characteristic is the service range without continuous operating restrictions related to the selected CMCR point.
The portion on the left of the nominal propeller characteristic is the service range for transient operating conditions (acceleration) and should be avoided for continuous operation.

Range A':
Maximum permissible engine power 40% CMCR from approx. 50% up to 67% of CMCR speed.

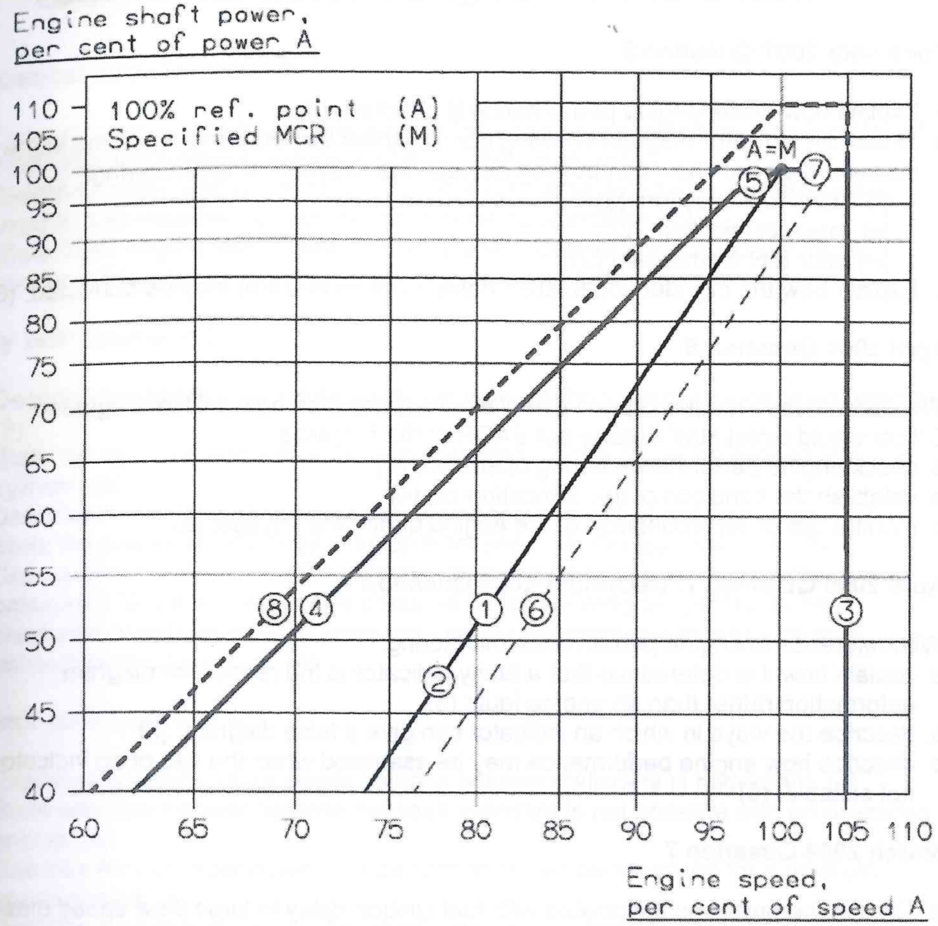
Range B:
Service range with operational time limit, follows a characteristic: $P \approx n^{2.45}$
This characteristic originates from the reference point 95% CMCR power and 95% CMCR speed. With longer operating time in this range, thermal overloading and possible resulting engine damage may be expected.

Range C:
Service range with overspeed of 104 to 108% of CMCR speed, only permissible during sea trials to demonstrate the CMCR power in presence of authorized representatives of engine builder.

Range D:
Recommended layout range for fixed pitch propeller, valid for the maximum draught, clean hull under contractual
weather and sea conditions.

Range E:
Overload range permissible only for maximum one hour during seatrials in presence of authorized representatives of engine builder.

Load Diagram for MAN B&W MC 60 Engine



Line 1: Propeller curve through point A.

Line 2: Propeller curve - heavy running, recommended limit for fouled hull at calm weather conditions.

Line 3: Speed limit.

Line 4: Torque/speed limit.

Line 5: Mean effective pressure limit.

Line 6: Propeller curve - light running (range: 2.5-5.0%), for clean hull and calm weather conditions.

Line 7: Power limit for continuous running.

Line 8: Overload limit.

Chapter 13: Chief Engineer Exam Questions

December 2007 Question 8

a. Explain how diesel engine performance is checked. (4)
b. Sketch an indicator diagram showing good cylinder combustion and on the same indicator diagram show combustion defects due to EACH of the following:
   (i) early fuel injection; (2)
   (ii) late fuel injection; (2)
   (iii) poor fuel atomisation (2)
c. Explain how the cylinder combustion defects drawn in Q8(b) may be corrected. (6)

April 2006 Question 5

With reference to medium speed diesel engines, describe how a Chief Engineer Officer would direct staff to carry out EACH of the following:
a. check engine performance during operation; (6)
b. establish the condition of the lubricating oil; (4)
c. monitor the general condition of the engine during the voyage; (6)

April 2005 Question 7, December 2002 Question 7

With reference to engine performance monitoring:
a. explain how it is determined that a faulty indicator is the reason for diagram deformation rather than an engine fault; (5)
b. describe the ways in which an indicator can give a false diagram; (6)
c. describe how engine performance may be assessed when the use of an indicator is not suitable. (5)

March 2004 Question 7

a. Explain the problems associated with fuel ignition delay in large slow speed diesel engines. (6)
b. Sketch a crank angle/cylinder pressure diagram, showing the results of variation in fuel ignition delay. (6)
c. Describe how variation in fuel ignition delay can be offset. (5)

December 2001 Question 8

The Chief Engineer Officer of a ship currently on a five year time charter has been informed by the owners that a charterers report indicates that the ship is not maintaining service speed over a four week period.
As Chief Engineer Officer explain the investigation and attempted remedy of the situation with respect to the main propulsion system assuming the hull has been recently cleaned. (16)

Chapter 14: Second Engineer Exam Questions

December 2007 Question 2

a. Determine the specific fuel consumption for a main engine with an output power of 10000kW and a daily fuel consumption of 36 tonnes. (4)
b. Sketch *in phase* and *out of phase* indicator diagrams, explaining how these are used to help maintain correct specific fuel oil consumption. (8)
c. State TWO engine faults which may cause an increase in the specific fuel oil consumption. (4)

July 2007 Question 3

a. Describe the operational parameters that indicate the state of cylinder load balance. (3)
b. State the equation and the units used to determine the power developed in a cylinder. (3)
c. Describe the conditions required to give an accurate set of indicator cards. (3)
d. State the problems that may develop from poor cylinder load balance. (3)
e. Draw a power indicator card for a two stroke engine cylinder with maximum pressure 110 bar and scavenge pressure 2 bar, showing temperatures and pressures at maximum firing pressure, maximum compression pressure and exhaust blow down point. (4)

March 2007 Question 4

a. Explain why approximate power balance between cylinders is necessary. (4)
b. State why exact power balance between cylinders is not possible with an operating engine. (4)
c. Describe how cylinder power and performance can be assessed for a medium speed engine. (4)
d. Describe how individual cylinder power can be adjusted. (4)

July 2003 Question 1

With reference to medium speed engines
a. explain why, after a period of operation, there is a variation in the power developed by different cylinders; (5)
b. describe how the power balance of cylinders may be assessed; (5)
c. describe how optimum cylinder power may be restored. (5)

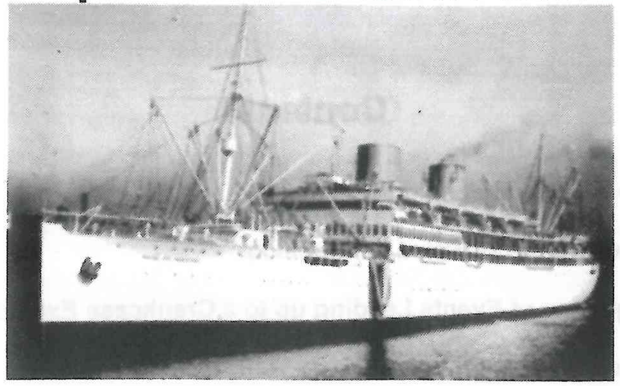
December 2002 Question 2, December 2001 Question 2 (Similar)

a. Outline why different methods are normally used for assessing cylinder power balance in crosshead engines and medium speed engines. (2)
b. State TWO reasons why it is important that all engine cylinders develop approximately the same power. (2)
c. Describe how cylinder power balance may be assessed for a medium speed engine driving an electrical generator. (6)
d. Describe how good cylinder power balance may be restored in a medium speed engine driving an electrical generator. (6)

Crankcase Explosions

Contents

Chapter 1: The Reina Del Pacifico



It was a long time ago (1947). However, it stands out because of the large loss of life; 28 killed and 23 injured. Most of the dead were employees of Harland & Wolff who were on board the vessel as it was undergoing sea trials after a refit.

The reason that the explosion was so devastating is because the crankcases of the four engines were linked together by sump ventilating pipes and vapour extraction pipes. Explosion doors as we understand them today were not fitted; Instead "explosion vents" were fitted at the forward and aft ends of the engines which consisted of a perforated plate covered by a bursting plate.

It is thought that when the engine was stopped because of an overheated piston in the port outer engine, oxygen continued to be drawn into the crankcase by the vapour extraction fan. The oxygen mixed with the oil which was vaporising on contact with the hot piston skirt and then circulated through to the crankcases of the other three engines.

When the port outer engine was restarted the crankcase explosion caused an increase in pressure in the engine room. This forced the crankcase doors of the other engines inwards bringing the revolving cranks of the engines into contact with the bottom of the doors. The flame from the explosion in the port outer engine entered the crankcases igniting the circulated condensed oil mist vapour blowing the doors outwards, and starting the fires in the engine room.

The lessons learnt from this tragedy are reflected in the modifications in design recommended to the inquiry:

1. The introduction of baffles between each individual crank chamber.
2. Complete separation of oil sump ventilating pipes.
3. Complete separation of crankcase vapour extractor pipes.
4. Re-design of crankcase doors to permit quick inspection of the crankcases.
5. Re-design of explosion discs on ends of crankcases.
6. Provision of carbon dioxide gas system as a blanketing medium for the crankcase.
7. Remote recording of cylinder liner temperatures.
8. Increasing the diametrical working clearance of the pistons

Some of these recommendations which have been adopted and built into modern designs have led to the safer engines built today. The replacement of the old type of explosion discs with the modern explosion doors has also reduced (but not eliminated) the risk of the severe explosion.

Chapter 2

THE MERCHANT SHIPPING ACT, 1894

REPORT OF COURT

(No. 7951)

m.v. "Reina del Pacifico" O.N. 162339

In the matter of a Formal Investigation held at the County Courthouse, Crumlin Road, Belfast, on the 20th, 21st, 22nd, 23rd, 26th, 27th, 28th, 29th, and 30th days of April, 1948, and the 1st, 3rd, 4th, 5th, 6th, 7th, 8th, 10th, 11th, 12th, and 13th days of May, 1948, before J. H. Campbell, Esq., R.C., KM., assisted by John Lamb, Esq., O.B.E., A.M.I.M.E., F.I.P., W. L. Nelson, Esq., O.B.E., M.I. Mar. E., and Ivor J. Gray, Esq., A.M.I.N.A., M.I. Mar. E, into the circumstances attending the explosion in the machinery space of the Motor Vessel" Reina del Pacifico," which occurred on the 11th of September, 1947, while the vessel was undergoing sea trials in the Irish Sea and Firth of Clyde.

The Court, having carefully inquired into the circumstances attending. the above-mentioned shipping casualty, finds, for the reasons stated. in the annex hereto, that the primary cause of the explosion in the main machinery space of the" Reina del Pacifico" was due to the piston of No. 2 port outer engine overheating and igniting an inflammable mixture present in the crank chamber of said engine.

Dated this 5th day of June, 1948.

J. H. CAMPBELL, *judge*.

We concur in the above Report.

JOHN LAMB
W. L. NELSON } *Assessors*.
IVOR J. GRAY

QUESTIONS AND ANSWERS.

The Court's answers to the questions submitted by the Ministry of Transport are as follows :

Q. 1. By whom was the vessel owned?
A. Pacific Steam Navigation Company, of Liverpool.
Q. 2. How long had the vessel been so owned?
A. Seventeen years
Q. 3. (a) By whom was the vessel built?
A. Messrs. Harland & Wolff, Limited, Belfast.
(b) By whom were the engines designed, built and fitted?
A. Messrs. Harland & Wolff's adaptation of Burmeister & Wain design; built and fitted by Messrs. Harland & Wolff, Limited, Belfast.
Q. 4. Where and when was the vessel built ?
A. At Belfast, in the year 1931.
Q. 5. Was the vessel disabled by explosion in the main machinery space at approximately 4.46 p.m. B.S.T. on the 11th day of September, 1947?
A. Yes.
Q. 6. Did such explosion, if any, occur when the vessel was approximately 7 miles N.E. of. Copeland Island in North Channel, Irish Sea?
A. Yes.
Q. 7. Was the vessel undergoing sea trials when the explosion occurred?
A. Yes.
Q. 8. (a) When did the vessel last leave port before the explosion occurred?
A. Vessel sailed on intended trial voyage at 7.15 a.m. on the 10th September, 1947.
(b) What port did she so leave?
A. Belfast.
(c) Was the vessel in good and seaworthy condition when she last left port ?
A. Yes.
Q. 9. How many persons lost their lives as the result of the explosion?
A. Twenty-eight.
Q. 10. For approximately 12 months prior to the date of the explosion had the "Reina del Pacifico" been undergoing a complete refit together with machinery overhaul at Belfast?
A. The ship did not undergo a "complete" refit during the twelve months prior to date of the explosion. The reconditioning was confined to that set out in the repair list dated 27th September, 1946.
Q. 11. Did such refit and overhaul follow a period of employment as a troopship on war service?
A. Yes.
Q. 12. Prior. to the war had the "Reina del Pacifico" been employed as a passenger liner between the United Kingdom and the West Coast of South America?
A. Yes.
Q. 13. Had any explosion similar to the one which is the subject of the Inquiry occurred on board the vessel previously?
A. No
Q. 14. Are there any records of any parts of the main engines overheating prior to the refit and overhaul referred to in Question 10? If so, what action was taken thereon.
A. Yes. Lubricating oil was applied by hand to the overheated parts, and fuel was either cut off the cylinder concerned, or the engine control lever was brought to the "stop position"

Q. 15. Had the main engines of the vessel given satisfaction during the service as a liner and as a troopship?

A. Yes.

Q. 16. Was a sea trial commenced on the 2nd day of September, 1947

A. Yes.

Q. 17. When was such trial (a) commenced, (b) abandoned; and (c) for what reason was it abandoned?

A. (a) 2nd September, 1947.
(b) 2nd September, 1947.
(c) Owing to serious overheating of piston and cylinder of No. 6 starboard outer, No. 11 port outer, and No. 2 port outer to a lesser degree.

Q. 18. What defects, if any, were discovered in the vessel's main engines as a result of such trial? What action was taken with regard to such defects?

A. (a) Port outer No. 11 piston and cylinder liner badly scored; starboard outer No. 6 cylinder jacket cracked; piston and cylinder liner badly scored; port outer No. 2 cylinder liner and piston found in good order.
(b) Port outer No. 11 cylinder liner arid piston renewed; starboard outer No. 6 cylinder jacket, liner and piston renewed; port outer No. 2 thoroughly cleaned, carefully examined and no renewals or repairs effected.

Q. 19. When the second trial was commenced on the 10th day of September, 1947,
(a) Who was in command of the vessel?

A. The Master, viz. John Whitehouse, who was on the ship's register on the second trials.

(b) Who was responsible for the manning and running of the machinery (other than main engines)?

A. Having regard to paragraph 3 of Exhibit "T", which document was proved by Robert T. Oxburgh, the manning and running of the machinery (other than the main engines) was the joint responsibility of the owners and repairers.

(c) Who was responsible for the manning and running of the main engines?

A. Witnesses on behalf of the Pacific Steam Navigation Company, Stated, and their Counsel contended, that no servant of the company was responsible for the manning and running of the main engines. It was submitted on behalf of the Pacific Steam Navigation Company, that the manning and running were the responsibility of Messrs. Harland & Wolff, Limited. Counsel on behalf of Messrs. Harland & Wolff, Limited, submitted that the responsibility was, joint. The Assistant Head Foreman of Messrs. Harland & Wolff swore that the control and responsibility was that of the Owners. In these circumstances it would appear that when the second trials commenced on the 10th September, 1947, neither of the parties considered themselves exclusively responsible for the manning and running of the main engines,

(d) Were the machinery and main engines adequately manned and were all proper precautions taken?

A. In view of answer to (c) it is clear that an essential precaution (viz. that of having some official of one or other of the parties in control and responsible for the manning and running of the main engines), was lacking. Further, it is apparent that there was divided control and responsibility in the manning and running of the main engines, which in the circumstances, was unsatisfactory.

Q. 20. (a) Did any material defects in the machinery (including main engines) arise during the second trial?

A. Yes.

(b) If so, what were they, and what action was in respect of them?

A. The fuel pump spill valve of No. 8 cylinder star outer engine not functioning; No. 2 cylinder lin port outer engine overheating. Spill valve chang without desired improvement.
No. 2 cylinder fuel valve of port outer engin passed, and in a short space of time control brought to "stop" position. Within four to minutes later, the control lever was put "starting" position.

Q. 21. (a) Were any such defects observed immedi prior to the explosion?

A. Yes

(b) If so, in which part of the, machinery inclu main engines were the defects, if any, observed?

A. See answer to Question 20.

(c) What were the defects?

A. See answer to Question 20.

(d) What action was taken in consequence of discovery of such defects?

A. See answer to Question 20.

Q. 22. (a) At what speed, in revolutions, were the engines running at the time .of the explosion?

A. The three engines, port inner, starboard inner, starboard outer, were running at 128 to revolutions per minute as far as could be ascertai from the evidence.
With reference to the port outer engine, it is diffi to estimate with any degree of accuracy revolutions at which this engine was runn However, it would appear from the evidence that revolutions of this engine, were, at the time of explosion, approximately fifty per minute.

(b) For how long had the main engines b running, and at what speeds?

A. From 7.15 a.m. on the 10th September, 1947, u 4.46 p.m. on the 11th September, 1947, at vary speeds up to full speed.

(c) Had the main engines been running at a hig speed at any time during the trial on which explosion occurred?

A. No accurate records of engine revolutions during part of the trial were forthcoming, but there indications that the main engines were operating their maximum speed from noon on the 1l September.

Q. 23. (a) Was the accident the result of one explosion more than one?

A. More than one.

(b) If more than one explosion, how many?

A. Four.

(c) If more than one explosion, were the explosio simultaneous or separate?

A. The explosions were separate, but according to t evidence were almost simultaneous

(d) If there were more explosions than one, were al the explosions the result of a common cause?

A. Yes

Q. 24. Where did the explosion or explosions originate?

A. The first explosion occurred in number two cran chamber of port outer engine. The other thre explosions occurred in the crank cases of port inne starboard inner, and starboard outer engines.

Q. 25. (a) What were the causes of the primary explosio or explosions and any subsequent explosions?

A. The primary explosion was caused by th overheating of No. 2 piston of the part outer engin The other three explosions were due to th inflammable, content of the crankcases bein ignited.

(b) What explosive mixtures were involved?

A. A mixture of atomized or vapourised lubricating oil and air.

Q. 26 Were all four main engines involved in the explosion or explosions?

A. Yes.

Q. 27. What was the effect of the explosion or explosions, (a) on the ship's machinery, including main engines?

A. Damage to the closing plates and covers .of crankcases of all four engines.

(b) On the ship's structure?

A. Negligible damage to ship's structure; access ladders and lifting beams in engine room badly damaged.

Q. 28. (a) Were any fires started in the vessel as a result of the explosion?

A. Yes

(b) Was fire fighting promptly organised and applied?

A. Yes.

Q. 29. Were rescue and first-aid measures promptly organised and applied?

A. Yes.

Q. 30. Were such explosion or explosions connected with the fuel oil system or with the lubricating oil system?

A. The lubricating oil system.

Q. 31. Was the lubricating oil of proper quality, quantity and condition?

A. Yes

Q. 32. Was the fuel oil of proper quality, quantity and condition?

A. Yes

Q. 33. Was the lubricating oil in any of the crankcases found to be contaminated by fuel oil or any other substance? If so, did such contamination, increase the danger of explosion?

A. Yes, to a very slight degree. This contamination did not increase the danger of explosion.

Q. 34. Were such explosion or explosions caused by or contributed to by any wrongful act or default or error of judgment of the owners of the vessel, The Pacific Steam Navigation Company, or any of their servants?

A. In view of the conflicting nature of the evidence given, the Court is not prepared to hold that the explosion in part outer No. 2 was caused by or contributed to by any wrongful act on the part of any servant of the Pacific Steam Navigation Company, except in so far as is disclosed in answer to Question 19 (c) and (d).

Q. 35. Were such explosion or explosions caused or contributed to by any wrongful act or default or error of judgment of the repairers, Messrs. Harland & Wolff, Limited, or any of their servants?

A. In view of the conflicting nature of the evidence given, the Court is not prepared to hold that the explosion in port outer No. 2 was caused by or contributed to by any wrongful act on the part of any servant of Messrs. Harland & Wolff, Limited, Belfast, except in so far as is disclosed in answer to question nineteen (c) and (d).

This Inquiry was held in Belfast, on the 20th, 21st, 22nd, 23rd, 26th, 27th, 28th, 29th, and 30th days of April, 1948, and the 1st, 3rd, 4th, 5th, 6th, 7th, 8th, 10th, 11th, 12th, and 13th days of May, 1948.

L.E.Curran,Esq., K.C., M.P., His Majesty's Attorney General for Northern Ireland, and F. A. L. Harrison, Esq., Barrister-at-Law (instructed by John B. Getty, Esq., for Treasury Solicitor), appeared on behalf of the Ministry of Transport. K S. Carpmael, Esq., K.C., W. W. B. Topping, Esq., K.C., M.P., and E. W. Jones, Esq., Barrister-at-Law (instructed by Messrs. C. & H. Jefferson for Messrs; Hill Dickinson and Company, Liverpool), appear on behalf of, Messrs. Harland and Wolff, Limited, Belfast, Contractors of the Repairers. C. L. Shiel, Esq., K.C., and G.B. Hanna, Esq., K.C, (instructed by Messrs. McKinty and Wright for Messrs. Batesons and Company, Liverpool), appeared on behalf of the Owners of the "Reina del Pacifico" (The Pacific Steam Navigation Company).

James McSparran, .Esq., K.C., M_P., and H. A. McVeigh, Esq., Barrister-at-Law (instructed by F. Hanna, Esq., M.P.) appeared on behalf of representatives of Edward McAllister, James E. Barnes, Robert Currie Ellis, William Mills, James McAllister, J.D. McBlain, Robert C. McClure, Wesley Patterson, John Redmond, Samuel Richmond, James Bernard Savage, and Thomas Wilson, deceased, and of Robert Robinson, William Magee, R Downey, Wm. Morrison, George Crowe and Cyril Osberg, injured, all of whom were made parties to the proceedings on the application of Mr. McSparran.

Francis Hamia, Esq., M.P., Solicitor, appeared on behalf of representatives, of Ferney B. Glenfield, deceased, and representatives of Patrick J. Dunn, deceased, and D. R. Martin, Esq., Solicitor for Messrs. Keightley Jenkins and Company, Meols, Cheshire, appeared on behalf of the representatives of Harold Fay, deceased, and on behalf of the Navigators and Engineer Officers Union.

G.B.H. Currie, Esq., Barrister-at-Law, (instructed by Messrs. Percy Hughes and Roberts, Solicitors, Birkenhead) appeared on behalf of representatives of Frederick Johnston, deceased. Mr. Currie (instructed by R J. Livesey, Solicitors, Liverpool), also appeared for the representatives of A. H. Jones, deceased; and both parties were made parties to the proceedings on the application' of Mr. Currie.

J. I. P. McCracken, Esq., Barrister-at-Law (instructed by Messrs. George McCracken and Company), appeared on behalf of representatives of Charles Thompson, deceased, and representatives of Hugh Doherty, deceased.

W. P. E. Alexander, Esq., Solicitor, appeared on behalf of representatives of Robert J. Thompson, deceased.

The m.v. "Reina del Pacifico" is a quadruple screw motor passenger vessel of 17,702 gross tons, being 551.3 feet in length, 76.3 feet in beam and 37.9 feet in depth registered dimensions. The vessel was built and engined by Messrs. Harland and Wolff, Limited, in 1931, to the order of the Pacific Steam Navigation Company, with registered offices at Liverpool, and at which port she is registered and continuously owned by this Company to the present day. She has five decks, A. to E., E. deck being uppermost and surmounted by a boat deck, which in turn is surmounted by a house which forms accommodation and which carries the funnels. She has five lower holds for cargo.

The vessel was designed for passenger service between the United Kingdom and the West Coast of South America, and was so employed, excepting during the war, when she was on charter to H.M. Government for trooping purposes.

The assigned summer draught is 31 feet 2¾ inches. At the time of the explosion, however, the draughts were 17 feet 11 inches forward and 26 feet 3 inches aft.

Classification. The vessel held the class of 100 A.1. with freeboard in Lloyd's Register Book and was undergoing periodical special survey of hull and machinery during reconditioning in association with a renewal survey for load line; and certificates for special survey of hull and machinery and for renewal of load line were to have been issued on the completion of satisfactory trials.

Passenger certificates. Since 1931 the vessel has held a Board of Trade or Ministry of Transport Passenger and Safety Certificate for International voyages, carrying 886 passengers and 301 crew. Her last certificate expired on the 5th October, 1946, at the time the vessel arrived for reconditioning at Belfast. She was surveyed during reconditioning, and a further twelve months certificate was to have been issued on the completion of satisfactory trials.

EQUIPMENT ON BOARD.

Life-saving appliances. These complied with the Ministry's requirements for a foreign-going passenger vessel of this size, and comprised the following:

16 lifeboats including 1 motor boat and 1 boat with auxiliary motor, capable of carrying a total of 1,349 persons.
14 Buoyant apparatus for 308 persons. 1,402 Lifejackets.
18 Lifebuoys.

Medical stores. These were supplied in accordance with the Merchant Shipping Medical Scales, 1945, Scale Ill, except that the Laudanum, Morphine tablets and Omnopon were not included. Owing to the dangerous nature of these drugs they had been removed from the ship in Belfast to avoid accidents by pilfering.

There were available, however, the 16 first aid kits from the lifeboats together with 50 supplementary "A" kits, and these provided an ample supply of Morphine tablets under normal circumstances.

The ship, being' on a coastal voyage, was not required to carry any medical stores beyond a first aid kit, but she was in fact amply provided for the numbers on board at the trial.

Fire fighting equipment. The fire fighting equipment on board at the time of the second sea trial on 10th and 11th September, complied with the Ministry's requirements for a foreign going passenger motor vessel of this size except in regard to a second smoke helmet which had been damaged and was then ashore for repair. The fire fighting equipment comprised the following:-

2 - 95 ton vertical rotary fire pumps in engine room.
1 - 95 ton vertical rotary emergency fire pump in generator room.
2½ inch fire main on each side of each deck with valves and couplings at 90 to 100 feet intervals.
41-50 feet lengths of 2tinch canvas hose. Fireproof bulkheads on all decks.
51-2 gallon foam type extinguishers distributed throughout accommodation.
12-2 gallon foam type extinguishers distributed throughout machinery spaces.
12-1 quart carbon tetra chloride extinguishers at switch boards and cinema.
4 Foamite hopper type continuous foam generators, two in main engine room and two in generator room, each capable of discharging 400 gallons of foam per minute.
2-10 gallon foam extinguishers, one at waste heat boilers and one in generator room.
"Rich" fire detection and steam file extinguishing installation for each cargo space.
1-Smoke helmet and equipment (see note above)

ARRANGEMENT AND DESCRIPTION OF MACHINERY

Arrangement of machinery. The machinery space amidships and extends from water-tight bulkhead 35 forward 49 aft, and is divided as follows:

Watertight bulkhead 35 Forward to 13 Forward.	Oil fuel bunkers and settling tanks
Watertight bulkhead 13 Forward to 2 Aft.	Auxiliary generator room.
Watertight bulkhead.2 Aft to 34 Aft.	Main engine room.
Watertight bulkhead 34 Aft to 49 Aft.	Refrigerating machinery

The main engine room is 83 feet mean length by 76 f breadth by 28 feet high with a light and air casing 40 feet lo and 17 feet wide extending upwards to the skylight.

Description of machinery. The propelling machin comprises four independent engines placed abreast, each drivi its own propeller direct, and each having 12 cylinders 6 millimeters (24.8 inches) bore by 1,200 millimeters (47 inches) stroke, the total service power being 16,000 Brake Ho Power at 130 revolutions per minute. giving the vessel a servi speed of about 19 knots. The engines are designed for 18,0 Brake Horse Power:

The engines are Harland and Wolff s adaptation Burmeister and Wain's design, and are of the four stroke sin acting trunk piston type, pressure charged by exhaust gas turbi driven air blowers.

The cam shaft is chain driven from the crankshaft betwe Nos. 6 and 7 cylinders. The fuel valves are of the airle injection type with a group of three fuel pumps on the inboa side of Nos. 2,5, 8, and 11 cylinders of each engine.

All auxiliaries are electrically driven by power supplied four Diesel engine-driven generators each of 350 kilowa capacity. The Diesel engines driving these generators are of t four stroke, six cylinder, trunk piston, airless injection type Harland and Wolff design.

Exhaust and pressure charge system. The exhaust gas fro the 12 cylinders of each engine passes forward through t respective manifolds to turbines mounted opposite the forwa end of each engine on a common flat 10 feet above the low platform level. Each turbine drives a rotary blower which dra air from the engine room through a mesh protected openi below the flat. The discharge from each blower is trunked a above and to the rear of each engine with distributing pipes in the cylinder air inlet valves. Arrangements are provided to b pass the exhaust gas in an emergency.

Waste heat boilers. On a platform at "D" deck two waste he boilers are each arranged to take the exhaust from two ma engines. Each boiler has two oil burners for use when t exhaust is being by-passed direct to the silencers and funn Diesel fuel is supplied from an overhead tank to the burners a is injected by air under pressure from electrically driv blowers. The steam is used to supply hot water for domest purposes by means of calorifiers.

When the incident, which has been the subject of this Inquir took place, the boilers had been working on exhaust gas f about three hours with fuel completely shut off.

Main engine fuel system. Diesel fuel is fed by gravity fro service tanks on the after bulkhead to two working and o stand-by electrically driven low pressure pumps of the rota type. These pumps normally deliver direct to the fuel pumps the front of the engines, which in turn deliver the fuel under hi pressure to the fuel valves in the cylinder head.

Whilst manoeuvring, the low pressure pumps deliver to t boiler fuel tank. The engine fuel pumps in turn draw their supp by means of gravity from this tank. The timing and control the fuel injection which determines the speed of the engine arranged at the fuel pumps by means of a spill device, the pum being main engine driven and located at approximately five fe above engine room platform.

The fuel pumps are mounted entirely outside the crankcase and there is no way in the event of leakage from these pumps, in which fuel can find its way into the crankcase.

The controls for the two port engines are brought to the forward end of the port inner engine, while those of the starboard engines are brought to the forward end of the starboard inner engine.

Crankcases. All crankcases are totally enclosed, the doors on the sides, both back and front, being readily removable, except those in way of the fuel pumps. The top of each crankcase is closed by cast iron diaphragm plates between the" A" frames. Cylinder liners project 7½ inches into the crankcases while the pistons, when at the end of their down stroke, project 16½ inches below the liners. The bottom of each crankcase is oil-tight and has been described to the Court under the heading" drain tanks."

The crankcase doors are of comparatively light steel 3/32 inch thick, suitably stiffened round the edge, and secured by a special fonn of dog. The doors are 6 feet 3 inches high by 3 feet 3 inches wide and weigh 155 pounds each. There are, in addition, small doors 2 feet high by 3 feet 3 inches wide of similar construction situated below the fuel pumps. Each engine thus has 20 large and 4 small crankcase doors, also a back gear case door. The end doors are of cast iron one-half inch thick, suitably ribbed, and held by studs and nuts.

The crankcase is common to the 12 units by way of apertures about 3 feet high by 2 feet 10 inches wide above the main bearings, and 1 foot 10½ inches high by 7 feet 6 inches wide below the main bearing together with a 4 inches by 5½ inches aperture in both upper corners of each "A" frame. In addition, the chain driving the cam and fuel pump shafts between Nos. 6 and 7 cylinders is totally enclosed in a cast iron casing 1 inch to ½ inch thick extending to the top of the engine, the whole of this casing being open to the crankcase at the bottom.

Sheet steel splash plates are fitted over each crank web to prevent oil being thrown up on to the cylinder liner.

Crankcase vapour extraction system. To obviate the fouling of the engine room atmosphere by leakage of fumes, an electrically driven fan is installed for each engine. The fans draw through 3½ inch bore pipes connected to the diaphragm plates at Nos. 3 and 10 cylinders, and a 4 inches bore pipe to the top of the chain drive gear case. The vapour is then led through a common 9 inches bore pipe to the fan which discharges through a sea water cooled oil extractor and thence to the top of the funnel. The fans for the two inboard engines are on the blower flat forward, and those for the outboard engines are on an after flat. The discharges from the two starboard engines run into a common trunk at the base of the funnel and those from the two port engines are joined similarly. Although means are provided for draining off any oil collected by the coolers, there is no record of any appreciable amount being collected. Examination of the pipes and coolers after the occurrence showed the cooler tubes to be completely free from oil. The pipes leading from the crankcase to the fan were coated with oil. Oil was also found trapped in the collecting drum at the fan inlet, and finally by the fan impeller, the casing of which was provided with a drain. It would, therefore, seem that the oil mist is trapped mechanically and that any gases present pass through the cooler without condensing.

No special means are provided for the admission of air into the crankcase, but a certain amount can enter by way of the air pipe, and sump drain pipe described under "drain tanks," and through minor leaks at the bottom of the crankcase doors and between the lower ends of the cylinder liners and diaphragm plates.

On the crankcase covers, at the forward and after ends of each engine an explosion vent is provided. These vents are covered by perforated steel plates on the outside of which are fitted millboard discs 0.134 inches thick. This millboard is intended to burst in the event of an explosion in the crankcase.

Lubricating oil system and drain tanks. On the after port side of the engine room there is one working and one stand-by electrically driven rotary pump, one of which draws lubricating oil from one or other of the drain tanks under the port inner and port outer engines. These pumps deliver the oil through the cooler to distribution pipes on the back of the outer engine and the front of the inner engine.

For each unit there is one pipe which conveys the oil to the main bearing, crank pin, and gudgeon pin bearing, by way of holes in crankshaft and connecting rod. From these bearings the oil drains to the engine sump and thence to the drain tank.

At each unit a pipe conveys oil to a trombone pipe situated under each piston. This oil flows through the hollow piston for cooling purposes, and returns through a second trombone pipe into an outlet pipe located along the front, in the case of the outer engine, and the back of the inner engine. The oil returning from the pistons is led into the same drain tank which receives the oil from the bearings. The same system supplies oil to the chain drive.

A similar arrangement applies to the starboard inner and starboard outer engines. with a pair of pumps on the starboard side aft.

The tank top is recessed 2 feet 5 inches deep throughout the length of each engine to form an oil sump, and between Nos. 6 and 7 cylinders two valve controlled outlets of 6 inches bore allow the oil to drain into one or other of the oil drain tanks formed in the double bottom below the sump. Two similar outlets are also placed below No. 12 cylinder on each engine. Sieve plates having 11/16 inch diameter perforations cover drain pots in way of these outlets, and in addition the sump has a working platform 7 inches above the bottom and made of steel plate, having ¾ inch diameter perforations. During the basin and sea trials, the oil from the port outer and port inner engines was being drained into the port inner tank, and the port outer tank which contained a certain quantity of clean oil was not in use. The starboard inner drain tank was similarly used for the starboard inner and outer engines. There is no connection between the drain tanks of the port engines and the drain tanks of the starboard engines except that from two 4 inch branches on each drain tank a 5 inch air pipe is led to the top of the funnel, the pipes from the two outer tanks being joined to a common pipe at "D" deck and the two inner tank air pipes are similarly arranged.

Independent centrifugal separators are used for purifying the lubricating oil and fuel oil. The pipe systems of these separators are not connected in any way. The separators are located on the starboard side, the fuel oil separators being forward, and those for the lubricating oil situated at the after end.

Lubricating oil supplies. Log entries dated 8th September, 1947, show that the main engine crankcase oil was disposed as follows :-

Port outer double bottom drain tank 1,680 gallons (not in circulation at trial).

Port inner double bottom drain tank 4,760 gallons (in circulation at trial).

Starboard inner double bottom drain tank 4,760 gallons (in circulation at trial).

Starboard outer double bottom drain tank 1,680 gallons (not in circulation at trial).

Reserve tank at "A" deck 800 gallons (not in circulation at trial).

The whole of the above oil was unused prior *to* the basin trial on 6th August, 1947, and was supplied by W. B. Dick & Company, 8,000 gallons being supplied on 25th July, 1947, and the remainder at various times prior the vessel arriving in Belfast for refit.

In addition, the following lubricating oils were on board :-

Used main engine crankcase lubricating oil in starboard outerstorage tank .	1,600 gallons
Auxiliary engine crankcase oil in starboard inner storage tank CY3 grade	1,660 gallons
Cylinder lubricating oil in cylinder oil storage tank	280 gallons

There was also on board small quantities of stern tube oil, compressor oil, and telemotor oil.

Fuel. No diesel fuel was taken aboard in Belfast, and at the time of the trial there remained 734 tons in deep tanks Nos.1 and 2,3 and 4. In addition there was 8½ tons in the service tanks. The fuel was supplied by the Shell Company of Egypt in july and August, 1946. It was gas oil having a specific gravity of 0.83 at 90° Fahrenheit, closed flash point of 190° Fahrenheit, and 31 Redwoods viscosity at 140° Fahrenheit in seconds per 50 cubic centimetres.

Engine Room Ventilation. A pair of 45 inch fans situated forward of the engine room skylight provide positive ventilation to all parts of the main engine room at various heights, and in addition a discharge trunk is carried below the bottom platform in each forward corner. A pair of 40 inch fans ventilate the auxiliary engine room.

In formulating this report it is deemed necessary to deal in detail with an outstanding consideration which has arisen in the course of the Inquiry. A glance at the history of this vessel from the viewpoint of the main engines, will indicate that overheating of the trunk type pistons was always a probability, especially when new liners or new liners and pistons, had been fitted. There is clear evidence that on at least three occasions minor crankcase fires occurred, which were directly attributable to overheated pistons. It is equally clear that the Engineer Officers of the "Reina del Pacifico" were fully alive to this danger at a relatively early date from the vessel coming into service. Details were given in the course of evidence of each occasion of overheating, the effect of the overheating, the measures taken to cool the overheated parts, and of occasions when renewals were found necessary. The inference to be drawn from this information is that an affected piston could reach a temperature sufficiently high to ignite lubricating oil and cause fire in a crankcase.

Coming to the incident of the 11th of September, 1947, it is significant that a different set of conditions prevailed, inasmuch as, on this occasion, the engine had been running - after an extensive overhaul - for a period of approximately thirty hours. During this period full power had been developed for the last 6½ hours. Over this latter period nothing untoward occurred, except that the fuel pump of No. 8 cylinder starboard outer engine had not functioned properly. This defect persisted notwithstanding that all measures possible in the circumstances had been taken. The first indication that something was seriously wrong with any of the engines was a warning shout given by an employee of Messrs. Harland & Wolff, to the effect that No. 2 cylinder liner of the port outer engine was hot; whereupon the control lever of that engine was immediately brought to the "stop" position by Mr. Oxburgh, the Owners' Chief Superintendent Engineer. After a few minutes, and without any action being taken to cool the over-heated parts, the control lever was moved to the "starting" position by Mr. Owen, Assistant to Mr. Oxburgh. Shortly after this last movement of the control lever an explosion occurred. It should be noted, however, that prior to putting the control lever to the "stop" position, the fuel valve of No. 2 port outer cylinder was by-passed by Mr. Forbes, one of the ship's engineers.

In order to appreciate the circumstances leading up to the explosion, it is necessary to review what occurred from the ship's arrival at Belfast for re-fit at Messrs. Harland & Wolff's, in September, 1946, until the date of the explosion. From the evidence it would appear that the usual procedure in regard to the supervision during the overhaul of the machiner followed. This being so, it was the duty of Messrs. Harl Wolff to carry out in a proper and workmanlike manner, the satisfaction of the Owners, the various items of enumerated on the repair list prepared by the Owners furnished to the Repairers. The Owners' representatives on part, one of whom was always present, considered it their to inspect each important part of the machinery, which inc pistons and cylinder liners, before such parts were refitted.

The question of responsibility for the manning and runni the machinery of the "Reina del Pacifico" during trials arises. It is to be noted that the "Reina del Pacifico" was new ship. In the case of a new ship, the manning and runni entirely the responsibility of the Builders, until the sh handed over to the Purchasers. When the machinery of a that has been in service for a number of years, as in the ca the "Reina del Pacifico," has undergone a big overhaul, customary for a reliability trial at sea to be carried out. It is customary, however, for such a ship to be subjected to s trials, unless some alterations have been made which m affect the ship's speed in relation to the revolutions of prop or propellers. No such alterations were carried out on the "R del Pacifico" during the re-fit preceding the sea trial question. When new machinery is installed the Builders concerned with every part of it, but when being overhauled service, the repairers are only concerned with the it mentioned on the repair list compiled by the Owners.

The question of the actual explosion will now be conside During the Inquiry, two theories were advanced in evidenc to the origin of the explosion:

(1) That an explosion of moderate violence occurred in engine room between starboard inner and starboard o engines in way of No. 7 unit, at a point about 6 inches above double bottom tank top. This explosion is alleged to have cau the inflammable contents of the starboard outer eng crankcase to ignite and explode, resulting in the inflamma contents of the other three engines being ignited. The in cause of this explosion was claimed to be due to fuel oil leak on to the tank top from a main engine fuel oil supply p through a hole of about 1 m/m diameter, and at a pressure about 30 lbs, per square inch. It was further alleged that igniting agent was a spark caused either by blowing through cylinder indicator cocks on the starboard inner engine, or by t pipes abrading at a position 3 inches distant from the afo mentioned leakage.

(2) That No. 2 piston on the port outer engine had reache temperature whereby it formed a "hot spot" of sufficient ar and temperature to ignite the inflammable contents in t crankcase of that engine, and to cause an explosion of relativ low pressure. It is further alleged that some of the crankca doors of this engine were blown off, and the burning conte reached the engine room in the form of pressure and flan waves, which found their way into the crankcase of the oth three engines, igniting the inflammable contents and causi further explosions. As the theory appended in (1) of abo paragraph was not pursued, and was actually withdrawn in t course of the Inquiry, no comment is necessary.

As the theory put forward in (2) of previous paragraph strongly supported by the evidence of the actual happenings, an by the technical evidence tendered, it must be concluded that th seat of the first explosion was in No. 2 crank chamber of th port outer engine. The presence of a "hot spot" in the port out engine must be accepted. This leads to the consideration of th possible cause of this particular piston overheating. In th respect it is necessary to have regard to testing, quality o manufacture, standard of repair, initial clearances, supply o lubricating oil, all of which, if correct, should obviate th possibility of overheating. Relative to the consideratio mentioned in the preceding sentence, it is learned from th evidence that:-

(1) The initial diametrical working clearance between piston and cylinder liner is twenty-five thousandths of an inch (0.63 millimeters). Further, that when the pistons are running at a calculated conservative temperature, in the way of the gudgeon pin, of 160° Fahrenheit, the twenty five thousandths clearance is reduced to something of the order of twelve thousandths of an inch. It must be obvious that such a working clearance on a diameter of 24.8 inches is undesirable. Again it should be noted that the primary reason for abandonment of the first sea trial was the serious overheating of two old pistons which had been fitted into new liners, viz. starboard outer No. 6 and port outer No. 11.

(2) The existence of a badly fitted gudgeon pin and/or bush which may cause overheating by mal-alignment, tending to destroy the initial working clearance.

(3) The inaccurate machining of connecting rod eye and foot and piston gudgeon pin boss, which makes for mal-alignment and reduction of initial working clearance.

(4) The existence of an additional piston ring and groove which was situated in such a position that when the piston reached the end of its upstroke, this ring overran the parallel portion of the cylinder liner bore and expanded into the bell mouth, thus making for mal-alignment of the piston and undue pressure on the gudgeon bearing.

(5) Inefficient lubrication of the piston owing to inaccurate timing of the cylinder lubricator, and the absence of a non-return valve fitted at the nearest point to the cylinder.

The reason for No. 2 port outer engine piston becoming overheated has not been fully determined to the satisfaction of the Court. The Court, however, is of opinion that whilst four of the reasons set out in preceding paragraphs (2,3,4,5), would be sufficient to cause an increased operating temperature, the extent of these faults found in the case of this piston was not sufficiently serious to bring about the degree of overheating attained. With regard to No. (1), it is felt that there is a distinct possibility that the fine tolerance given to this piston would in itself cause serious overheating, if the temperature of the piston was raised unduly by one or more of the faults enumerated in (2), (3), (4), and (5) above.

The existence of a "hot spot" in No. 2 port outer cylinder having been accepted by the Court, the theory advanced for the presence of ignitable material in the crankchamber must now be considered. That such material did exist, and was ignited, resulting in ultra rapid combustion which brought about a mild explosion, must be accepted. A reference was made in the course of the evidence to the experiments conducted by H. Bara, Engineer-in-Chief of the French Navy, directed to establishing (a) that the crankcase of a diesel engine operating normally is able to propagate flame, (b) that the origin of the ignition of crankcase contents can be traced to serious overheating of some moving part situated inside the crankcase. .

Turning to the evidence of the Chemical and Mechanical Engineering Experts the following deductions can be made :

(1) That the piston of port outer No. 2 engine had reached an average temperature of at least 500°F., with surface temperatures at certain areas of the piston well above 500°F., and in the vicinity of 1,500°F., as a maximum.

(2) According to the evidence this piston overheated within a period not exceeding ten minutes.

(3) That no previous overheating of pistons occurred under such conditions as the case under review, as to time under full speed, and revolutions per minute attained.

The following observations are made in reference to the explosion. The first is closely allied to the findings of Bara and other Investigators, and is supported by the evidence given at the Inquiry. The analysis of the contents of the crankcases in engines similar to those fitted in the "Reina del Pacifico", has revealed an oxygen content something of the order of 20%. Under normal working conditions crankcases contain no combustible gas. The above analysis is taken when the atmosphere of the crankcase is composed of lubricating oil in an atomised state, the fine particles being surrounded by air, and so forming a mist. This mist is due to the turbulence and general agitation set up in the crankcase by the rapid gyrating and reciprocating of the parts of the engine. This condition is intensified by the severe surging effect of the trunk pistons. Such an oil mist can be ignited by a "hot spot" of sufficient intensity to reach the spontaneous ignition temperature of the lubricating oil. The spontaneous ignition temperature of the oil in use was approximately 720°F. The oil mist was ignited by (1) hot body of the piston (most probable). (2) The burning lubricating oil leaving the piston bearings and cylinder liner (probable). (3) Vertical streaks of very high temperature surface metal on the skirt of the piston and walls of the cylinder liner (most probable).

It would appear from the evidence that when the mist became ignited, the burning would be consistent with the percentage of oxygen present in the crankcase, and would come under the term "rapid." The building up of pressure would be relatively slow and would not assume appreciable magnitude until the advent of fresh air, drawn through the mist by the extractor fans into the burning mixture, which would cause ultra-rapid combustion, and build-up of pressure taking the form of a mild to moderate explosion. This is supported by the fact that no damage was sustained by the engine apart from crankcase doors and covers, Most of the damage to these was caused by their striking objects external to the engines.

There is a further possibility based on certain evidence given at the inquiry. Expert knowledge suggests that whilst small particles of lubricating oil intimately mixed with the right proportion of air is highly inflammable, it cannot be detonated. As the evidence is that detonation in a mild form did occur in No. 2 port outer engine crankcase, it might well be that explosive gas existed, and that the lubricating oil particles had been evaporated to form, with the air, an explosive mixture. The evidence goes no further than that the hot piston ignited the gaseous contents of the crankcase. As no evidence was submitted to show how this gas was formed, the following theory is submitted by the Court:

The evidence leaves no doubt that No. 2 piston had reached an exceedingly high temperature, the estimate being between 500°F. and 1,500°F., the cause of which will be referred to later. As an interval of time elapsed between stopping the engine and the explosion occurring, the Court cannot completely accept the theory that the heat in the piston alone was sufficient to ignite the explosive gas, since it is apparent that when the explosion occurred. the piston would be appreciably cooler than when the engine was stopped four to five minutes earlier. The information was given in evidence by Mr. Bull, of the Ministry of Transport, who examined No. 2 port outer engine piston soon after the accident, that a most noticeable feature was that the inside of the piston skirt was in a dry condition, i.e., free from lubricating oil. This would suggest that during the four to five minutes which elapsed between the engine being stopped and the explosion occurring, the heat in the piston was sufficiently high to cause the oil on the inside of the piston skirt to evaporate. During the four to five minutes the engine was stopped, this lubricating oil would be gasified, and the major portion of this dense gas would remain inside the piston skirt as long as the piston remained stationary. To explode this gas it would be necessary to supply the requisite proportion of air and an igniting agent.

Whilst this gas was being produced in the stationary piston, the crankcase extraction fans continued to operate, and in this way air would find its way into the crankcase to mix with that already present, so that the air necessary to form an explosive mixture with the dense gas produced inside No. 2 piston skirt would he present.

The heat in the piston was produced by excessive friction

between its outer surface and the inner surface of the cylinder liner, so that the outer surface of the piston skirt would be at a higher temperature than the inside surface. If the temperature of the inside surface of the piston was sufficient to evaporate the oil on this surface, the oil on the outer surface would be even more readily evaporated.
Prior to the engine being stopped, the contact. surfaces of the piston and cylinder liner had become badly scored. During the time the engine was stopped, the working clearances would be still further reduced by the liner cooling more rapidly than the piston, so that when the engine was re-started a tearing action would take place, and it is reasonable to assume that sparks were produced immediately or very soon after the engine began to revolve on starting air. In considering this suggestion, it must be remembered that during the four to five minutes that the engine was inoperative, water at sea temperature continued to circulate through the cylinder jackets, and lubricating oil at about 130°F., continued to circulate the pistons. The effect of this would be that the cylinder liner would tend to close in on the overheated piston.

A further effect of setting the engine in motion would be for the dense gas inside the piston skirt to be thrown out into the crankcase, It would then mix with the requisite proportion of air with which it would be intimately mixed by the revolving crank and ignited by sparks produced in the manner suggested in the preceding paragraph.

During the Seventeen years that this ship has been in commission, five pistons had become overheated to the extent that on a number of occasions fire was produced inside the crankcase, It was observed that these instances of overheated pistons have always occurred after an overhaul which included the fitting of new pistons or cylinder liners, or both. It is the view of the Court that the cause of these trunk type pistons overheating is that the diametrical. working clearances were made to an absolute minimum in order to avoid Piston "slap" and consequent excessive noise.

The original diametrical working clearances on these 630 m/m diameter pistons was 0,63 m/m in way of the skirt, and this clearance has been adhered to throughout the life of the ship.

It is the view of the Court that during the "bedding in" process of new pistons and/or cylinder liners, the temperature of these parts rises above the normal working temperature, and with ,such fine clearances unless the lubrication is fully efficient and the contact surfaces free from dirt, and the cylinder not loaded above its rated power, the expansion of the piston reduces still further the working clearance and there is grave danger of the piston and cylinder liner becoming seriously overheated.

The time which elapsed between starting the engine on air and the explosion occurring could not be accurately ascertained the. time varying from immediately the starting lever was operated to two or three minutes after the operation of the starting lever. Had the explosion occurred simultaneously with the operation of the starting lever the foregoing theory as to the production of the igniting agent would not be disputed. An interval of two to three minutes, between starting the engine on air and the explosion occurring, however, does not rule out the theory that the sparks were produced as described.

The lack of lubricant on the contact surfaces and the absence of working clearance would be sufficient to continue the sparking for quite two or three minutes. If this interval of time between stopping and restarting the engine is accepted, the reason why the explosion did not occur until two to three minutes after the engine had been started up on compressed air could be due to the mixture of vapourized oil and air not being in such proportion as to be within the explosive range, until the revolving crank had brought about not only the correct proportions of air to gas, but the intimate mixing of the two.

The explosion in No. 2 port outer engine crankcase resulted in

the weakest door of that crankcase being blown outwar flame entering the engine room atmosphere. Another res the explosion in No. 2 crankchamber was to ignite the oil present throughout the length of the engine crankcase, an rapid increase in pressure resulting from this burning blew out other crankcase doors in this engine and liberat the engine-room the innumerable small particles of oil had not been consumed inside the crankcase.

There is evidence that the whole of the engine atmosphere below the level of the top platform, which is sit at the top of the cylinder jackets, was ignited and the entered the crankcases of the other three engines which, as is known, did not contain explosive gas, but air h impregnated with minute particles of lubricating oil, would be inflammable and be readily ignited by the entering the crankcases, The increase in pressure which result from this rapid burning process, in each of these engines was sufficient to blow out the crankcase doors of engines

As to how the flame from the port outer engine entered crankcases of the three other engines, the consensus of op was that the pressure produced in the engine room atmosp forced inward the unsupported lower ends of certain crank doors of each engine, and that when the inflammable va inside the crankcase was ignited the pressure blew the outward. The view that the lower ends of these doors had forced inward before the whole of each door had been bl outward is supported by marks which indicated that the l ends of the doors had made contact with the revolving cr The only other way in which flame could enter the crankcas these three engines would be through an annular opening m/m width between the cylinder liners and the crank diaphragm plates. Whilst this is unlikely, it is not impossible

Reverting to the opinion that the bottoms of the crankcase doors were forced inward by the pressure produce the engine room, it was not possible to ascertain, reasonable accuracy, the dynamic force required to distort doors in this way.

As there is an element of doubt as to how the flame from port outer engine reached the other three engines, we stro recommend that the matter be pursued by explosion specia who have not hitherto investigated the problem Which arisen on the "Reina del Pacifico."

The following modifications in design have been sugge by witnesses with a view to preventing recurrence of crank explosions in this ship:-

(1) The introduction of baffles between each indivi crank chamber.
(2) Complete separation of oil sump ventilating pipe
(3) Complete separation of crankcase vapour extra pipes.
(4) Re-design of crankcase doors to permit q inspection of the crankcases.
(5) Re-design of explosion discs on ends of crankcas
(6) Provision of carbon dioxide gas system a blanketing medium for the crankcase.
(7) Remote recording of cylinder liner temperatures.
(8) Increasing the diametrical working clearance of pistons.

In regard to the proposed modifications put forward in course of the Inquiry, we wish to place on record the follow observations. We agree that such alterations and additions w reduce the explosion hazard in the "Reina del Pacifico." In case of the proposal to sub-divide the crankcases, however feel that there would be practical difficulties. We are prepared to say that a moderately light baffle between chamber would have the effect of avoiding progressi increasing pressure in parts of the crankcase remote from point where an explosion takes place.

Although there is no evidence whatsoever to support the suggestion that the flame spread from the port outer engine to the other three engines by means of the crankcase vapour extraction pipes or the lubricating oil sump venting pipes, we are of the opinion that each of these pipes should be led to the atmosphere, i.e., that no two pipes should be joined.

If the diametrical working clearance of the pistons is increased as proposed at the Inquiry, and each piston is provided with four lead/bronze rings, two above and two below the gudgeon bearing, we are of the opinion that the main reason for pistons in this ship becoming hot will be removed. Nevertheless, it is our view that as there are so many cylinders and pistons (the number is 48), for the normal complement of engineers to attend, some means of registering an increase in temperature of the exposed part of the cylinder liners should be adopted. It would be sufficient to register an increase in temperature of the ahead thrust side of the cylinder liners, but the dials recording the temperatures should be grouped together and situated conveniently near to the engine control stations.

If the course of the flames and inflammable vapour released from the port outer engine was as stated at the Inquiry, it is probable that the pressure wave would not have reached the other three engines had the flames and vapour been able to take an upward direction soon after leaving the port outer engine.

They were prevented from doing so, to a large extent, by the plated platform on either side of each engine, and at. cylinder-head level. If these plates were replaced by open gratings any pressure wave released from an engine would be able to take an upward direction.

The alignment of the connecting rods of these trunk piston engines is of importance, if overheating of pistons is to be avoided. The alignment when the parts are at atmospheric temperatures appears to have had proper attention, but the Court is not convinced that due regard was given to the possible effect upon alignment of these parts when the unusually long crankshafts attain normal working temperature.

J. H. Campbell - JUDGE.

John Lamb
W. L. Nelson} ASSESSORS.
Ivor J. Grey

(Issued by the Ministry of Transport in London, in August, 1948)

Chapter 3: Sequence of Events Leading up to a Crankcase Explosion

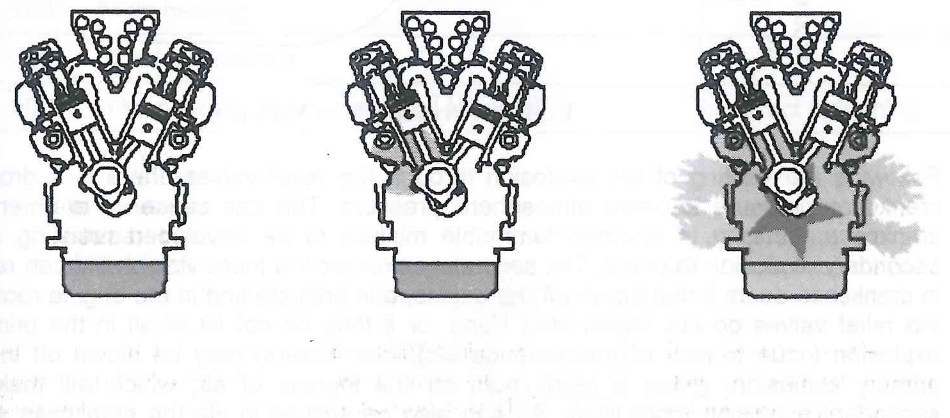
INTRODUCTION

Between 1990 and 2001 143 crankcase explosions were reported to Lloyds Register which have about 20% of the worlds shipping in its class, so if we use that as a factor, we can estimate the total reported incidents were 715 in 11 years or about 65 a year. Don't forget that these are reportable incidents, i.e. those where the damage sustained has warranted a major repair or has resulted in injury. Minor explosions may have gone unreported, and it is possible that the actual number of incidents is more than double those reported. - maybe 3 a week!!

Of those incidents reported to Lloyds, 21 explosions happened in two stroke marine diesel engines and 122 in four stroke marine diesel engines. But this doesn't mean that four stroke engines are more likely to have an explosion; there are 7 times as many four stroke engines at risk than two stroke engines.

SEQUENCE OF EVENTS LEADING UP TO A CRANKCASE EXPLOSION

For an explosion to occur there must be a source of air (oxygen), fuel and ignition. Oxygen is present in the crankcase, but the lubricating oil splashing around in the crankcase is in too large droplets to start burning at the speed needed to cause an explosion, and the oil/air concentration is too weak.



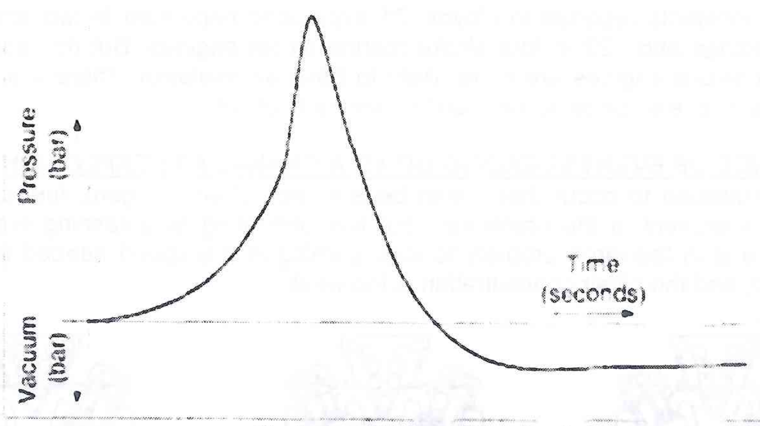
If, however a mechanical fault develops with the consequent rubbing of moving parts, then a hot spot will occur. This could happen in the crankcase, chaincase, or camcase. When the temperature of the hot spot reaches 200°C the lubricating oil splashing on to this hot spot vaporises. The vapour then circulates to a cooler part of the crankcase where it condenses into a white oil mist. The oil droplets in this oil mist are very small - 5 to 10 microns in diameter. When the concentration of oil mist reaches 50mg/l (about 13% oil mist - air ratio), it is at its lower explosive limit. If this oil mist is now ignited by the hot spot - and tests have shown that it is necessary for a temperature of about 850°C to ignite oil mist in a crankcase under operating conditions - then an explosion will occur.

Although the most common cause of a localised hotspot is due to friction, it is not the only cause of a crankcase explosion. A cracked piston crown, blowby or an external fire have caused crankcase explosions in the past.

PRIMARY AND SECONDARY CRANKCASE EXPLOSIONS

Severity of explosions vary between a puff which may lift a relief valve to a violent explosions which causes major damage and may injure personnel and cause a fire. Evidence indicates that the longer the combustion path, the more violent the explosion. This has become an area of concern with the large two strokes of today which may have a crankcase volume of 500m^3 +.

When an explosion occurs a flame front travels down the crankcase with a pressure wave in front of it. The turbulence caused by moving engine components causing churning and mixing of vapours increase the speed of the flame front and its area, which contribute to the increase in pressure. Turbulence caused by venting of the pressure through relief valves can also influence the explosion.



Explosion Pressure/Time Variation

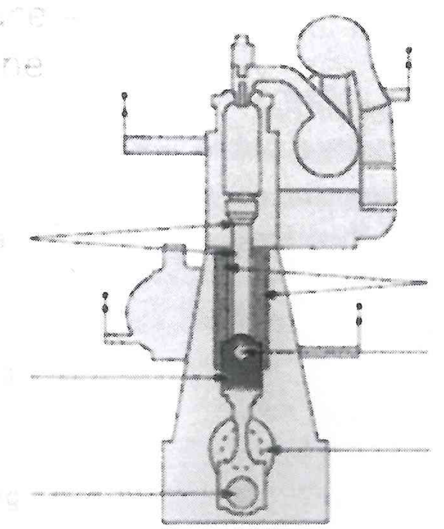
Following the venting of the explosion through the relief valves, there is a drop in crankcase pressure to below atmospheric pressure. This can cause air to enter the crankcase resulting in another flammable mixture to be developed resulting in a secondary explosion to occur. The secondary explosion is more violent and can result in crankcase doors being blown off the engine, and fires starting in the engine room. If the relief valves do not reseal after lifting, or if they do not lift at all in the primary explosion (due to lack of maintenance etc), then door(s) may be blown off in the primary explosion, giving a ready path for the ingress of air, which will make a secondary explosion more likely. Air can also be sucked in via the crankcase vent, although rules state that this must be as small as practicable and new installations must have a non return valve fitted.

If a primary explosion occurs, the pressure wave may send a large amount of oil mist out into the engine room. Although the flame arrestors on the relief valves should prevent ignition of this oil mist by the flame front, the mist will be sucked up towards the turbocharger where it may be ignited by an unlagged hot exhaust manifold. This ignition of oil mist can cause severe damage to plant and personnel.

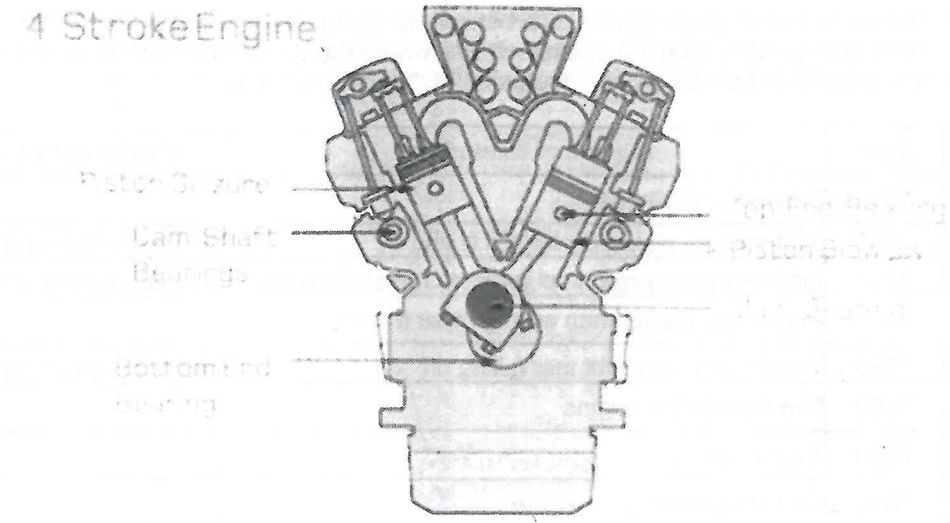
CAUSES OF CRANKCASE EXPLOSIONS

The table below gives details of a number of accidents which have occurred since 1995 to large slow speed 2 stroke engines where the cause is known. In a number of cases death or serious injury to members of the crew occurred.

Year	Cause of Explosion	Cause of Failure
1995	Bearing in PTO gearbox	
1996	Inlet pipe for piston cooling oil falling off	Incorrect tightening
1997	Incorrect spring mounted in piston rod stuffing box	Unauthorised spare part
1997	Piston rod interference with cylinder frame	
1999	Weight on chain tightener falling off	Incorrect tightening
1999	Fire outside the engine	
2000	Main bearing	
2000	Camshaft bearing	
2000	Incorrect shaft in camshaft drive	Unauthorised spare part
2001	Crankshaft failure	
2001	Piston crown failure	
2001	Main bearing	
2001	Crankpin bearing	
2001	Inlet pipe for piston cooling oil falling off	Incorrect tightening



Areas of Failure – 4 Stroke Engine



Inspection and Testing of Valves in Service.

Explosion relief valves are to be periodically inspected visually. The valve should be inspected for damage, deformation, leakage and loose fittings on a monthly basis. Particular attention should be paid to the condition of the flame arrester to ensure that it has not become choked. It is vitally important that the flame arrester is in good condition. A damaged flame arrestor will render it useless, and will result in the ignition of the oil mist outside the crankcase.

The O-ring used for sealing the valve is subject to hardening and should be renewed within 5 years of delivery. Only manufacturers supplied spare O-rings should be used in the renewal process.

Crankcase relief valves are subject to survey. This usually entails stripping the valve for inspection, and renewing the O ring seal. After reassembly the valve is tested using a spring balance to measure the force needed to open the valve.

Hoerbiger recommend testing their valves using a pressure rig to ensure that the valve opens and seals again after opening: Their instructions state:

Air pressure testing of explosion relief valve is to be carried out using a suitable test rig assembly. The valve seat must be clean before lowering onto the rig. With the valve bolted to the test rig open the air supply ball valve connection to atmosphere and record the gauge pressure. Close the ball valve.

Connect a clean 3 -10 bar air supply to the test rig and open the ball valve. Audible chattering of the valve plate opening/closing will be observed. Close the ball valve and observe the gauge pressure. A pressure indication of between 40 and 60 mbar should be held for one minute to demonstrate the valve tightness.

If the pressure is not held for one minute, the valve should be dismantled and the rubber O ring renewed.

Chapter 5: Oil Mist Detectors

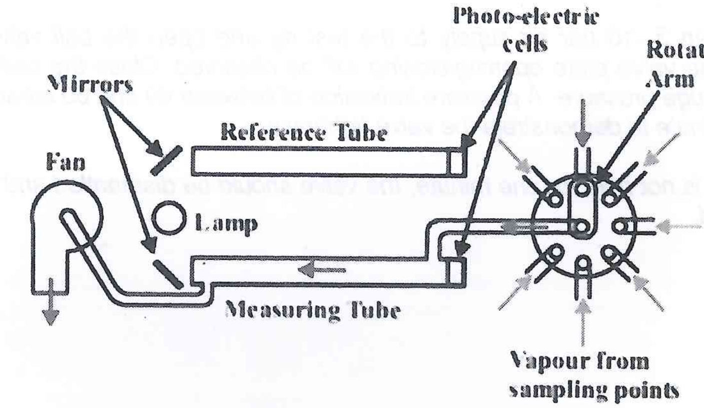
DETECTION OF OIL MIST

Oil mist detectors must conform with IACS Unified Requirement M67
They must alarm at 5% LEL or 2.5mg/l
The oil mist monitoring arrangements are to be capable of detecting oil mist in air concentrations of between 0 and 10% of the lower explosive limit (LEL) or between 0 and a percentage corresponding to a level not less than twice the maximum oil mist concentration alarm set point.

There are two methods of oil mist detection, **Obscuration** and **Light Scatter**. The earlier forms of oil mist detector used an obscuration type detector, the most well known type being the early Graviners. **Schaller** use obscuration in their **Visatron** range of OMDs. Light scatter is a modern method of oil mist detection used by **QMI** and the Latest **Graviner Mk6**

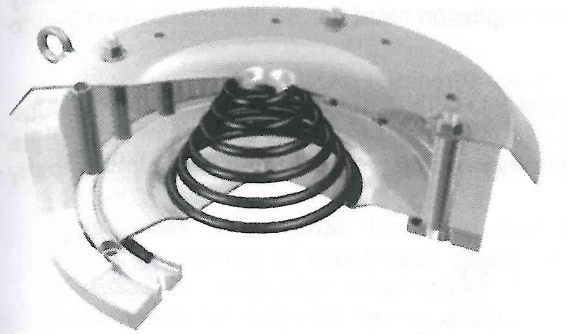
Under IACS UMS Regulations M36, engines of over 2250kW or 300mm bore operating in an unmanned machinery space must be fitted with a method of oil mist detection.

OBSCURATION



The older type of obscuration type of detector consists of two parallel tubes of equal size, each having a photoelectric cell at one end which generates an electric current directly proportional to the intensity of the light falling on its surface. Lenses are fitted to seal the ends of each tube but allow light to pass. Two identical beams of light from a common lamp are reflected by mirrors to pass along the tubes onto the cells which are then in electrical balance.

The samples drawn from the crankcase are drawn in turn along the measuring tube by means of a selector valve. If a concentration of oil mist is present in the sample, the light will be obscured in the measuring tube: electrical balance between the two cells will be disturbed and an alarm will be operated.

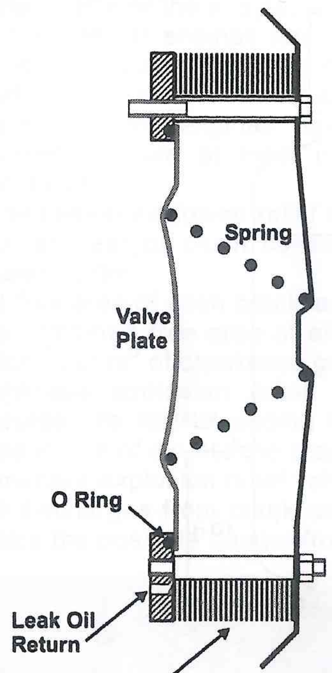


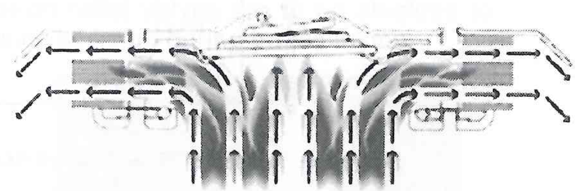
The other type of relief valve in common use especially on large two stroke engines is the Hoerbiger type.

The main difference between this and the BICERI relief valve is that the flame arrestor is on the outside, where it is easily visible for signs of damage.

It should be emphasized that the flame arrestor should be undamaged (especially have no holes through it) and be clean and free from paint or dirt.

The operation of the valve is similar to that of the BICERI door, except the spring, being of conical design allows the door to open further, thus flattening the spring.





HOERBIGER TYPE CRANKCASE RELIEF VALVE

FLAME ARRESTORS AND SHIELDING

The intention of the requirement to shield the valve discharge was to reduce the possible danger to personnel from flame emission. However explosion testing has shown that whilst a flame arrestor will work satisfactorily when shielding is not fitted, when shielding is fitted, the energy from the discharge is focused in one direction, and there will be an emission of flames during an explosion. The fitting of shielding also reduces the effective outflow area of a valve.
Since July 2002 it has been a Lloyds Register and IACS rule requirement to fit flame arrestors and to test the relief valve with any proposed shielding to be fitted to the valve when installed on the engine.

Under IACS Unified Requirement M66, all relief valves/flame arrestors must be tested using methane gas and air mixture to demonstrate that classification society requirements are satisfied for crankcase explosion relief valves intended to be fitted to engines and gear cases.

The relief valve is bolted to a test vessel which is then filled with a methane/air mixture. A plastic bag 0.05mm thick surrounds the relief valve. The mixture is ignited and the effectiveness of the flame arrestor ascertained. The test is then repeated immediately after without the bag.

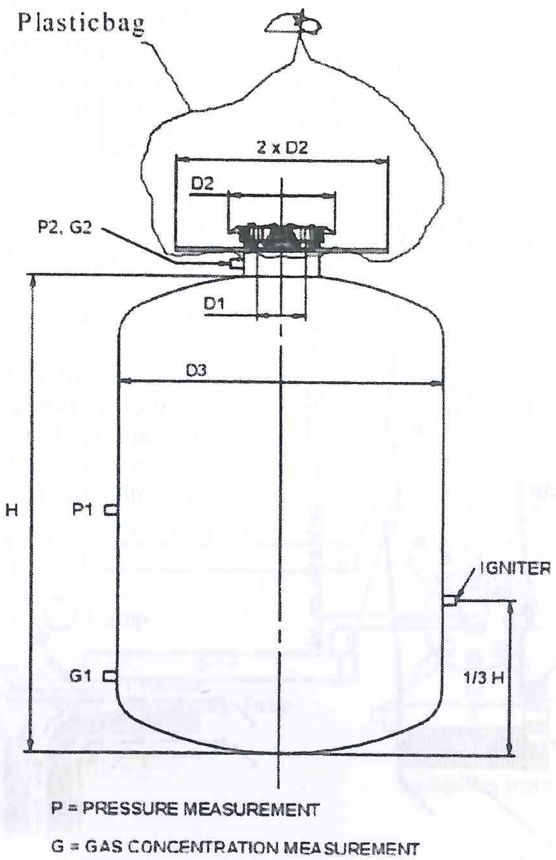


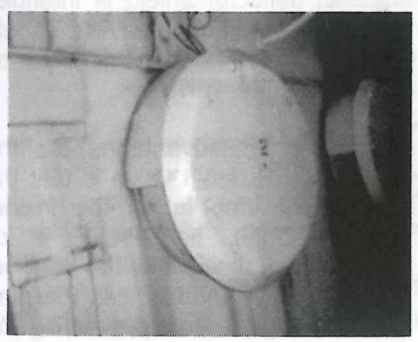
Fig. 5 Test vessel

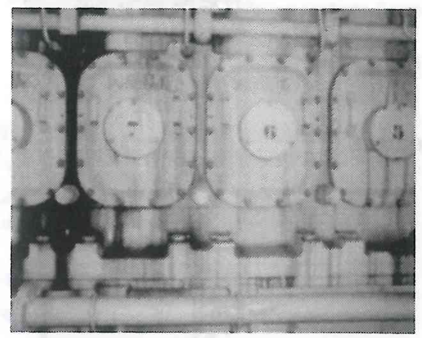
Chapter 4: Crankcase Relief Valves

REGULATIONS

The provision of crankcase explosion relief valves for marine diesel engines is an international safety requirement stemming from two sources, IMO SOLAS Chapter II-1 and IACS Unified Requirements M9 and M10. Summarising, the requirements that relate to explosion relief valves are:

i. Crankcases for engines having a cylinder bore greater than 200mm and above or having a crankcase gross volume exceeding $0.6m^3$ are to be provided with explosion relief valves.
ii. Crankcases for engines having a cylinder bore not exceeding 250mm are to be provided with at least one explosion relief valve at each end of the engine. If an engine has more than eight crank throws an additional valve is be fitted at the middle of the engine.
iii. Crankcases for engines having a cylinder bore greater than 250mm but not exceeding 300mm are required to have at least one crankcase explosion relief valve at each alternate crankthrow with a minimum of two valves.
iv. Crankcases for engines having a cylinder bore exceeding 300mm are required to have at least one crankcase explosion relief valve at each crankthrow.
v. An additional explosion relief valve is required on separate crankcase spaces such as gear or chain cases for camshaft drives where the gross volume exceeds $0.6m^3$
vi. The free area of each crankcase relief valve is to be not less than $45cm^2$.
vii. The combined free area of all crankcase relief valves is to be not less than $115cm^2$ per m^3 of crankcase gross volume.
viii. Crankcase explosion relief valves are required to open quickly at an overpressure not exceeding 0.2bar in the crankcase and close quickly to avoid inrush of air into the crankcase following an explosion.
ix. Crankcase explosion relief valves are required to be of an approved type.
x. The discharges from crankcase explosion relief valves are to be shielded to reduce the possible danger from emission of flame during an explosion.





Crankcase Relief Valves: Sulzer RTA (left) Diahatsu Med. Speed (right)

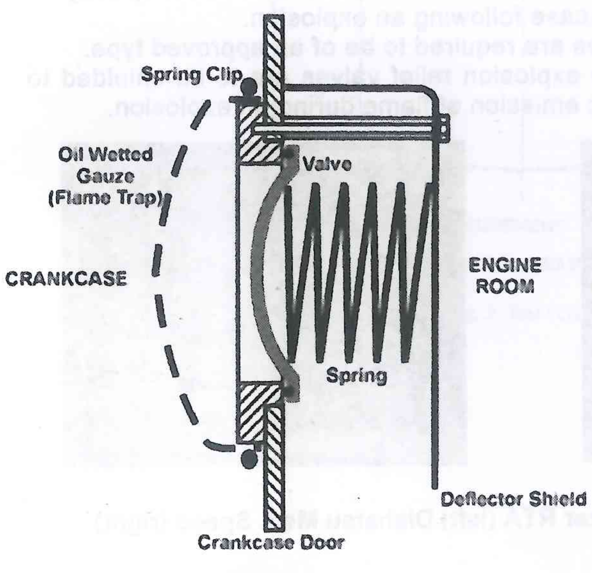
The size of a crankcase relief valve relates to the free area of the open valve. For a circular valve this is determined by the expression πDH, where **D** is the diameter of the valve and **H** is the valve lift at a pressure not exceeding 0.2 bar. The diameter of relief valves ranges between 87mm and 705mm, with associated free areas of 59cm^2 and 3905cm^2.

The ratio of 115cm^2 free area per m^3 of crankcase volume is designed to limit the crankcase pressure rise to 1.3 bar. It is thought by some experts in the field, that in the very large 2 stroke engines in use today, that the present rules relating to the size of the relief valves allow the pressure in the crankcase to exceed this 1.3 bar limit, with a corresponding rise in the incidents of damage and injury to personnel.

Because the valve must be fully open at 0.2 bar, it will start to open at pressures well below this (as the spring is compressed, the force required to open the valve increases). Typical opening pressure is about 0.05 bar.

It is important that the valve closes smartly once the excessive pressure has been relieved to minimise the mass of air drawn in after a primary explosion. Tests on large doors have shown that the weight of the valve disk and unevenness of spring load on the disk may cause jamming when the door is mounted in the vertical position. It may be prudent to fit two smaller relief valves instead of one large valve. However, it is thought that this could cause extra turbulence during an explosion, which in turn increases the violence of the explosion. This is currently under investigation, but no results are yet available.

There are two main types of relief valve in use.
The older type is that developed by B.I.C.E.R.I. – The British Internal Combustion Engineering Research Institute.

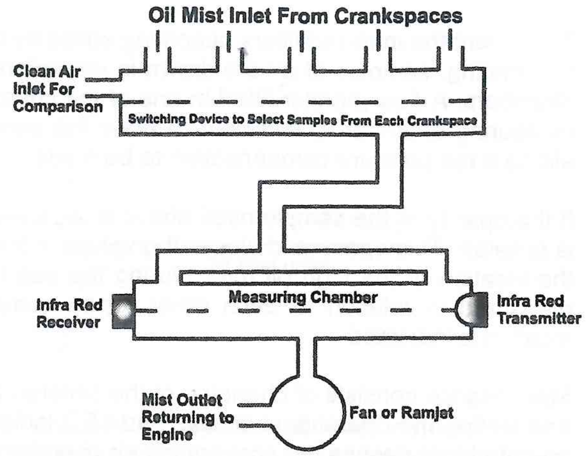


Explosion Relief Door

The device consists of a light low inertia aluminium disc, sealing by means of an O ring, held in place by a light spring. A deflector shield is intended as a protection device, should a person be standing close by when the valve lifts.

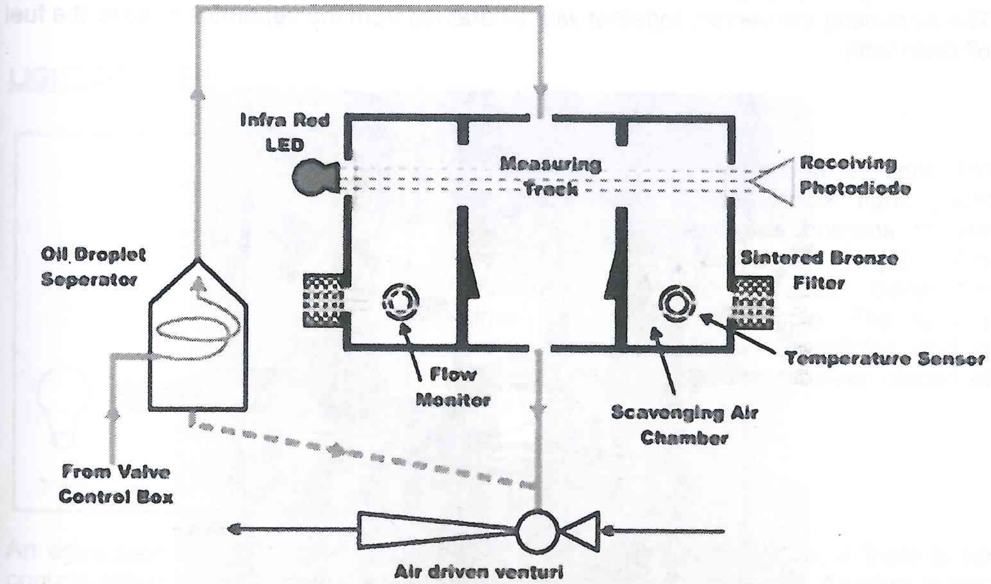
One disadvantage of the BICERI relief valve is that the flame trap is on the inside. Although, because it is oil wetted, it will take the heat out of any burning gas passing through it, thus preventing the passage of a flame, poor maintenance can result in it becoming choked with sludge deposits, thus rendering the relief valve inoperative.

Later types of obscuration detector use an infra red light source at one end of the measuring chamber and an infra red receiver at the other end.



Photoelectric Tube Unit

THE SHALLER VISITRON OIL MIST DETECTOR



Operating Principle of Schaller Visatron Type Oil Mist Detector

A venturi driven by low pressure compressed air draws the samples from the crankcase compartments. The samples from the crankcase pass through an oil droplet separator which removes any droplets of oil by centrifugal force before passing across the measuring track of the oil mist detector. A receiving photodiode converts the intensity of an infra red beam of light from an infra red light emitting diode into an electrical signal which is fed to the electronic evaluation unit. As the opacity of the sample increases so the intensity of the infra red beam being measured by the receiving photodiode will decrease. This will produce a lower electrical signal.

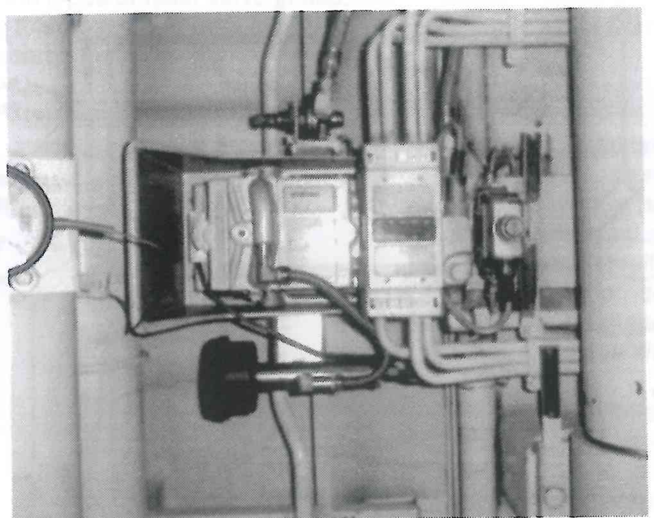
To prevent the infra red filters becoming soiled by the crankcase samples and affecting the reading, samples of air are drawn in via sintered bronze filters into scavenging air chambers. A flow monitor fitted in one of the chambers verifies that the vacuum in the measuring chamber is sufficient to draw the sample through. A temperature sensor allows a temperature compensation to be made.

If the opacity of the sample rises above a set level, an alarm sounds and a search run is initiated. This opens and closes the valves in the control box in sequence to indicate the location of the high oil mist. During the search run, groups of compartments are measured in relation to each other. Upon completion of the search run, the fault location is indicated.

Maintenance consists of changing of the sintered bronze filters if they become blocked and testing the operational circuits and LED indicators using the test button. it should be noted that despite the scavenging air chambers, soiling of the infra red filters is still possible, especially with age. The equipment will interpret this as an increase in opacity. This can be compensated for by adjusting the opacity reference point to a maximum of 70% opacity.

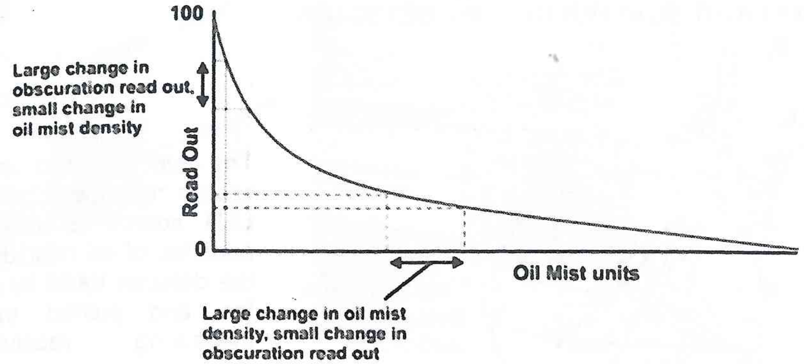
Physical testing of the equipment can be carried out by introducing a test smoke into one of the sample tubes.

The air leaving the venturi, together with oil drained from the separator is led to the fuel oil drain tank.



Schaler Oil Mist Detector

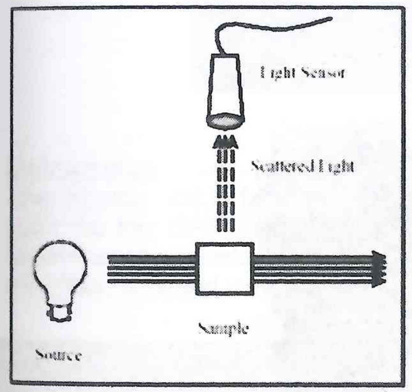
The obscuration type of detector has the advantage of simplicity of operation. The disadvantages are that comparatively long runs of pipework from sampling points to detector are required, only one sample point is measured at a time, and regular maintenance is required if false alarms are to be avoided. The actual monitor is located close to the engine and therefore if a high oil mist alarm is activated, the natural instinct to check the detector should be resisted.



Another major disadvantage of the obscuration type of oil mist detector is the relationship between oil mist density and the read out. As can be seen from the graph, the relationship is non linear. When there is no oil mist, then the read out is 100%. As the oil mist increases then a large change in the density of the oil mist is needed to change the obscuration read out by a small amount. Because of this non linear relationship, the monitor can not be graduated in units such as mg/l. It looks for deviation to trigger an alarm.

Manufacturers claim that it is set to alarm at an oil mist concentration of 2mg/l.

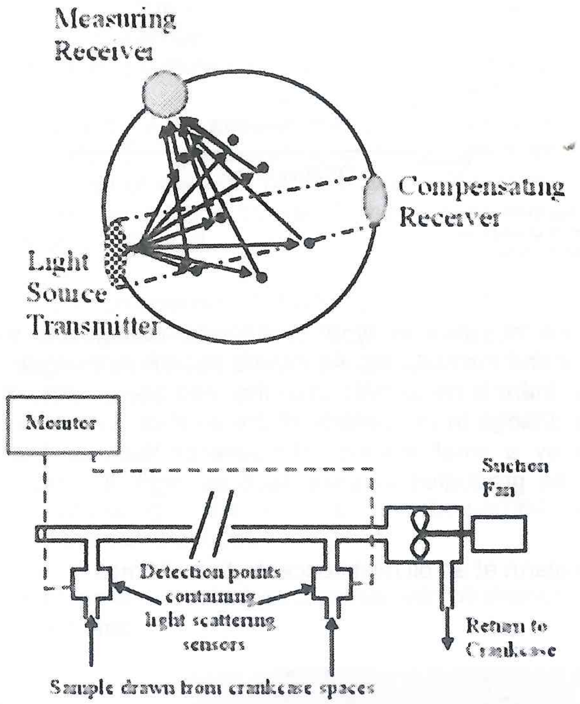
LIGHT SCATTER



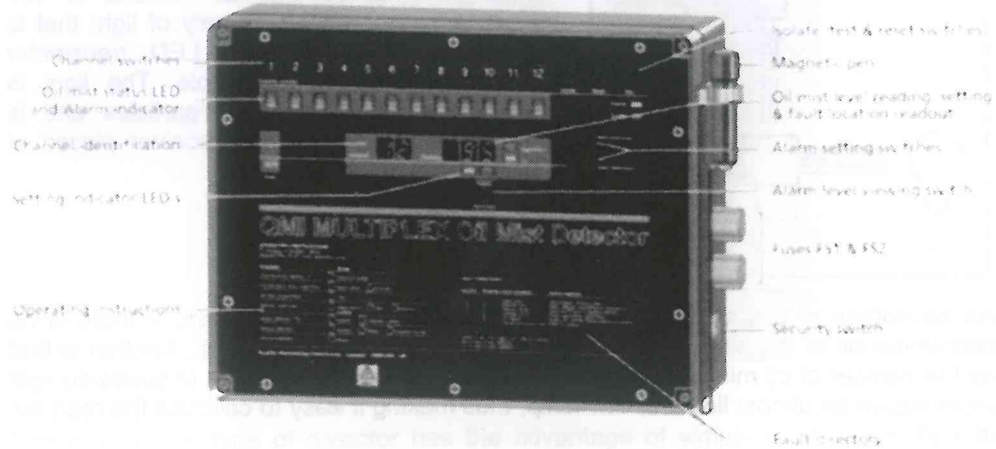
Light scatter is the interaction of light and matter. All materials will scatter light. (Light scatter is what enables us humans to see objects.) It is composed largely of light that is reflected or refracted. An LED transmitter shines light through a sample. The light is scattered from the oil mist particles and is picked up by a measuring receiver placed at 90° to the light source.

An advantage of this system is that it is measuring from a true zero. If there is no contamination of the sample (ie no oil mist) then there is no read out. Another is that as the number of oil mist droplets increase in the sample the amount of scattered light increases in an almost linear relationship, thus making it easy to calibrate the read out in mg/l.

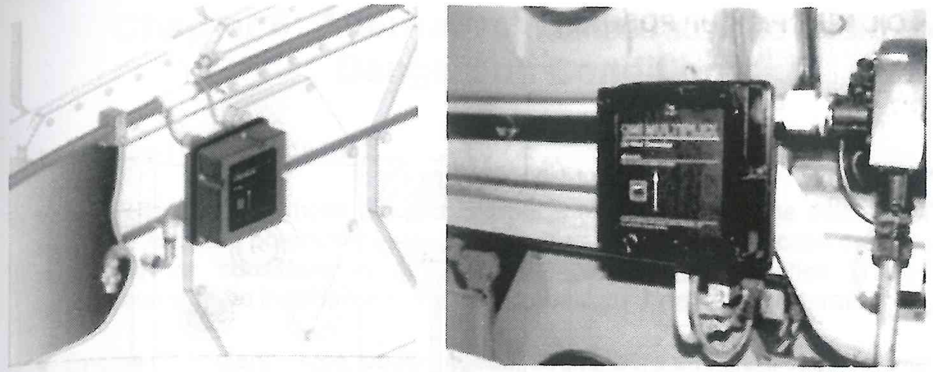
THE QMI LIGHT SCATTER OIL MIST DETECTOR



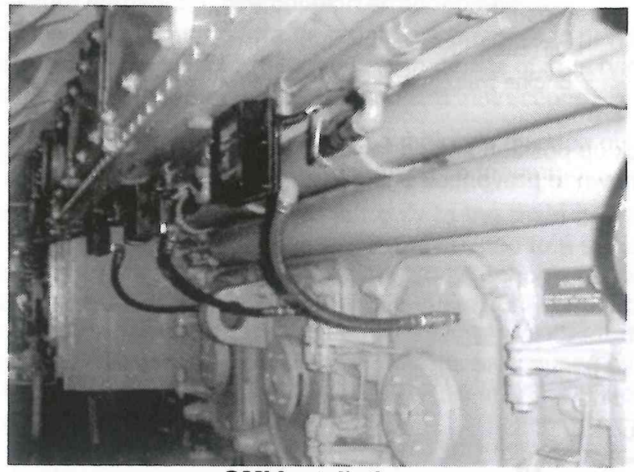
The detector works on the light scatter principle. Light from an LED source is reflected off particles of oil mist drawn into the detector head by a suction fan and picked up by a measuring receiver. A compensating receiver opposite the light source detects a fall off in intensity of the light due to a dirty lens and compensates accordingly. The measured signal from this and other detectors (up to 12) is fed back twice per second to the central processing unit situated in the main control room.



QMI Central Processing Unit



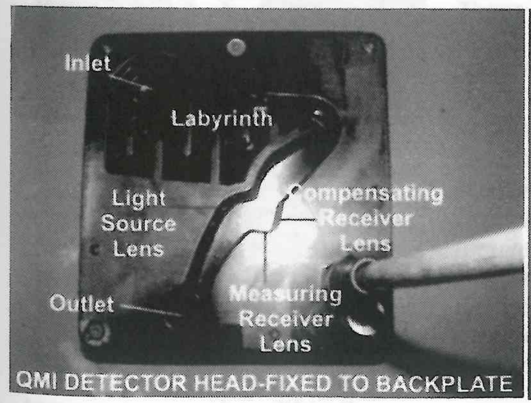
QMI Detector Heads



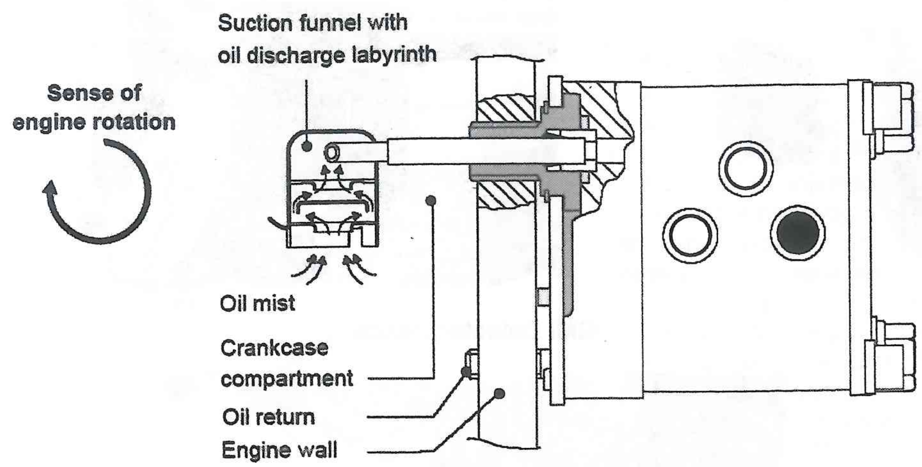
QMI Installation

Maintenance

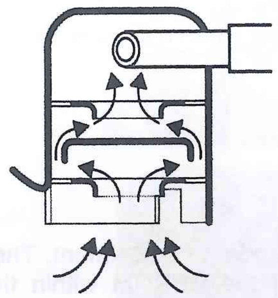
The detector unit is factory calibrated and therefore needs no adjustment. The alarm point on the CPU can only be altered by changing the EPROM within the unit. Maintenance consists of allowing the CPU to do a self test procedure and cleaning the inlet filter, lens and labyrinth on each of the detector units.



NOTE ON OIL MIST PICK UP POSITION



The oil mist pick up position should face downwards with the engine rotating ahead in the direction shown to prevent oil being thrown into the pickup.



The pick up should incorporate an oil separator which removes any oil droplets by changing the direction of the oil mist.

Chapter 6: Alternative Methods of Detecting a Dangerous Condition

IACS regs (10.21) state that:

Where alternative methods are provided for the prevention of the build-up of oil mist that may lead to a potentially explosive condition within the crankcase details are to be submitted for consideration of individual classification societies. The following information is to be included in the details to be submitted for consideration:

- Engine particulars – type, power, speed, stroke, bore and crankcase volume.
- Details of arrangements prevent the build up of potentially explosive conditions within the crankcase, e.g., bearing temperature monitoring, oil splash temperature, crankcase pressure monitoring, recirculation arrangements.
- Evidence to demonstrate that the arrangements are effective in preventing the build up of potentially explosive conditions together with details of in-service experience.
- Operating instructions and the maintenance and test instructions.

Bearing Temperature Monitors

Bearing temperature monitors can be fitted not only to the main bearings, but also to the bottom end bearing where the temperature is transmitted to a static pick up device from the revolving transmitter once per revolution. Some schools of thought claim that bearing temperature detection is not reliable enough using only one probe per bearing, and that the temperature should be monitored at several points around the bearing. However, although the majority of crankcase explosions are caused by bearing failure, there are other sources, esp. piston/ liner overheating in trunk piston engines, and monitoring of this should also be considered.

Splash Oil Monitoring

Picture showing the catcher for Splash Oil Monitoring. A temperature probe within the catcher measures the temperature

MAN B&W, as well as using oil mist detectors and main bearing temperature monitors, also offer a system on their medium speed 32/40 to 58/64 engines known as splash oil monitoring. The company's service letter states

These facilities (Bearing temperature monitors and OMD) cannot prevent damage, but they can considerably restrict its extent and consequences. To do this, they must respond rapidly and early to incipient damage, so as to prevent expensive components being affected to such an extent that complicated repairs or complete replacement become necessary. Experience shows that this last requirement is not always met satisfactorily. To eliminate this uncertainty, we have developed a monitoring system that reacts spontaneously and functions reliably - the "Splash Oil Monitoring System".

The system works by constantly monitoring to within very close limits the temperature of the oil being splashed about in each crankthrow space, and comparing it with the temperatures in the other spaces. It will alarm when either maximum value has been exceeded or when a maximum permissible deviation has been exceeded, generating an engine stop signal if the permissible operating values have been exceeded.

Intensive experiments have shown that the "Splash Oil Monitoring System" responds even to a slight increase in the local lube-oil temperature, due to bearing damage or seizures. This ensures an exceptional level of protection against malfunctions, and proven prevention of severe and expensive damage, by stopping the engine immediately.

Of course an engineer may be alerted to a potentially dangerous situation when on watch that all is not well using his senses of sight hearing and smell.

Chapter 7: Reducing Risks of an Explosion

Crankcase explosions which are violent and cause the maximum damage are thought to be caused by turbulent flame fronts having enough room to accelerate to sonic velocities thus raising the crankcase pressure high enough in a very short time to have serious consequences. The large two stroke engines currently being built have a crankcase volume approaching 600m^3 with overall lengths of 23 metres. Experience has shown that in two similar engines having crankcase explosions, one where the ignition source was close to the centre of the engine caused minimum damage, whilst the other, where the ignition source was at one end of the engine suffered a severe explosion in which crankcase doors were blown off.

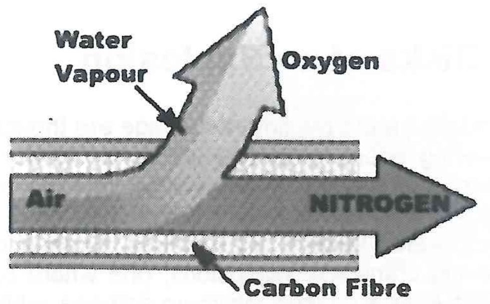
Therefore it should be possible to limit the consequences of an explosion which travels the length of the crankcase by subdividing the crankcase into separate compartments. However this might not be as simple as it first sounds. To begin with the subdividing bulkhead would have to be removable for maintenance purposes. And if it was removable it would have to be put back correctly to maintain adequate strength and subdivision. (People have enough trouble putting double skinned fuel pipes back correctly; how do you think they would manage with crankcase subdivisions?) Lastly, no engine builder is going to carry out this form of modification unless he is forced to by regulation. It would make his engine more costly and therefore less competitive.

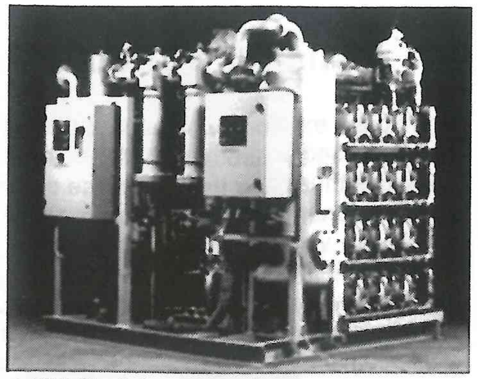
Once an oil mist has built up due to an overheated bearing etc, it may be possible to prevent ignition of the oil mist. MAN B&W have done experiments and taken out a patent on injecting water mist with a flame retardant chemical, the droplet size the same as that of the oil mist. The idea being that the energy from the flame would be absorbed by the fully mixed water droplets, and that the water droplets would also displace any oxygen present. Tests carried out have shown that while the water mist principle functions well at low revolutions, when engine revolutions are increased to normal working speed, the water mist is destroyed. This gives limited functionality, but could be used in conjunction with engine slow down on a high oil mist level being detected in the crankcase. They (MAN B&W) have also taken out a patent on a method of mixing a large amount of water with the lube oil in the event of an oil mist being detected, again using the principle of absorbing the heat energy using the high specific heat capacity of water.

There has been an idea put forward that once an oil mist has been detected of injecting an inert gas fast enough and at high enough pressure to displace the air in the crankcase. However, the risk here is that not only may the relief valves lift allowing the oil mist/air to be displaced into the engine room, but also any static charge releases with the gas, could ignite the oil mist/air mixture in the crankcase.

Another idea which has been under discussion is the possibility of inerting the crankcase using nitrogen on a permanent basis. This is an idea which the Royal Navy has been investigating for the past few years. The RN has suffered several oil mist explosions in their turbine gearboxes, which have caused extensive damage and loss of life.

The nitrogen would be provided by nitrogen generators: - these are already in use for inerting cargo tanks on chemical tankers




1 500 Nm3 Generator for chemical tanker

The nitrogen generator is based on hollow fibre membranes. The system works by passing compressed air through a bundle of hollow fibre, semi permeable membranes. The membranes divide the air into two streams; one is essentially nitrogen and the other oxygen plus carbon dioxide, moisture and other trace gases. The capacity of the membrane generators will be dependent on the required purity, air pressure feed and air temperature feed.

The crankcase cannot be completely filled with nitrogen, excluding all oxygen, because lubricating oil EP additives require oxygen to work. Instead about 5% oxygen will allow the lubricating oil to maintain its properties, but keep the oxygen content below that necessary for an explosion to take place.

The main arguments against inerting the crankcase are the cost and space required for a nitrogen generator, which for a large 2 stroke engine may take up the space of 2 container units, and it must be also borne in mind that before any maintenance can be carried out, the crankcase must be vented which will add to any down time. Nitrogen is only slightly lighter than air, and so venting the crankcase would have to be done using fans.

Chapter 8: Action To Be Taken In The Event Of A High Oil Mist Alarm

IACS regulations M35 & M36 state that a medium speed engine having a power of more than 2250 kW or a cylinder bore of more than 300 mm operating in an Unmanned Machinery Space must alarm and shut down on detection of a high oil mist concentration. One oil mist detector for each engine having two independent outputs for initiating the alarm and shut-down would satisfy the requirement for independence between alarm and shut-down system. This applies to either main or auxiliary engines.

The regulations also state that a slow speed engines having a power of more than 2250 kW or a cylinder bore of more than 300 mm operating in an Unmanned Machinery Space must be fitted with an alarm and slow down.

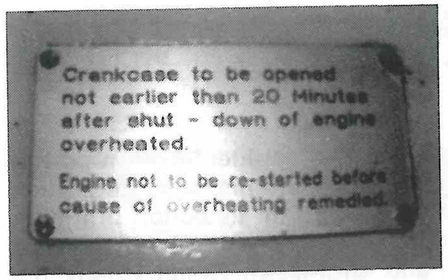
The action to be taken in the case of an alarm indicating a high oil mist in the crankcase, high bearing temperature or other indication of the possibility of an explosion occurring will depend on the age and type of installation, however the general rule is to STOP THE ENGINE!!

If the main propulsion is multi engine, either driving through clutches or diesel electric, then the engine may automatically shut down. If the engine is a single propulsion unit, e.g. a slow speed diesel engine, then the load must be reduced on the engine by at least 50% and the engine stopped as soon as it is safe to do so.

In the case of a bridge controlled 2 stroke engine on a UMS installation, then an alarm will inform the watchkeeper of impending slowdown. He will be given about 10 seconds to override the slowdown (in case of a potentially dangerous navigational situation), after which the load on the engine will be automatically reduced, bypassing any load reduction program. The engine should then be stopped as soon as safe to do so.

The reason for the immediate reduction is because the load on any overheated bearing which may be the cause of the generation of the oil mist and possible ignition source will be reduced, leading to a reduction in temperature, and thus lowering the risk of an explosion occurring. If a shaft alternator is connected to the switchboard, or if a turbo generator is running from the steam produced by the exhaust gas economiser, then the auxiliary generators should be run up and synchronised before slowing down. This will happen automatically in the case of an automated bridge control system described above.

If the engine is manually controlled from the engine room, then on an alarm situation occurring, the engine should be reduced immediately (ring telegraph to half ahead, reduce load, contact bridge, ask to stop), and then stopped as soon as possible. Call for assistance.



IACS 10.7 states: A warning notice is to be fitted either on the control stand or, preferably, on a crankcase door on each side of the engine. This warning notice is to specify that, whenever overheating is suspected within the crankcase, the crankcase doors or sight holes are not to be opened before a reasonable time, sufficient to permit adequate cooling after stopping the engine.

Once the engine has been stopped, keep the lube oil pumps running. Engage turning gear (if it can be approached without passing the crankcase). Evacuate personnel from engine room. Do not approach crankcase and do not remove crankcase doors for at least 20 minutes - a sudden inrush of air may cause an explosion.

The cause of the high oil mist alarm should be established before restarting the engine. If the cause is not immediately evident in the crankcase, check the camcase and chaincase. It should be noted that the cause of the overheating may not be in the area which showed a high oil mist concentration, due to the fact that the oil mist circulates through the crankcase.

Chapter 9: Crankcase Explosion - Case History 1

The story starts when the oil mist detector alarm for No 10 unit (a QMI type) on a 12 cylinder 2 stroke engine went off when the Chief Engineer was in the engine control room together with two of his staff.

The Chief Engineer thought that the alarm was due to a dirty detector head of the Oil Mist Detector and immediately sent one of the engineers down in the engine room to check the detector heads.

The Chief Engineer noticed that the oil mist level was increasing in crankthrow spaces 10 and 11, so he called the bridge and asked the bridge watchkeeper to reduce the revolutions of the engine.

5 minutes later he noticed that the oil mist level had increased further. It was at this point that he cancelled the load reduction programme and reduced the main engine revolutions to 75 rpm. At almost the same time a heavy explosion occurred in the engine room.

The engineer sent to clean the detector heads was standing close to a relief valve and was severely burnt.

A fire started in the engine room and purifier room caused by the shockwave from the explosion bursting an oil pipe. The fire was eventually extinguished by injecting CO_2. (after blocking off access doors that had been damaged in the explosion).

The Cause of the explosion was an over heated main bearing (no 13). Pieces of white metal were found in the crankcase and subsequent investigations established that some of the white metal had detached due to fatigue and poor bonding. As the metal detached, the load on the existing white metal increased, due to the now reduced area to support the crankshaft journal leading to breakdown in the lub oil film and wiping of the bearing. The crankshaft journal was now running on the bare steel backing.

When the explosion occurred, 4 doors on the "safe side" (the side without the explosion doors) blew off, starting the fire.

The vessel had to be towed to port for repairs which took 9 weeks.

The crankshaft journal was badly scored with extensive heat cracks. It was reduced in diameter and undersize bearings fitted.

It came to light in subsequent investigations that fragments of white metal had been found in the LO filters 2 weeks before the explosion. It was assumed that this was white metal from the guide shoes and no further action was taken.

It is easy to be wise after the event. However the oil mist detector is designed to protect, and a sophisticated set up like the QMI, which works on the light scatter principle, should be believed when it indicates a high oil mist. The first mistake was sending someone down to "clean the lenses". A crankcase explosion is a very serious occurrence. The engine room should be evacuated of all personnel not dealing with the situation from the control room.

A seemingly simple solution is "as soon as the alarm sounds, the engine should be stopped" But if this is the case why isn't the oil mist detector linked to the engine shut downs. On some ships it has been, but consider this: The ship may be manoeuvering in a risky situation, or there may be a shaft alternator in use.

The general procedure should be this. On a bridge control set up with the engine room in UMS mode, if alarm conditions are reached, the auxiliary generators should run up, load transferred, and the shaft alternator isolated. After a 10 second delay the engine should immediately power down to 50% full load, bypassing the load control programme. This reduction in load will reduce the likelihood of a crankcase explosion. The delay is to give the navigating officer time to override the slow down incase safety of the ship is at risk. Once the engine has slowed down it should be stopped as soon as it is safe to do so.

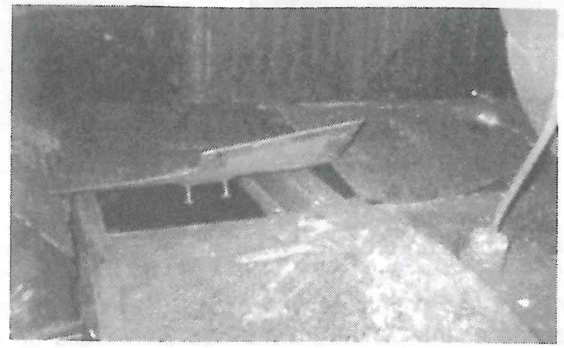
If the engine room is manned, and on bridge control, the watchkeeping engineer may choose to take control of the engine, reduce the load and stop as soon as it is safe to do so. Once the engine has stopped, the cooling and LO must be kept running, the turning gear should be engaged and the engine turned continuously. The engine room should be evacuated and time given (30 minutes) for the engine to cool down before opening up the crankcase for inspection.

Purifiers are fitted in separate rooms for a reason: Fire risk. The doors should be closed and dogged shut when not passing through them.

What about the white metal found in the filters? If, instead of making assumptions, the crankcase had been opened up they may have found the white metal in the vicinity of the damaged bearing.

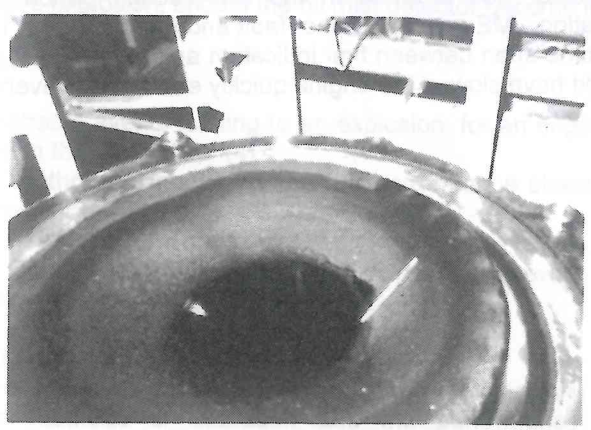
The fact that the oil mist detector alarmed for a unit not adjacent to the overheated bearing should be noted. The oil mist can circulate within the crankcase. This should be borne in mind when investigating the cause of a oil mist alarm.

Chapter 10: Crankcase Explosion - Case History 2



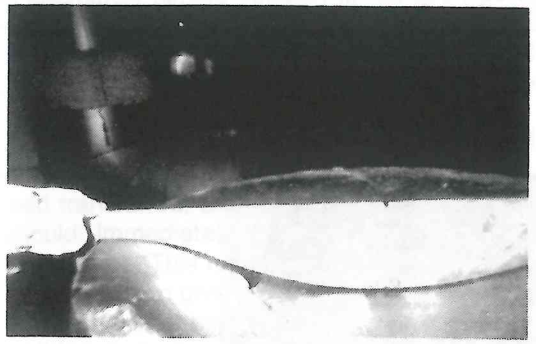
It was lucky that no one was injured in this crankcase explosion which lifted and buckled the engine room floor plating, blew the spare gear store door (which was 2 decks above the crankcase) through the frame , blew the oil mist detector off the engine, and caused extensive damage to lighting, vent trunking, paintwork and of course the crankcase relief valves and doors. The workshop door was buckled and de-laminated The control room door wire reinforced glass was blown through into the control room where shards of glass became embedded in adjacent woodwork.

On investigation the cause of the explosion was found to have been a hole in number 1 cylinder piston crown. Hot combustion gases had then been able to pass down the piston oil cooling return to the crankcase, forming and then igniting the oil mist.



The photograph shows the oil puddle on top of the piston, with a pen lying next to the crack for scaling.

Burn away was excessive on the piston crowns, with numbers 1 and 6 out of tolerance. (number 6 piston crown was also cracked)




The hole in number one piston crown had been caused by erosion due to poor fuel injection. All pistons were examined, and erosion damage and cracking found on all of them. The "burning" of the piston crowns giving them the telltale "elephant skin" surface. The underside of the piston crowns had 3mm build up of carbon deposits which would have compromised the cooling of the piston.

The fuel valves were examined: Blockages were found, the nozzles were not genuine spare parts and the push rod profiles were incorrect. The injectors were lifting at 30bar below the recommended pressure, and the nozzle holes had worn so that a jet of fuel was being injected rather than an atomised mist. Bearing damage had resulted from overloading due to poor combustion.

The crankcase relief valves were found to have been incorrectly assembled with defective flame arrestors. This caused them to jam open once they had lifted, and this may have contributed to a violent secondary explosion.

The oil mist detector had alarmed in the five second period between the first abnormality indication - ME governor minor fault and the ME cyl no.1 outlet high temp. Due to the short time-span between first indication and explosion, it would be doubtful if the monitor could have slowed the engine quickly enough to prevent an explosion.

Chapter 11: Chief Engineers Exam Questions

December 2005 Question 1

a. Describe the sequence events that may lead to a crankcase explosion.(6)
b. Describe an oil mist detector, explaining how it operates.(8)
c. Describe how an oil mist detector operation is checked for accuracy.(2)

October 2005 Question 5

a. Outline the probable events leading to a crankcase explosion, describing the likely consequences. (12)
b. As a Chief Engineer Officer state the standing orders that should be issued with regard to the risk of a crankcase explosion. (4)

December 2004 Question 5

a. Write the Chief Engineer Officers Standing Instructions for the duties of a watchkeeping engineer in the event of failure of the engine room monitoring and alarm system. (8)
b. Describe with the aid of sketches, an engine crankcase oil mist detector, explaining how it operates. (8)

October 2003 Question 6, March 2002 Question 7

a. Describe the operation of an oil mist detector fitted to a main engine. (5)
b. Explain how the location of the sampling heads may influence the performance of an oil mist detector. (5)
c. Explain the actions the Chief Engineer Officer must take to safeguard against the risk of crankcase explosion, should the oil mist detector become inoperative. (6)

July 2002 Question 5

a. Outline the probable events leading to an explosion, for an engine where a hotspot has developed in the crankcase. (4)
b. Compare the methods of operation and the performance characteristics of BOTH obscuration and light scatter type mist detectors. (6)
c. Explain how the severity of a crankcase explosion may be limited. (3)
d. Explain why stopping rather than slowing down in the event of an overheated crankshaft bearing is preferable. (3)

December 2000 Question 5

a. Outline the probable events leading to a crankcase explosion. (4)
b. Compare the methods of operation and the performance characteristics of obscuration and light scatter type mist detectors. (5)
c. Explain how the severity of a crankcase explosion may be limited. (3)
d. State the additional risk present when a crankcase explosion occurs. (1)
e. As Chief Engineer, state the standing orders you would issue with regard to the risk of a crankcase explosion (3)

Chapter 12: Second Engineers Exam Questions

October 2007 Question 3

With reference to crankcase explosions:
a. describe how a primary explosion occurs and how this may lead to a secondary explosion; (8)
b. explain why the action to be taken in the event of an oil mist detector alarm sounding. (8)

April 2006 Question 6, April 2005 Question 5

a. Describe a sequence of events leading to a crankcase explosion. (8)
b. Describe an oil mist detector explaining how it operates. (8)

December 2005 Question 5, October 2000 Question 5

With reference to crankcase explosions:
a.
(i) describe the operating principle of a crankcase mist detector; (5)
(ii) state THREE other indications of overheating or existence of conditions that might result in a crankcase explosion; (3)

b. state the course of action a Second Engineer Officer would take following the operation of the crankcase mist detector alarm. (8)

July 2005 Question 5

With reference to crankcase explosions:
a. describe, with the aid of a sketch, the crankcase explosion relief door for a diesel engine: (9)
b. explain how a crankcase explosion door functions in the event of an explosion; (4)
c. state a reason for the crankcase explosion relief doors opening, other than because of an explosion. (3)

March 2004 Question 7

a. Describe a crankcase oil mist detector system, including the sampling arrangement. (6)
b. Describe how an oil mist detector operates and sampling is controlled. (6)
c. State the maintenance required for the instrument described in Q7(a) and how testing is carried out. (4)

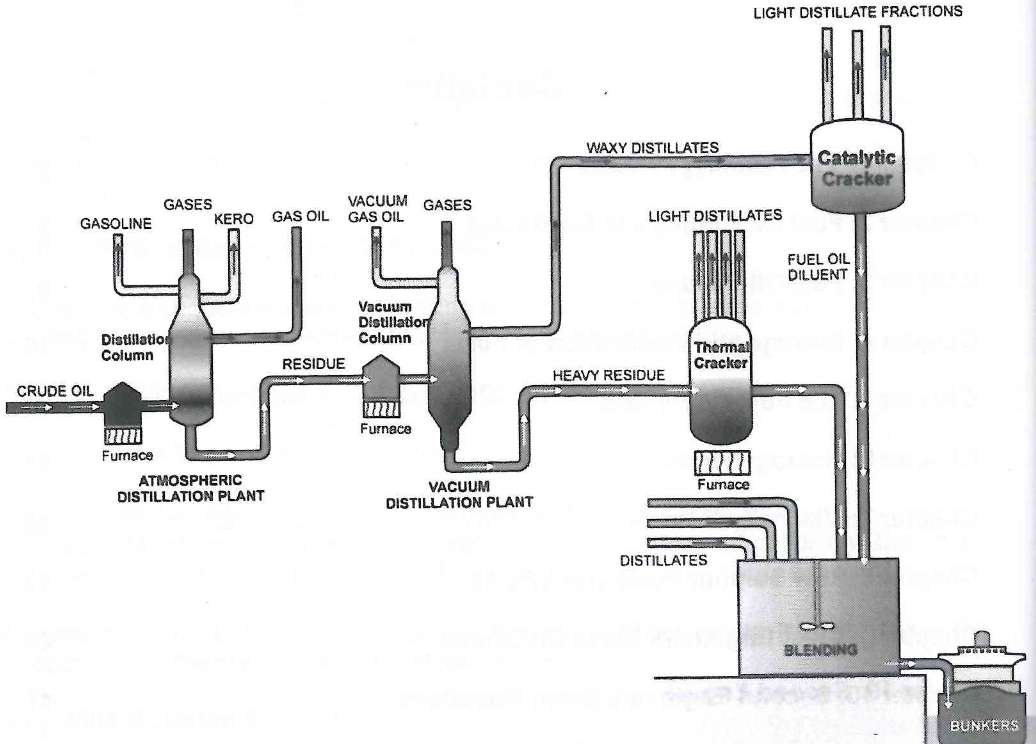
October 2003 Question 6

Describe the conditions in a crankcase which may:
a. create an explosive atmosphere; (7)
b. initiate a primary explosion; (5)
c. propagate a secondary explosion. (4)

Fuel Oil

Contents

Chapter 1: The Refining Process



The residues that make up residual fuels come from several refinery processes. In the first instance the fuel is heated and fed into the atmospheric distillation column where the lighter hydrocarbon molecules are distilled off as gases, gasoline, kerosene and gas oil, leaving about 50% of the crude as residue. Some of this residue is fed into the vacuum distillation column where it will be further processed to produce vacuum gas oil, waxy distillates and a residue. This residue can form part of the blend in marine bunkers or can be fed into thermal crackers. Here, under high pressure and temperature more light hydrocarbon molecules form and are removed as light distillates. Also the heavy gas oil from the vacuum distillation column may be subjected to catalytic cracking where as a result of chemical reaction yet more light hydrocarbon molecules are formed and separated into even lighter distillate fractions. The residues from all these processes can be blended with a variety of distillates to make up the specified marine bunkers.

Chapter 2: Fuel Oil Grades and

Bunker Specification

So that the ship owner can be sure of receiving the c… use in his engine, fuels are classified to an in… ISO8217:1996 There are four grades of distillate fuel… The distillate fuels are DMX, a pure distillate; DMA, g… DMC a blended diesel which contains up to 10% residual fue… Marine)

The residual fuels, which we are mainly concerned with are distinguished by the… (Residual Marine) classification. the two letters RM are followed by a letter A through to K which defines the characteristic of the fuel. A further two or three numbers gives the viscosity of the fuel in centistokes or mm^2/s at 50°C. The table which gives the specification of the different residual fuels is reproduced below.

Parameter	Unit	Limit	RMA 30	RMB 30	RMD 80	RME 180	RMF 180	RMG 380	RMH 380	RMK 380	RMH 700	RMK 700
Density at 15°C	kg/m^3	Max	960	975	980	991	991	991	991	1010	991	1010
Viscosity at 50°C	mm^2/s	Max	30	30	80	180	180	380	380	380	700	700
Water	% V/V	Max	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5
Micro Carbon Residue	% m/m	Max	10	10	14	15	20	18	22	22	22	22
Sulphur (1)	% m/m	Max	3.5	3.5	4.0	4.5	4.5	4.5	4.5	4.5	4.5	4.5
Ash	% m/m	Max	0.1	0.1	0.1	0.1	0.15	0.15	0.15	0.15	0.15	0.15
Vanadium	mg/kg	Max	150	150	350	200	500	300	600	600	600	600
Flash Point	°C	Min	60	60	60	60	60	60	60	60	60	60
Pour Point, Summer	°C	Max	6	24	30	30	30	30	30	30	30	30
Pour Point, Winter	°C	Max	0	24	30	30	30	30	30	30	30	30
Aluminium + Silicon	mg/kg	Max	80	80	80	80	80	80	80	80	80	80
Total sediment,Potential	% m/m	Max	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1
Zinc (2)	mg/kg	Max	15	15	15	15	15	15	15	15	15	15
Phosphorus(2)	mg/kg	Max	15	15	15	15	15	15	15	15	15	15
Calcium (2)	mg/kg	Max	30	30	30	30	30	30	30	30	30	30

(1) A sulphur limit of 1.5% m/m will apply in SOx Emission Control Areas designated by the International Maritime Organization.

(2) The Fuel shall be free of Used Lubrication Oil (ULO). A Fuel is considered to be free of ULO if one or more of the elements are below the limits. All three elements shall exceed the limits before deemed to contain ULO
Previously, bunkers were ordered with an IF number e.g. IF380. The letters IF were used by some oil companies to denote Intermediate bunker Fuel or InterFuel. The number after referred to the density in cSt at 50°C. Generally speaking an RME180 or an RMF25 correspond to an IF180 and an RMG380 or an RMH380 correspond to an IF380.Bunkers

Prior to arrival:

Although there is not a universal standard for bunkering procedures, several attempts have been made including The Singapore Bunker Procedure. Although not specifically mentioned in the ISM code, closer inspection will reveal that a bunkering operation involves most sections of the code, esp. section 7 relating to procedures for the prevention of pollution.

The Chief Engineer will hold a pre-bunkering meeting
Present: all those involved in bunkering operation and chief officer.
Agenda:

- Quantity and grades of fuel to be bunkered (Low sulphur RM High sulphur RM, MDO).
- Method of delivery (e.g Barge, stbd side).
- Order of filling tanks.
- Transfer of fuel prior to arrival so to avoid loading on top.
- Personnel responsibility and places of duty during operation.
- Details of SOPEP (Ships Oil Pollution Emergency Plan) for that port.

Copies of the above details to be posted at bunker station, ECR, bridge.

- Empty overflow tank and check operation of alarms.
- Check fuel lines earth bonding
- Check blanks and valves on manifold on opposite side to bunker station to be used.
- Check communication equipment between manifold and tank control board in ER/ECR

On Arrival in Bunker Port:

- Take final tank soundings and calculate free space to tanks 95% full
- Drip trays placed under manifold connections.
- Scuppers plugged.
- Oil spillage emergency kit is to hand.
- Foam fire Extinguishers placed adjacent to bunker station.
- No smoking signs erected and visible.
- Bridge hoist red bunkering flag.
- Record fuel meter reading (if fitted)

Once the barge is alongside, check with the barge master that the fuel on the barge according to the delivery sheet is of the grade and quantity ordered.

MARPOL Annex VI requires the following information to be recorded on Bunker delivery Note:

- Name and IMO number of receiving ship
- Bunkering Port
- Date of commencement of bunkering
- Name, address, and telephone number of marine fuel oil supplier
- Product name
- Quantity (metric tons)
- Density at 15°C (kg/m3)
- Sulphur content (% m/m)

- A declaration signed and certified by the fuel oil supplier's representative that the fuel supplied has a sulphur level below 4.5% and that the fuel is free from inorganic acid, does not include any added substance or chemical waste which either jeopardises the safety of ships, adversely affects the performance of the machinery, is harmful to personnel, or contributes overall to additional air pollution).
- Seal Number of the MARPOL Annex VI sample

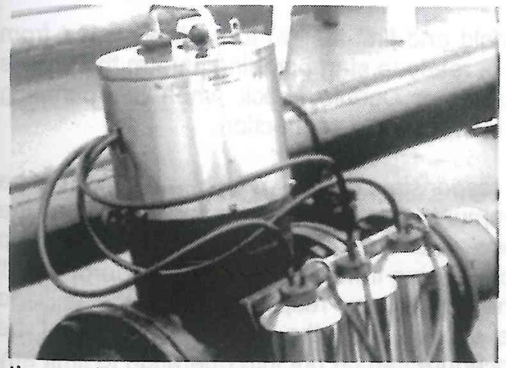
The Bunker Delivery Notes are to be kept on board and readily available for inspection at all times. They shall be retained for a period of three years after the fuel oil has been delivered on board.

The Chief Engineer or his deputy should dip the tanks on the barge before bunkers commence. They will be dipped again at the end of the transfer to check that the fuel quantity delivered tallies with that received on board. Because the fuel will most probably be heated, a correction factor may be applied which will correct the volume of the fuel to that at 15°C. If a meter is fitted on the barge discharge line, record the reading.

Before the line is connected, a sample should be taken on board and tested for compatibility. This is especially important if the fuel must be "loaded on top", that is put into a tank which already has fuel in it. Although blended fuels are stable as supplied, there is a small risk that some fuels when mixed together could produce asphaltic sediment or sludge.

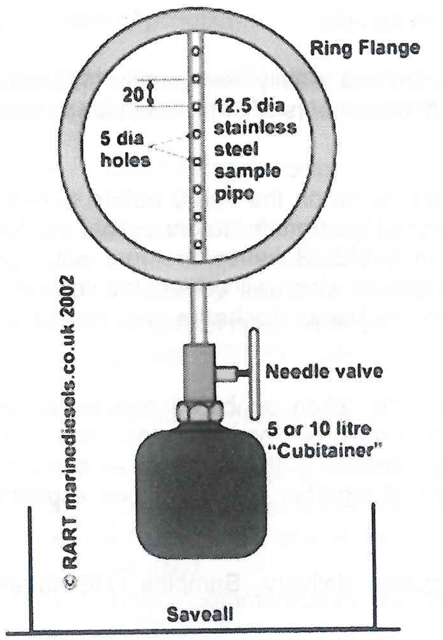
A representative sample must be obtained during delivery. Samples (750ml) are required for the following:

- Suppliers sample
- Ship sample in case of dispute over quality.
- Sample for sending to independent test agency (e.g.FOBAS)
- Sample for on board testing. (optional)
- MARPOL VI Sample



It is not a good idea to accept a sample in a sealed container, without knowing its source. The supplier will often use a flow proportional automatic continuous sampler, which, as its name suggests, will take a sample of the fuel as the fuel is pumped on board. Before bunkering commences, the sampler is set to the quantity of fuel to be delivered, the bottles connected and the sampler sealed. The Chief Engineer or his representative will be asked to witness the setting and sealing of the sampler. At the end of bunkers, the sample bottles are sealed, again witnessed, and signed. Two bottles must be given to the ship; One should be retained on board for at least 12 months in case of dispute over the bunker quality, when the sample can be sent for independent analysis, the other is the MARPOL sample which is to be sealed and signed by the supplier's representative and the master or officer in charge of the bunker operation on completion of bunkering operations, and retained under the ship's control until the fuel oil is substantially consumed, but in any case for a period of not less than 12 months from the time of delivery.

The vessel may want to carry out its own on board tests, or send a sample to an independent organisation like FOBAS for testing. In this case, the supplier may give the ship an additional sample. If not, or if the sampling of the fuel cannot be witnessed, then it may be necessary for the ship to take its own samples. The most practical and economical method of doing this is to employ a continuous drip sampler.



The sampler consists of a ring flange with a 12.5mm dia sample pipe spanning the diameter of the flange. 5mm dia sample holes facing the direction of flow are drilled at 20mm intervals. At the end of the sample pipe is a needle valve for controlling flow and a connection for attaching a semi transparent flexible collecting container.

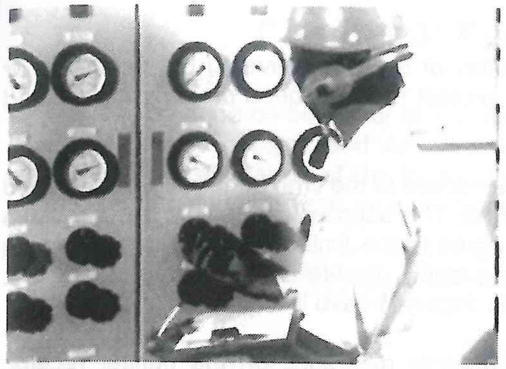
The normal place to position the sampler is between two flanges at the bunker station adjacent to the isolating valve.

The pumping rate, maximum delivery pressure, and means of communication will also be agreed with the bargemaster as will be signals for emergency stop.

Ensure all valves are open between manifold and first tank(s) to be filled, apart from the valve at the manifold. When the barge is ready to start pumping, inform the engine room and open the manifold filling connection. Double check when the pumping commences that there are no leaks around the manifold connection.

Whilst the fuel oil transfer is under way, a responsible person must be stationed at the manifold at all times. He must be in communication with the engine room and the bunker barge. Checking the filling hose and the water surrounding the ship periodically, he must tell the barge to stop pumping if necessary. In the case of a spill occurring it must be dealt with using the emergency response kit. If the spillage contaminates the water surrounding the vessel, then the port authority must be informed so that their oil spill response procedure can be put into operation to limit any damage caused. (SOPEP).

The tanks being filled must be monitored. As they approach 95% capacity then the next tank to be filled must be opened. Always open the valve of an empty tank before shutting the valve to the full one. As the end of bunkers approach, the pumping rate must be slowed down as the tanks are topped off. Once bunkers have finished the filling line may be "blown through" to ensure no oil remains in the line.



Tanks on board are then sounded and/or the fuel oil meter reading recorded, as are the tanks/meter on the barge. If the quantities tally then the fuel oil receipt can be signed by the Chief Engineer. The filling valve at the manifold is then closed, the filling line disconnected and the manifold blanks replaced. The fuel oil record book should be filled in with details of the amount bunkered, tanks filled and times.

If the fuel received does not tally with bunker receipts, then malpractice may have taken place. These are rare but can include rigged soundings, air pumped in with the fuel (giving false readings on meter readings), delivery temperature higher than stated etc. If this is suspected, then a letter of protest may have to be issued.

Often the fuel is tested as it comes on board using modern sophisticated test kits. This can be used to check that the fuel is within the specification ordered. Unscrupulous suppliers have been known to mix water in with the fuel during pumping, and a test for water will detect this.

LETTER OF PROTEST

Should there be any dispute over the quantity of bunkers delivered the purchaser or his representative should issue a letter of protest that has been properly signed and stamped by both parties

Most bunker contracts stipulate that, unless notified at the time of delivery, any alleged difference in quantity delivered is time barred. The supplier can legally reject a claim because it was not presented within the agreed time limits. Initially, this may seem unfair, but the best chance of resolving a quantity dispute is at the time of delivery. Realistically, claims made after a vessel has departed have little chance of success.

It has been common practise to note a quantity difference on the bunker receipt, however, in some ports the customs authorities will not permit the receipt to be endorsed. In this instance, the only way the buyer can prove the claim was registered at the time of delivery is to issue a Note of Protest.

The ships crew should be aware that the contract will almost certainly have stipulated that the final and binding quantity shall be in accordance with the seller's gauges. It is therefore critical that these gauges are properly checked, before and after delivery, and it is important that the ship's figures be used only as an indication of quantity.

In Duplicate

Date ______________________

Ref ______________________

To ______________________ (Master Cargo Officer of Barge etc)
______________________ (Address)
______________________ (Address)

Dear Sirs

LETTER OF PROTEST FOR BUNKERING OPERATION ON __________ (date)

I, Chief Engineer of M/V ______________________ (name of vessel)

short received __________ tonnes of ______________ (grade of bunkers)

out of the __________ tonnes requested for on __________ (date bunkers received)

The bunkers were supplied by barge / tanker / shore tank ______________ (name)

Ref No __________ on ______ (date) at ______________ (location)

I hereby lodge a protest against short delivery.

Yours faithfully

____
(name of Chief Engineer)

cc: Ship Owner Manager Fuel Analysis Service
Duplicate copy to: Fuel Supplier

ACKNOWLEDGED RECEIPT

____
Signature of Master Cargo Officer of Bunker Barge/Tanker/Shore Tank

____
Name of Master/Cargo Officer of Barge/Tanker/Shore Tank   Date & Time

Chapter 3: Fuel Oil Testing

Testing can be done on board, or the sample sent to one of the agencies i.e. FOBAS (Fuel Oil Bunker Analysis and Advisory Service) for more comprehensive testing which includes calculation of the higher and lower calorific value of the fuel. Below are the results of FOBAS tests on a sample of RMG 380.

Property	Value	Limit
Viscosity 100°C	31.11 mm/s^2 (CSt)	(Max 35)
Viscosity 50°C	300 mm/s^2 (CSt)	(Max 380)
Density Kg/L	0.9899	(Max 0.9910)
Water % v/v	0.05	(Max 0.5)
Ash % m/m	0.03	(Max 0.15)
Micro Carbon Residue	13.88	(Max 18)
Sulphur % m/m	1.96	(Max 4.5)
Total Sediment % m/m	0.01	(Max 0.1)
Pour Point °C	<- 3	(Max 30°)
Flash point °C	>70°	(Min 60°)
Gross CV MJ/kg	42.86	
Net CV MJ/kg	40.56	
Si (cat fines) mg/kg	13	(Max 80)
Al (cat fines) mg/kg	5	(Max 80)
Vanadium mg/kg	68	(Max 300)
Sodium mg/kg	5	
Iron mg/kg	21	
Phosphorus mg/kg	1	
Lead mg/kg	< 1	
Calcium mg/kg	5	
Zinc mg/kg	<1	
CCAI	853	

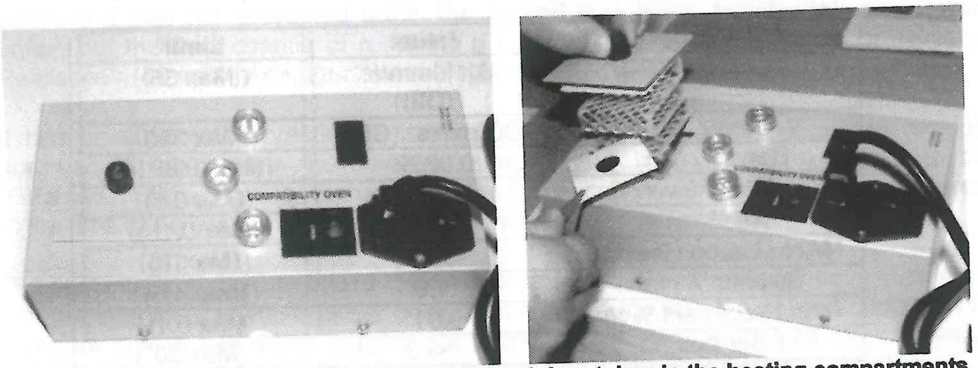
Note: v/v = volume for volume by proportion; m/m = mass for mass by proportion

On Board Testing

Tests usually carried out on board are for compatability, water content, density, viscosity, pour point and calculated carbon aromacity index (CCAI). Tests can be carried out for pour point and undisolved solids. Passenger vessels must carry a means of testing the flash point.

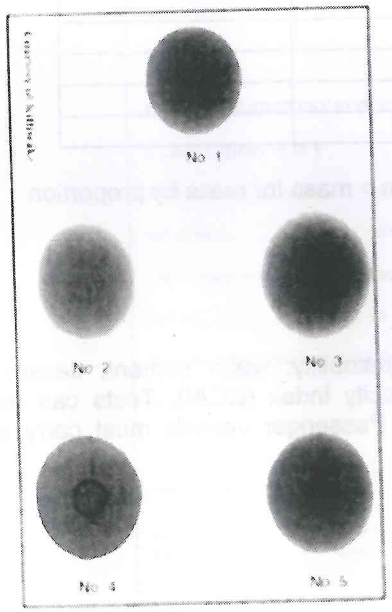
Stability and Compatibility

Before taking on bunkers it is advisable to take a sample from the suppliers tanks and test it for stability and compatibility. If the fuel is unstable it is likely to stratify in the tanks and form asphaltenic sludges. If it is incompatible, then although it may be stable on its own, when mixed with existing bunkers in the ships tanks, it may form sludges. This is why fuel should be bunkered into empty tanks whenever possible.



Stability testing. The samples are placed in aluminium tubes in the heating compartments. A drop of the fuel is placed on the chromatographic paper before being placed in the oven.

A small sample of the fuel is heated to 50°C to encourage instability. A sample of the preheated mixture is placed on chromatographic paper and dried in a small oven. The resultant spot is compared against reference samples. In the case of a test for compatibility the two fuels are mixed in the same proportions as would be found in the bunker tanks and the procedure repeated.

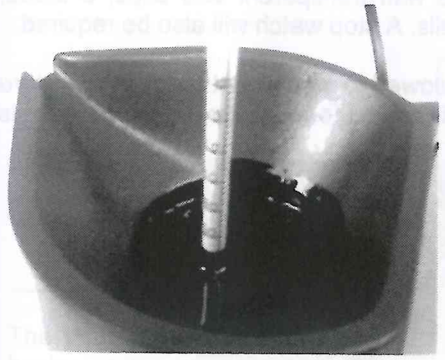


If a dark spot can be discerned in the centre of the fuel oil spot then the fuel can be considered unstable or in compatible when mixed.

Note: It is quite possible for a fuel to be stable in its own right, but become incompatible and unstable when mixed with another bunker fuel.

Density

A sample of the fuel is heated to 50°C and a hydrometer lowered into the oil. When a steady condition has been reached, a reading is taken where the meniscus touches the hydrometer stem and the density corrected to that at 15°C in vacuo.

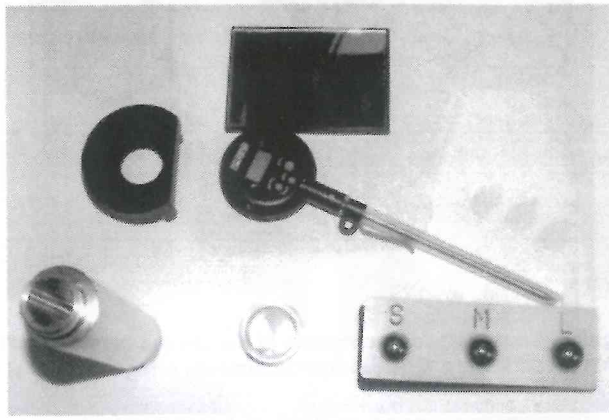


The Kittiwake test kit shown contains a heater and an on board processor, and a range of hydrometers to measure density from 0.8 to 1.1. When the sample reaches 50°, the densiity is measured and entered into the processor. The reading will automatically be corrected back to 15° in vacuo. If the viscosity is then entered, the Calculated Carbon Aromacity Index will automatically be calculated.

Viscosity

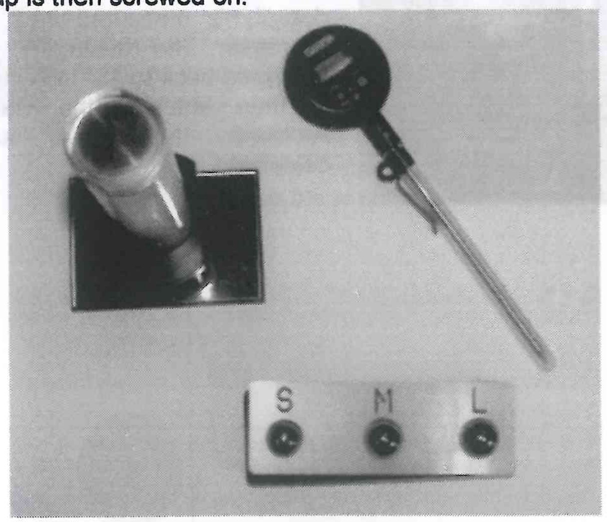
Although the laboratory test for viscosity of fuel involves timing a quantity of fuel passing through an orifice, the on board testing of viscosity usually involves rolling a ball down a tilted or inverted tube, which contains a sample of the fuel.. The time the ball takes to roll down the tube is measured and the result read directly from the instrument or from a conversion chart.

Method 1

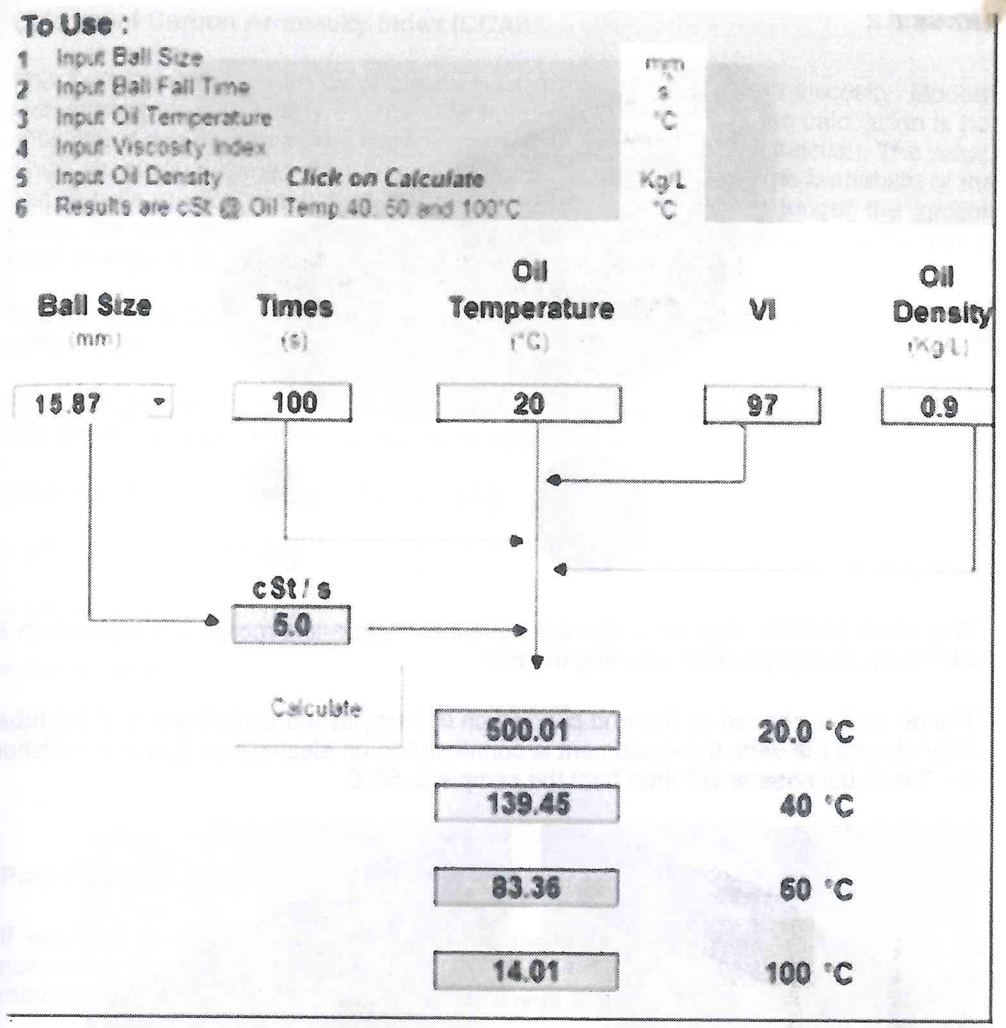


The equipment consists of an aluminium tube with transparent end caps, a stand, mirror, thermometer and three different sized balls. A stop watch will also be required.

The tube is half filled with oil which has been allowed to stabilize at room temperature and a ball placed in the tube. The tube is filled to the very top so that all air is excluded. The cap is then screwed on.



The tube is then inverted and placed on the stand over the mirror at the same time as the stopwatch is started. When the ball reaches the bottom of the tube, as observed in the mirror, the stopwatch is stopped, and the time noted. Invert the tube and repeat the timing test several times to check for a consistent reading. Unscrew the top and measure the temperature of the oil



The viscosity can be calculated using the spreadsheet supplied with the kit. (More basic versions will obtain the viscosity from a supplied graph).

Input the ball size, time, temperature, and Density of the fuel. Viscosity Index applies only to lube oil so leave this at its default setting.

If the time taken for the ball to move from top to bottom is too long, then warm the fuel to 30 – 40° by placing the tube full of oil in warm water. If the time taken for the ball to move down the tube is too short, then use a larger ball.

Method 2

This more sophisticated viscosity test equipment again incorporates a tube which is filled with the sample after inserting the ball.

The air vent is opened on the end cap, which is then pushed on to the end of the tube. After closing the vent, the equipment is connected to an electricity supply and switched on. The in built heater will then heat the sample to 50°C.

The device is then tilted from one position to the other as shown. The time taken for the ball to roll down the tube is automatically measured and converted to viscosity.

Again the instrument contains an on board processor, so by inputting the density, once the viscosity has been established, the CCAI for the fuel will be calculated.

Calculated Carbon Aromacity Index (CCAI)

This is calculated from an empirical formula using the density and viscosity. Modern test kit computers will carry out the calculation for you. However the calculation is not difficult and can be carried out using a calculator with a logarithmic function. The value, which should lie between the limits of 800 and 870 give a guide to the burnability of the fuel and its ignition characteristics. The higher the number, the longer the ignition delay, the more explosive the detonation, and the hotter it will burn. This number is used as a guide to setting the fuel quality lever on the fuel pumps.

The formula is CCAI = (density in kg/m^3 @ 15°C) - 81 - 141log[log(viscosity in cSt @ 50°C + 0.85)]

For example: We have bunkered a fuel and measured its density at 15°C as 989.9kg/m^3, and its kinematic viscosity as 300cSt at 50°C.

CCAI = 989.9 - 81 - 141log[log(300 + 0.85)]

= 908.9 - 141log[2.47835]

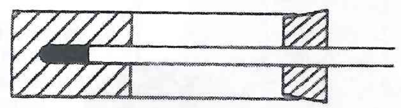
= 908.9 - (141 × 0.39416)

= 908.9 - 55.6

= 853.4

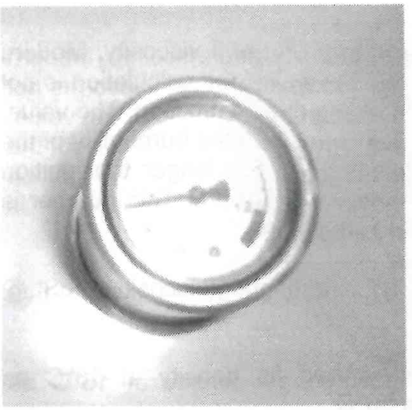
Pour Point

If required, an on board estimation of the fuel's pour point can be made. It is not necessary to purchase special equipment for this purpose. The following on board method indicates if the fuel has a high or low pour point.



A sample of 200ml of fuel oil is placed in beaker of 250ml and heated by immersing the beaker in a water bath of boiling water. It may be necessary to hold the fuel oil beaker in the water while the fuel is heated to about 50°C. The beaker is then placed in a refrigerator and the temperature noted by means of either a digital or stem thermometer. At increments of 3°C the beaker should be removed from the refrigerator and the contents tilted to see if they flow. This should be repeated until no flow is observed.

Water

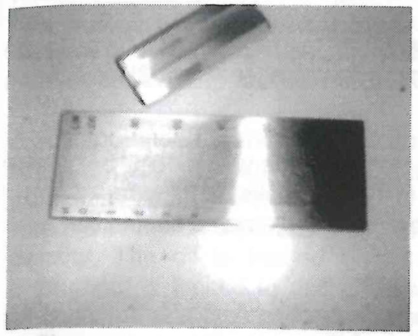


The water test for fuel oil is similar to that for lubricating oil. A sample of the fuel (5ml) is diluted with kerosene (20ml) and placed in the test container. A satchet of calcium hydride is opened and placed upright in the container without allowing the chemical to come in contact with the diluted fuel. A lid containing a pressure gauge is screwed on carefully and the container inverted and shaken for 2 minutes. The water in the fuel reacts with the metal hydride producing hydrogen. The pressure in the container is proportional to the water content and can be read from the scale.

If water is present then a further test should be made for salt water content. This involves separating the water from the oil and testing for the presence of sodium chloride using a go/no go indicator.

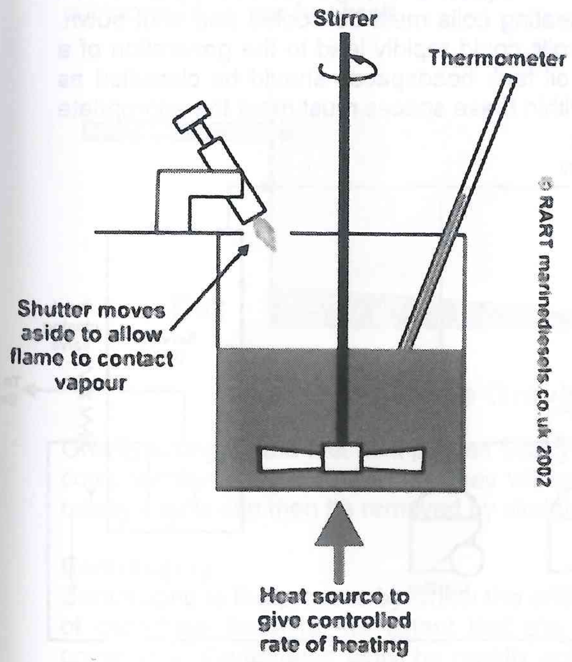
The latest equipment has a pressure transducer fitted in the base of the container with a digital read out.

Particulates



The equipment consists of a steel plate which has a very shallow groove machined into it that tapers in depth between 10 and 100 microns. A spot of fuel is placed at the top of the plate and is then scraped down the length of the plate in a single sweep with the supplied straight edge. The particles can then be seen lying at the position relative to their size on the plate.

Alternatively a spot of fuel is placed between two clear perspex plates and rubbed together for about 10 seconds. The amount the plates are scratched is indicative of the abrasive solids present.



Flash Point

In the Pensky Martens apparatus approximately 75ml of sample is heated at a slow constant rate with continual stirring.

A small flame is then directed through an opening at regular intervals with simultaneous interruption of the stirring.

The flash point is the lowest temperature at which application of the test flame causes the vapour above the test portion to ignite.

Alternatively an automatic go/no go flash point tester can be used.

Minimum flash point for bunker fuel is 60°C.