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INTERNAL COMBUSTION ENGINE FOR FISHING BOATS (IV)

Lubrication and Cooling

Compiled

by

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PREFACE

The present textbook contains the fourth set of lectures on internal combustion engine, prepared for the trainees of the Marine Engineering course at the SEAFDEC Training Department during 1983-85.

The first volume in this series is a general introduction to the basic principles of the internal combustion engine. In the second volume I have dealt more specifically with the main parts of diesel engines, particularly the four-stroke cycle engine which have a wide marine application in the Southeast Asian Region.

The third volume, subtitled "How Fuel Burns in an Engine" described all the engine parts directly involved in fuel combustion, as well as properties of different types and qualities of fuel.

This volume deals with lubrication and cooling of diesel engines. The equipment described here, although sometimes referred to as "accessories", is nonetheless indispensable for engine operation.

April 1985 Kazuyuki Tobo

LESSON 12 LUBRICATION

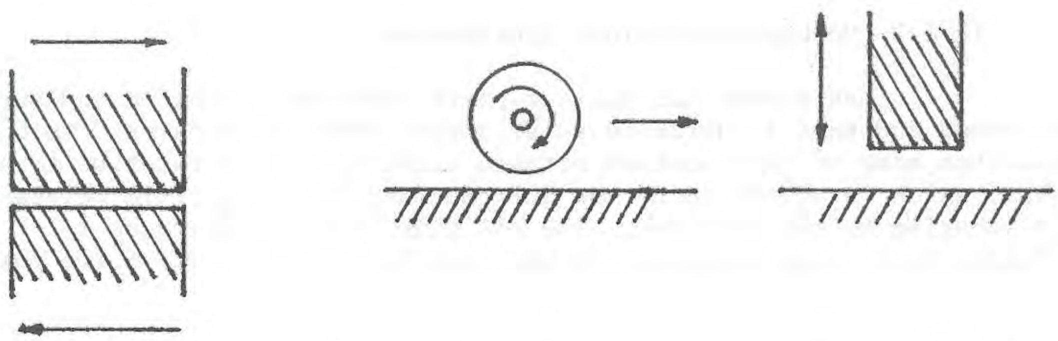
12.1 Function of Lubricating Oil

There are five main reasons why lubricating oil is used in engines. They are: prevention of wear, minimizing friction between moving parts, cooling, sealing and cleansing.

12.1.1 Prevention of wear

A metal surface in contact with another metals surface is either in fixed or unfixed condition. If two metal surfaces are fixed to each other, there is little possibility of trouble between them. On the other hand, a moving contact of two metal parts sometimes causes some trouble, if there is nothing to prevent a direct contact between them.

The moving contact can be classified into three tupes; face-to-face sliding, rolling on a surface, and collision contact, as shown in Fig. 12-1. Face-to-face contact with sliding occurs between the piston ring and the cylinder liner, between the crankshaft and the main metal, and so on. Rolling contact occurs between the camshaft and



(a) sliding (b) rolling (c) colliding

Fig. 12-1

the roller glide for fuel injection pump. Collision contact is seen between the valve and the valve seat, between the rocker arm and the valve tapet, and so on. The contact of gear teeth set is a mixture of those three. If contact occurs only between metal surfaces, without any lubricant, heat will be produced by friction and both surfaces wear off.

When contact is made in a violent manner, a great amount of heat generated causes a seizure, in which both metal parts melt and stick to each other. To avoid seizure it is advisable to avoid using homogeneous metals for the contacting parts; instead, heterogeneous metals, for example, steel and copper, or steel and cast iron, are generally in use. In addition, white metal and kelmet metal are applied as anti-friction alloy metals.

The purpose is to diminish to the minimum the melting stick which the inter-metal contact may cause, by differentiating friction coefficient, melting point and hardness.

However, when the speed and the pressure in sliding contact are greater, it increases wear and seizure. Lubricating oil is used to reduce the frictional heat by making an oil film between the metals, so that they are not in direct contact, but in "rolling contact" created by droplets of lubricating oil. Such contact made between metals by lubricating additive is called "three dimension touch" in contrast to "duality touch" made between the metals in direct contact.

12.1.2 Mechanics of fluid lubrication

An engine has various parts that are in motion and must be guided and kept in definite relationship with other parts. This indicates more or less contact between surfaces moving relative to each other, which is minimized by the introduction of an oil film between and adhering to the surfaces. The oil film offers resistance to shearing when relative motion of the surfaces occurs. The force per

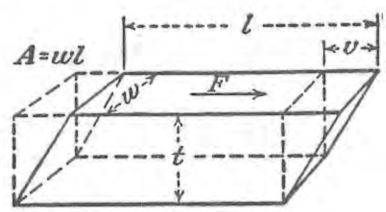


Fig. 12-2

unit area of oil film, F/A (Fig. 12-2) required to move one of the surfaces relative to the other is directly proportional to the rate of shear v/t and the viscosity μ.

Thus, $F/A = \mu v/t = \mu dv/dt$ (12-1)

Equation 12-1 defines the *absolute viscosity* μ as the force per unit area required to move a surface at unit velocity when it is separated by a fluid of unit thickness from a stationary surface. The unit of absolute viscosity is the poise.

Thus, $1 \text{ poise} = 1 \text{ dyne cm}^{-2}(\text{cm/sec})^{-1} \text{ cm}$

$= 1 \text{ dyne sec/cm}^2$

$= 2.09 \times 10^{-3} \text{lb force sec/ft}^2$ (12-2)

The application of a load or force to the moving surface, some component of which is perpendicular to the direction of motion and toward the stationary surface, will force the fluid film out of the space between the two surfaces and result in metal-to-metal contact. A reversal of this force will increase the clearance space and permit a thicker oil film. Thus, a film of oil may be maintained between two surfaces if the time required to force the film out of the space between the two surfaces is greater than the time between load reversals, provided that a supply of lubricant is available to fill the clearance space when this is increasing.

If one surface is inclined at a small angle to the other surface (Fig. 12-3) the relative motion of the surfaces will develop an oil-film pressure that will support a load. Assuming that the bearing surfaces are of infinite width (no end leakage of the oil) and that the oil adheres to the surfaces,

$$P = P_o + \frac{6\mu vm}{(2t + ml)} \times \frac{lx - x^2}{(t + mx)^2} \qquad (12\text{-}3)$$

where P = pressure, kg/cm^2 at a distance x, cm
P_o = external pressure on oil film
v = velocity, cm/sec of surface motion
t = smallest clearance, cm
l = surface length, cm
m = tangent of angle of inclination

The solution of Eq. 12-3 for a given condition provides information regarding the pressure distribution (Fig. 12-3)

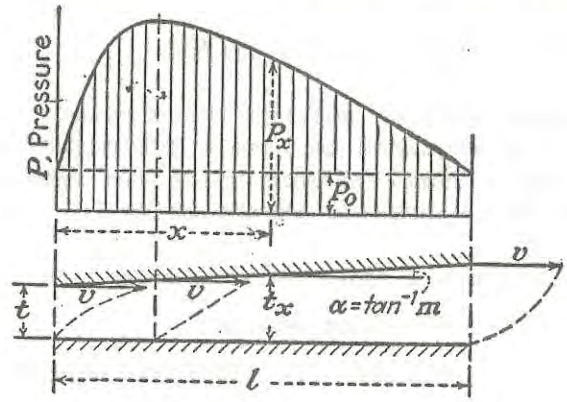


Fig. 12-3

The area beneath the total-pressure curve is a measure of the load that can be supported by the oil film, provided that the external pressure P_o does not also act downward on the upper surface. Equation 12-3 indicates that an increase in loading or decrease in surface velocity or viscosity of lubricant decreases the clearance between the surfaces. With a very small clearance the surface irregularities will break through the oil film and come in contact with each other. Thus, a high degree of surface smoothness is desirable to minimize wear under thin-film conditions.

Surface irregularities are measured with a profilometer in microinches (millionths of an inch). Some surfaces in regular production have average irregularities as low as 1 to 2 microinches.

The *coefficient of friction* is defined as the ratio between the resistance due to friction in the direction of motion and the load carried normal to the plane of motion. This coefficient varies appriciably with the type of bearing and its lubrication as shown in the following table.

Type of bearing	Range	Average
Unlubricated or very poorly lubricated ...	0.10 -0.40	0.160
Semilubricated..........................	0.01 -0.10	0.030
Well lubricated.........................	0.002-0.010	0.006
Roller bearings.........................	0.002-0.007	0.005
Ball bearings...........................	0.001-0.003	0.002

12.1.3 Cooling

The heat generated in combustion chamber is used as power. At the same time, some of it escapes with exhaust gas and is also carried away with cooling water. The heat conducted to piston and cylinder is carried away by the lubricating oil. Moreover, the heat generated in each sliding part of engine is carried away by the lubricating oil, thus all the sliding parts of the engine are being cooled. The lubricating oil transfers the acquired heat to the cooling water in the lubricating oil cooler again acquires heat from piston, cylinder liner and other sliding parts and so on. Thus, the lubricating oil has a repeated cooling effect in the engine.

12.1.4 Sealing

The oil film made between the piston ring and the cylinder liner prevents the combustion gas from being blown-by into the crankcase.

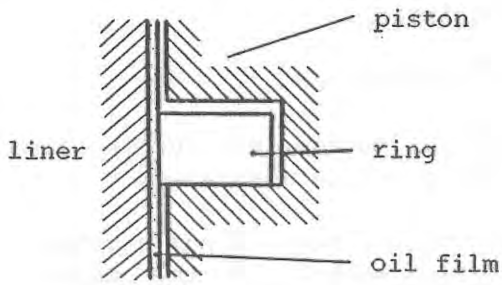


Fig. 12-4

This effect can be verified by the fact that a sufficient compression pressure is produced for easy start even in the engine with worn piston ring and cylinder liner when a small amount of lubricating oil is poured into its combustion chamber.

12.1.5 Cleaning and rust-proofing

With improved additives, the recent lubricating oils have the ability to remove stains in the moving parts, metal dust and combustion products, which would otherwise remain between bearings and shafts resulting in their increased wear. Covering the surface of metals, lubricating oil also acts as an anticorrosive. However, one has to be careful not to use a lubricating oil of greater oxidation and deterioration, which is sure to accelerate the production of rust.

12.2 Methods of Lubrication

The methods of lubrication can be classified broadly into the following categories:

1. Manual lubrication;
2. Drop-feed lubrication;
3. Splash or bath lubrication;
4. Ring, chain, collar oilers;
5. Centrifugal oiling;
6. Grease oiling;
7. Force-feed lubrication.

Although the different lubricators within any category differ quite markedly in outward physical appearance and detail, they have similar basic characteristics.

12.2.1 Manual lubrication

Lubricating methods may require human action in one form or another at some point in operation.

The amount and type of human action vary depending on the particular lubricating method. In the above classification, the term manual lubrication applies to methods in which the operator is directly responsible for the quantity of lubricant applied to the bearing surface, the interval of lubrication, or both. If an operator is required periodically to maintain the lubricant level in a reservoir but the quantity of lubricant reaching the bearing and the interval of lubrication are controlled mechanically and in dependently of the operator, the lubricating method is still classed as manual.

Although the initial cost of manual lubrication is low, the maintenance costs can be very high. In addition, reliability may be low owing to considerable dependence on human action.

With the manual method the lubricant is quite prone to contamination.

Lubricant flow with manual lubricators is poorly regulated, nonuniform, nonadaptable, and noncontinuous.

Generally speaking, manual lubrication is satisfactory only for lightly loaded or low-speed bearings and other machine elements infrequently used.

Typical applications include open gears, chains, wire rope, and other inexpensive or rough machinery.

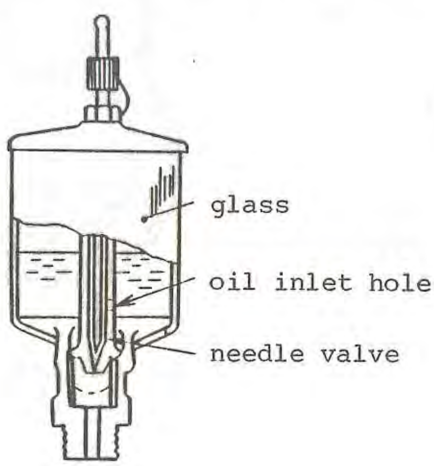
12.2.2 Drop-feed lubrication

Drop-feed oilers are gravity-flow lubricators employed to deliver lubricant drop by drop to individual bearings or other machine elements as shown in Fig. 12-5(a) and (b). They are used to best advantage when lubrication points are readily accessible and few in number.

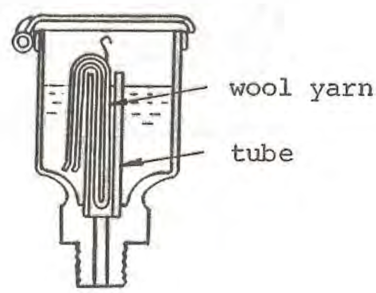
Their cost depends on the lubricator in question but, generally speaking, is relatively low. Maintenance cost depends on the type of service and location. For many lubrication points or for poorly accessible locations, maintenance expenses can be quite high.

Depending on the lubricator, lubricant flow may or may not be stopped and started automatically. Automatic operation increases reliability.

Oil delivery from such lubricators is characterized as gravity induced, variable with time (due to variation of flow rate with oil level), nonadaptable, and dropwise. Depending on the lubricator, flow regulation may or may not be readily accomplished.



(a) by needle valve



(b) by wool yarn

Fig. 12-5

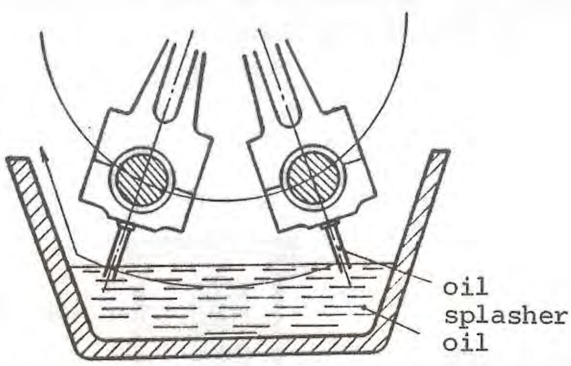
12.2.3 Splash and bath lubrication

This type of lubrication is commonly used for machinery having high-speed moving parts which dip into oil and splash it onto the bearings or other machine elements. A splash system requires enclosing the mechanism to be lubricated.

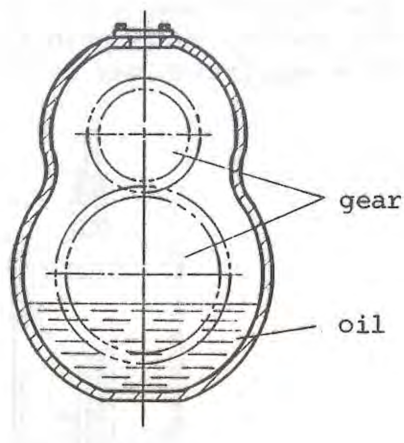
The initial cost of a splash system depends primarily on the expense incurred in enclosing the mechanism. Maintenance costs for splash systems are relatively low. A splash system is automatic, requires little operator attention, and is quite reliable. The fact that the system is enclosed is conducive to preventing lubricant contamination. Lubricating oil is recirculated, thereby making splash lubrication attractive from the points of view of economy as well as avoiding leakage.

Lubricant flow is characterized as fairly uniform, somewhat adaptable to changes in speed, and fairly continuous. Flow regulation is not possible except by changing the oil level in the crankcase or by adding scoops or dippers to the moving machine elements.

Typical applications include internal-combustion engines, chain drives, and enclosed gear sets, as shown in Fig. 12-6(a) and (b)



(a) splashing



(b) oil bath

Fig. 12-6

In some diesel engines, an oil splasher as shown in Fig. 12-7 is fitted to the large end of the connecting rod (Fig. 12-6a). The lubricating oil stored at the bottom of the crank case is splashed by this oil splasher to be supplied to the inner wall of cylinder liner, piston, cam shaft bearings and other main bearings. In addition, the crank pin is lubricated through an oil groove in the center of the oil splasher.

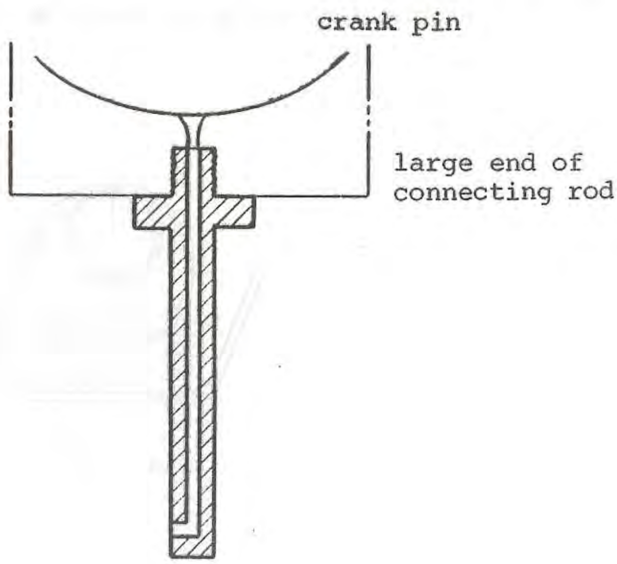


Fig. 12.7

12.2.4 Ring, chain, and collar oiler

These lubricators are applicable to horizontal rotating shafts. The ring or chain oiler encircles the shaft and turns freely on it. The collar oiler is rigidly fixed to the shaft. Each provides an automatic oiling system by bringing oil to the bearing clearance from the oil reservoir into which it dips as it turns.

Initial cost depends primarily on the expense of the special bearing housing that must be built to contain these lubricators. Maintenance cost is usually low.

Ring, chain, and collar oilers are reliable as long as the proper bearing reservoir oil level is maintained. When the bearing seal is tight, oil loss is small and the possibility of lubricant contamination is low.

Oil feed starts and stops automatically and consumption is low, for recirculation is readily accomplished.

Lubricant flow is characterized as uniform, continuous, and somewhat adaptable to changes in speed. Flow regulation is not readily accomplished.

Figure 12-8 gives an example of oil-ring lubrication.

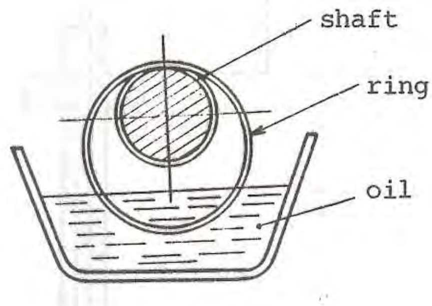
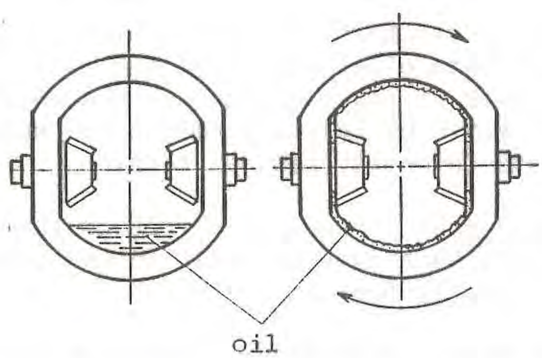


Fig. 12-8

12.2.5 Centrifugal oiling

In this method, the lubricating oil is supplied by centrifugal force. This lubrication is applied to the clutch reverse gear housing. As shown in Fig. 12-9(a), the lubricating oil is stored at the bottom of the housing. When the housing revolves, the oil is scattered to the inner wall of the housing by centrifugal force, as seen in Fig. 12-9(b).



(a) in rest (b) in rotation

Fig. 12-9

12.2.6 Grease oiling

A manual screw-type grease-cup device for dispensing soft lubricating grease is shown in Fig. 12-10. Height of the handle above the cup indicates the amount of lubricant left in the cup. Some grease cups have a spring-loaded piston that permits the grease to feed automatically.

Fig. 12-10

There is a great variety of grease guns that differ in size and dispensing pressure. A typical lever-type grease gun is shown in Fig. 12-11.

Turning handle *A* cause the spring-loaded plunger *B* to move grease through port *C* to the lever-operated pump whose plunger, traveling in the high-pressure chamber *D*, forces the grease past the ball valve *E* to the coupling and fitting.

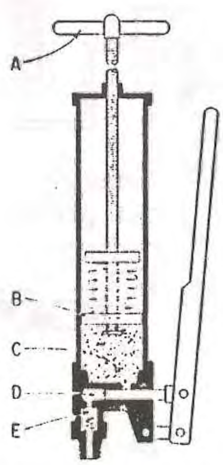


Fig. 12-11

12.2.7 Force-feed lubrication

Pressure circulating systems for lubricating oils employ either gravity or pumps to develop the operating pressures necessary. Generally, a pressure circulating oil system is designed to lubricate a number of parts on the machine. Since oil is recirculated, maximum oil economy is possible and cooling by the lubricant is practicable.

Since a pressure circulating system is built into the machine, the initial cost of piping, controls, pumps, and lubricant reconditioning equipment can be high, but the resistance of the system to entry of contaminants and its durability lead to quite low maintenance costs.

Once the various controls have been set, the operator need only make routine inspections and add make-up oil when required. In systems employing pumps to circulate the lubricant, flow is positive. In general, pressure circulating systems permit regulation of oil flow and provide uniform, continuous delivery. Pressure circulating systems are often employed when heavy-duty costly bearings must be protected and maximum dependability is desired.

Fig. 12-12 shows the schematic diagram of the forced oiling system with an oil cooler. There are other system of forced oiling without oil cooler or on the contrary, even a manual supply of oil can be made by means of a wing pump. The system shown in Fig. 12-12 is most commonly adopted.

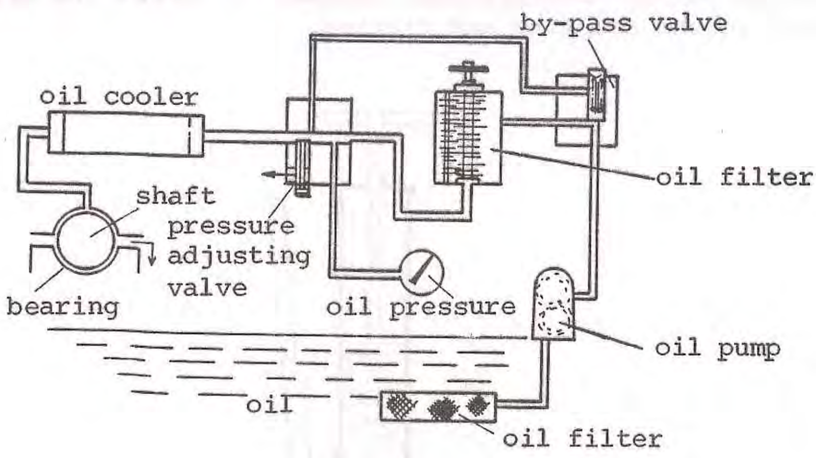


Fig. 12-12

12.3 Equipment for Lubrication

12.3.1 Oil pump

The oil pump is driven by means of gear transmitted to the crank shaft, and draws up the lubricating oil at the bottom of the crankcase. The bottom of the crank case is provided with an oil filter which prevents the carbon and sand deposited in the crank case from being drawn up into the oil pump. The mesh of this oil filter is approximately 250 meshes in length and 24 meshes in width.

The oil pump, which is of a gear type, has a construction as shown in Fig. 12-13. The gear is driven by the revolution of the main gear, and the oil is carried by the gear teeth and its hollow part.

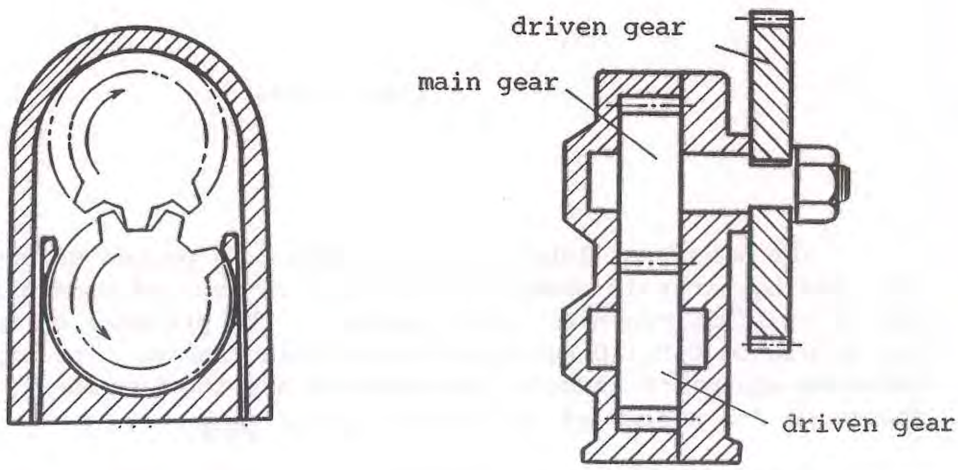


Fig. 12-13

As the oil pressure is low at the inlet, there occurs a reversing flow of the oil due to a pressure drop between outlet and inlet. The power of this reversing flow is greatly affected by the side clearance between the gear and the wall of the case.

If the side clearance is bigger, as shown in Fig. 12-14, the pressure is lowered by the reversing flow. Consequently, sufficient attention is required when the oil pump is assembled after disassembling. The side clearance or oil pump in marine diesel engines is the range of 0.046 to 0.075 mm.

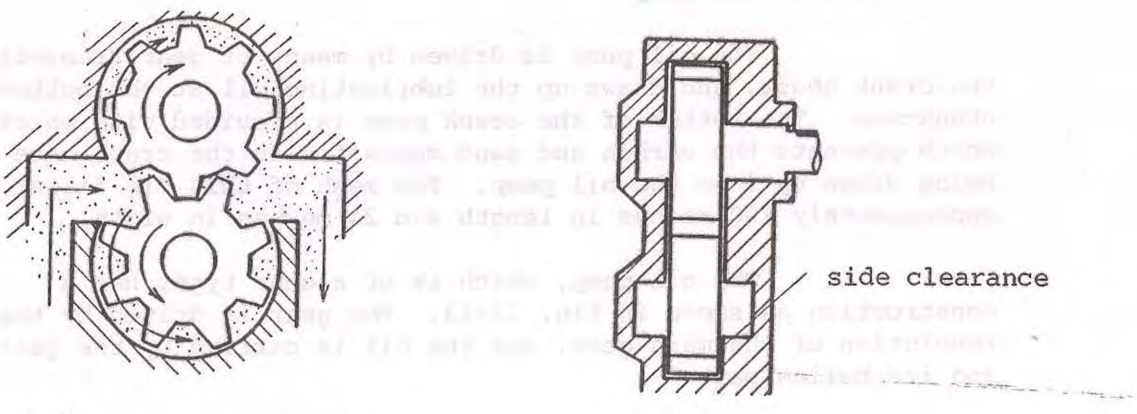


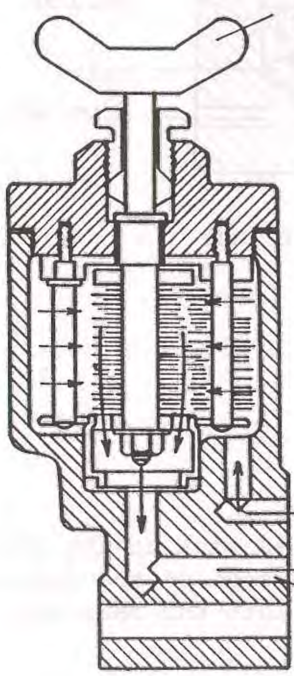
Fig. 12-14

The amount of lubricating oil which is pumped up and sent to the sliding parts is about 20-40 L/PS.hr in low and middle speed engines, and 35-55 L/PS.hr in high speed engines. The pressure of lubricating oil should be 0.5-1.0 kg/cm² at the sliding parts. The piping and oil cleaning equipment reduces the pressure and the lubricating oil pressure should be 2-5 kg/cm² at the outlet of oil pump.

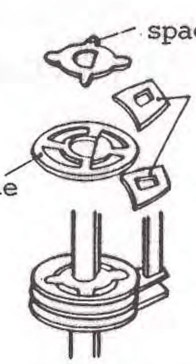
Engines of more than 100 PS have a hand priming pump which can send lubricating oil to all sliding parts in advance of engine starting.

12.3.2 Lubricating oil filter

It is necessary to remove carbon soot and sand before the lubricating oil drawn up by the oil pump is delivered to each sliding part of bearing. For this purpose a lubricating filter is provided on the lubricating system. The comb type lubricating oil filter is used for many diesel engine, and the construction is shown in Fig. 12-15.



(a)



(b)

Fig. 12-15

All foreign matter deposited on the filter plate can be taken off by revolving the handle. As the clearance between the filter plates is 0.1 to 0.15 mm, very small dust particles cannot be removed. But, this lubricating filter is cleaned during operation.

There is also a double type filter which consists of two filters of the above type with a change valve, permitting to clean its interior alternatively during operation.

Some large engines are equipped with by-pass oil filter. In this type of lubricating oil filter, the lubricating oil which is cleaned is not sent to sliding parts, but is returned directly to the crankcase. The purpose of this is to help prolong life of in-line lubricating filter and to take off more fine substance than in-line filter system without the pressure dropping.

12.3.3 By-pass passage and by-pass valve

The lubricating system is also provided with a by-pass circuit, by which the lubricating oil is supplied to the oil cooler or each lubricating circuit of the engine through the by-pass valve from the oil pump without passing the lubricating filter. Fig. 12-16 shows how the by-pass valve acts.

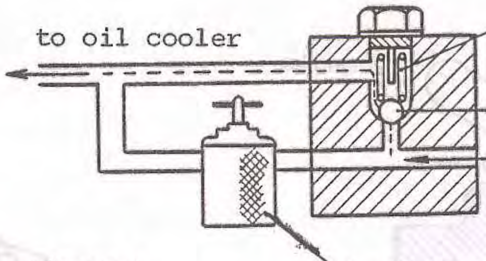


Fig. 12-16

Usually the oil delivered from the oil pump is sent to the oil cooler through the lubricating filter. But, in the case of the lubricating filter being clogged, the oil does not flow into the bearings, resulting in their seizure.

As a rule, oil should be delivered after being cleaned through the lubricating filter. To prevent seizure, however, a measure is taken so that even the unfiltered oil is supplied to the bearings through the by-pass circuit with the by-pass valve. The pressure of the oil which is delivered by the oil pump and passes through the oil filter is 2 kg/cm². The pressure of the oil before the filter gets higher and higher by the continuous delivery of the oil by the oil pump. The by-pass valve is pushed up when the pressure reaches about 9 kg/cm². So, the oil filter is required to be cleaned periodically to prevent its clogging.

Further, one must pay attention to the fact that the unfiltered oil supplied to the bearings through the by-pass circuit is not measured by the pressure gauge. If the oil passes always through the by-pass circuit, not only is dirty oil supplied to the bearings, but also the oil pump is always under high pressure.

This leads to the increased wear of the bearings and the pump. It is necessary to learn where the by-pass valve is by a trial disassembling, because it is very difficult to know from appearance where it is.

12.3.4 Pressure gauge and pressure adjusting valve

As seen from the schematic diagram of the lubricating system shown in Fig. 12-12, the oil delivered through the oil filter or by-pass circuit is adjusted in pressure by means of the pressure adjusting valve before reaching the oil cooler (or the bearing in case of the engine with no oil cooler). The oil delivered in excess is sent back to the crank case.

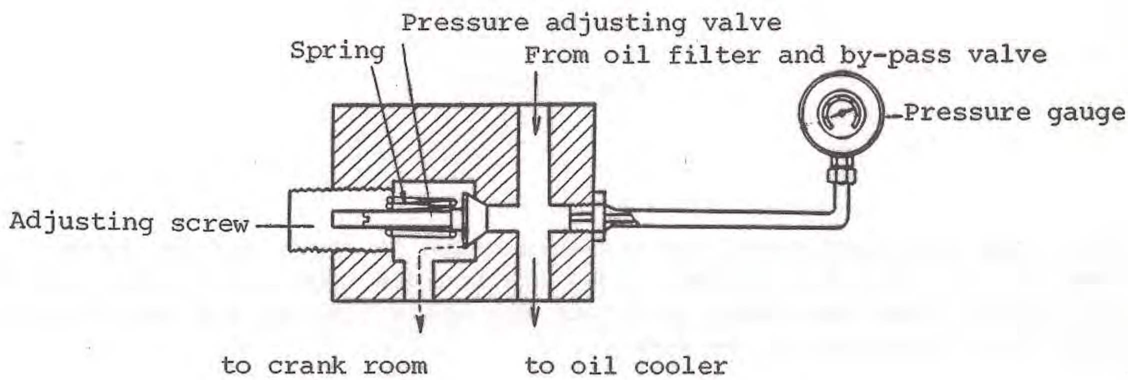


Fig. 12-17

The pressure after adjustment is indicated on the pressure gauge. Figure 12-17 shows the equipment for the above purpose. As the oil delivered from the oil filter always exceeds the necessary volume, the spring of the pressure adjusting valve is compressed to send back the excessive oil to the crank case. The adjusted oil pressure is indicated on the pressure gauge. When the pressure is low, tighten the adjusting screw to compress the spring. Then, the pressure is raised as the volume of the oil returned decreases. When the pressure is high, loosen the adjusting screw. Then, the pressure is lowered. A flaw in the seat of the pressure adjusting valve often causes an excessive movement of the indicator of the pressure gauge. In this case it is necessary to file the seat to remove the flaw.

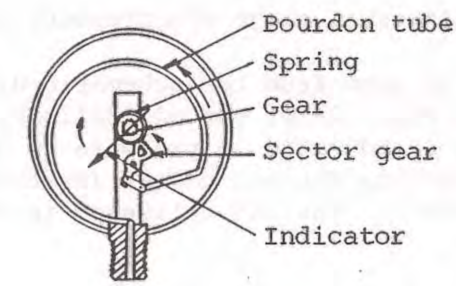


Fig. 12-18

The inner construction of the pressure gauge is shown in Fig. 12-18. When the oil pressure becomes high, the bourdon tube filled with oil lengthens toward the arrow to rotate the wheel through the sector gear. Thus, the indicator can be moved.

12.3.5 Oil cooler

Figure 12-19 shows the construction of the lubricating oil cooler.

A number of cooling water pipes are brazed to the tube plates on each end in their outer circumference and several buffer plates are provided on the cooling water pipes. The lubricating oil sent from the oil pump passes through the outside of the cooling water pipes and its heat is carried away by the cooling water. The temperature of the oil usually falls by 6-12°C in the cooler. It, however, falls less if the cooling water is of high temperature.

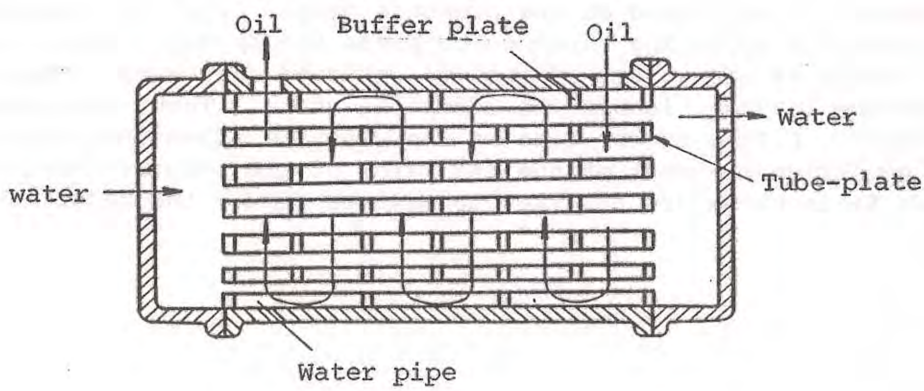


Fig. 12-19

It is necessary to keep the engine lubricating oil temperature below 60°C. But, it is not good to keep it at too low temperature in winter. In winter it is recommended to keep the lubricating oil above 40°C by passing no water through the cooling water pipes or by introducing no oil to the oil cooler.

Special attention should be paid to any oil leakage in the oil cooler, which may be caused by corrosion. When the brazing of the cooling water pipes and tube plates are stripped off, the lubricating oil is mixed with the cooling water during the operation as the oil pressure is higher than the water pressure and discharged out with the water. This causes a lack of the lubricating oil, bringing the seizure. If the operation of the engine is made in daytime, you may take note of the trouble by finding that the oil is in the water. In the operation at night, however, it is difficult to take note of the trouble.

Therefore, it is very important to make a periodical check of the oil cooler. Also, the anticorrosive zinc bar, if it is used, should be replaced periodically. The oil cooler should be disassembled once a year to remove the scale deposited on the cooling water pipes, and at the same time, a water pressure test should be made. The zinc bar should be removed for checking every six months at least, and replaced with a new one, if necessary.

The area of radiant heat in oil cooler is calculated by the formula:

$$F = Q/k\ \Delta t$$

where F = Area of radiant heat (m^2)

Q = Heat amount to be taken off (cal/hr)

Δt = Difference of temperature between inlet and outlet of oil cooler (℃)

k = Coefficient of overall heat transmission.

The coefficient of overall heat transmission is about 200-300 Kcal/m^2.hr°C. The area of radiant heat is usually calculated simply as 0.01-0.03 m^2/PS.

The velocities of cooling water inside the pipes and lubricating oil outside the pipes are 0.6-1.2 m/sec and 0.4-0.6 m/sec, respectively.

12.4 Lubricating Oil

12.4.1 Functions and classification of lubricating oil

As explained in section 12.1 above, lubricating oil has the following functions:

A) Lubricating

B) Cooling

C) Sealing

D) Cleaning

There are many kinds of lubricating oils according to the use. The oils for internal combustion engines for ships are classified by the viscosity and the type of service.

(1) Classification by viscosity

The lubricating oil is generally classified by the viscosity number which is established by SAE (Society of Automative Engineers in USA). There are seven classes: EW, 10W, 20W, 20, 30, 40 and 50. "W" is attached to the lubricating oil which should be used in winter. There are three kinds of viscosity units as shown below.

SUS (Saybolt Universe Second)

RW (Red Wood)

CST (Centistokes)

In Japan, only CST is used. In SAE the viscosities are specified at -17.8°C and 98.9°C. There is, however, no specification on the apparent viscosity at room temperature. The following table shows the relation between SAE Number and viscosity.

Table 12-1

SAE No.	Viscosity -17.8°C			Viscosity 98.9°C		
	SUS	RW	CST	SUS	RW	CST
5 W	4,000 or less	3,500 or less	896 or less	-	-	-
10 W	6,000 - 12,000	5,200 - 10,500	1,303 - 2,606	-	-	-
20 W	12,000 - 48,000	10,500 - 42,000	2,606 - 10,423	-	-	-
20	-	-	-	45 - 58	40.9 - 51.9	5.73 - 9.26
30	-	-	-	58 - 70	51.6 - 62.0	9.62 -12.93
40	-	-	-	70 - 85	62.0 - 75.3	12.93 -16.77
50	-	-	-	85 -110	75.3 - 97.5	16.77 -22.68

(2) Classification by faculty or service

The API classification established by the American Petroleum Industry is made in accordance with the oxidation stability, cleaning capability and other properties that are brought by additives. This classification plays a very important part in defining the quality or engine oil.

The API classification is as follows:

Table 12-2

Classification		API engine service description	ASTM engine oil description
Old	New		
DG	CA	Light duty diesel engine service. Service of diesel engines operated in mild to moderate duty with high quality fuels.	Oil meeting requirements of MIL-L-2104A
DM	CB	Moderate duty diesel engine service. Service of diesel engines operated in mild to moderate duty, but with lower quality fuels which necessitate more protection from wear and deposits	Oil for use in naturally aspirated diesel engines. Includes MIL-L-2104B oils where diesel engine test was run using high sulfur fuel.
	CC	Moderate duty diesel engine service. Service of lightly supercharged diesel engines operated in moderate to severe duty.	Oil meeting requirements of MIL-L-2104B, MIL-L-46152. Provides low temperature antisludge, antirust performance.
DS	CD	Severe duty diesel engine service. Service of supercharged diesel engines in high speed, high output duty requiring highly effective control of wear and deposits.	Oil meeting requirements of MIL-L-45199B, MIL-L-2104C

Some of the older operation manuals for diesel engines list lubricating oils by old classification system, DG, DM, and DS. The classification system by API was modified to new one as described above about tem years ago.

The description by the old classification system is as follows:

a) Service DG: lubricating oil suitable for common diesel engines which use high quality diesel fuel.

b) Service DM: lubricating oil suitable for diesel engines operating under severe working conditions or with the low quality fuel which may accelerate wear.

c) Service DS: applied for the engines which are put in operation under very severe condition (for example, with supercharger), the engines which are of special type or in which there is the danger that an abnormal wera and deposits are produced because of the use of low quality fuel.

12.4.2 Properties of lubricants

Lubricants obtained by the refining of petroleum are used in internal combustion engines. The customary laboratory inspection provides data concerning various properties, the more common of which and their significance are listed below.

The gravity, degrees API, of a lubricant is an index of its specific gravity. It is of little significance as an index of the quality of a lubricant.

(1) Flash point

The *flash point* of an oil is the temperature at which a flash appears on the oil surface when a test flame is applied under specified test conditions. It is a rough indication of the tendency to vaporize.

(2) Fire point

The *fire point* is the temperature at which the oil ignites and burns for at least 5 sec under specified test conditions. It is of little significance.

(3) Pour point

The *pour point* is determined by noting the temperature at which the oil will not flow when under specified test conditions. It indicates the temperature below which oil will not flow to the oil-pump suction line. If dewaxed, the paraffin-base oils may have a lower pour point than asphalt-base oils.

(4) Viscosity

The *viscosity* of an oil is determined by the time required for a given quantity to flow through a capillary tube under specified conditions. For all but very small capillary tubes,

$$\text{Kinematic viscosity} = \nu = \mu/\delta = kt \text{ stokes} \qquad (12\text{-}5)$$

where μ = absolute viscosity, poises

δ = mass density, g/cm³

k = const for the viscometer

t = time for flow, sec

The relation between kinematic viscosity and Saybolt universal viscosity is

$$\text{Kin.visc., centistokes} = 0.226t - (195/t) \qquad (12\text{-}6)$$

when t = 100 sec Saybolt universal or less

$$\text{Kin.visc, centistokes} = 0.220t - (135/t) \qquad (12\text{-}7)$$

when t = 100 sec Saybolt universal or more.

Actually, kinematic viscosity is computed from Poiseuille's flow equation corrected for loss of kinetic energy. Thus

$$\text{Kin.visc.} = \frac{\pi g r^4 h t}{8lV} - \frac{mV}{8\pi l t} = kt - \frac{k'}{t} \qquad (12\text{-}8)$$

in which r and l are the radius and length of the capillary tube, cm; h is the head causing flow, cm; and V/t is the flow rate, cm³/sec. Viscometers are designed usually to make the *KE* correction term negligible.

The viscosity of lubricating oil has a tendency of becoming high when temperature becomes low. This tendency is expressed by means of the Viscosity Index. This is explained in detail in the next section in this lesson.

(5) Carbon residue

This is determined by evaporating, under specified test conditions, a known weight of oil and weighing the residue. It is not a sure indication of the relative carbon deposits that may form in an engine. Paraffin-base oils usually show a higher Conradson carbon value than do the asphalt-base oils.

(6) Oxidation Stability

During combustion, the sulfur content in the fuel is converted into the sulfuric acid which mixes up with oil. And the engine oil, exposed to the high temperature of cylinder and piston, is oxidized in contact with the oxigen in combustion. The oxidation of the oil not only produces a number of acids which corrode bearings, liners, ring and other internal components of the engine, but also brings the asphalt-like chemical compound through other reaction that accelerates the production of sludge.

Therefore, an oxidation inhibitor is added to the high quality lubricating oil to decrease the oxidation to the minimum. This is called "premium type engine oil".

(7) Cleaning Property

It is necessary for engine oil to be provided with the property which prevents the combustion products and the oxides of engine oil, both produced during the operation of the engine, from depositing at each part of the engine, and diffuses the insoluble matter, created by the deterioration of the oil, into very fine grains. Such a property is called the cleaning property. The engine oil to which a cleaning and diffusing agent is added is specified as "heavy duty type engine oil".

12.4.3 Viscosity index

(1) Idea of viscosity index

The viscosity represents the fluidity of oil. Generally, viscosity decreases at high temperature, and increases at low temperature.

The viscosity index represents the viscosity variation against the variation of temperature. The oil with a high viscosity index means its viscosity varies less against the variation of temperature. The relation between the viscosity and the temperature is shown in Fig. 12-20.

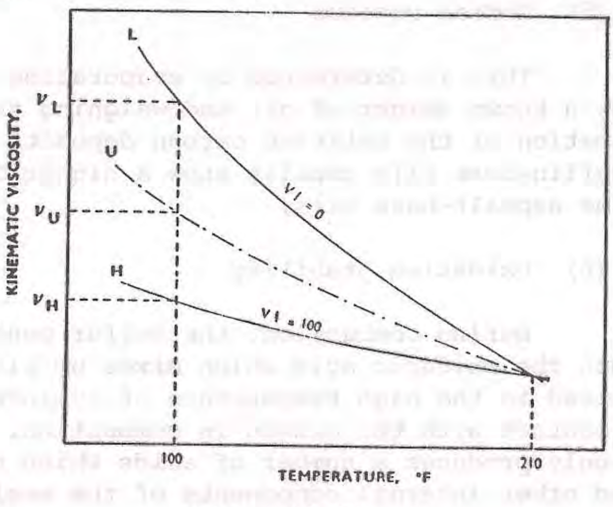


Fig. 12-20

(H) is the ideal case where the axis of abscissa represents the temperature. There are, however, few such oils. Most oils show properties as indicated in (U) or (L). Here (L) means that viscosity varies greatly against the change of temperature. (U), compared with (L), varies less in viscosity against the change of temperature, which means it has a high viscosity index. It can be said that the oil having a high viscosity index is excellent as engine oil, because it not only requires less power to start the engine, but also makes the start of the engine easy at low temperature.

(2) Viscosity index in reality

The viscosity index is a measure of the change of viscosity with temperature of an oil compared with two reference oils having the same viscosity at 210F, one of naphthenic base and the other of paraffinic base. If L, H, and x are the viscosities in Saybolt universal seconds at 100 F for the naphthenic, paraffinic, and unknown oils, respectively.

$$\text{Viscosity Index (VI)} = 100(L - x)/(L - H) \qquad (12\text{-}9)$$

Thus, the naphthenic reference oil has a VI of 0, whereas the paraffinic reference oil has a VI of 100. Evaluation of VI is facilitated by the use of a diagram (Fig.12-2) based on the foregoing relation and reference oils.

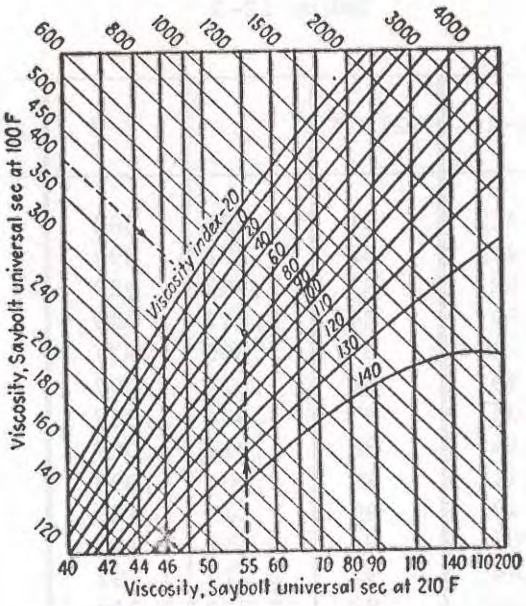


Fig. 12-21

How to determine the VI of lubricant using the diagram in Fig. 12-21? Suppose that the viscosity of a lubricant is 400 and 55 Saybolt universal second at 100 and 210°F, respectively. The intersection of two dotted lines representing the given viscosities occurs at a VI of 80.

(3) VI calculation by table.

Table 12-3 shows the viscosities in Saybolt universal second. There are two oils which have the same viscosity in Saybolt universal second at 210°F. One is naphthenic oil whose viscosity at 100°F is L, and the other is paraffinic oil whose viscosity at 100°F is H. This table show the difference of viscosity of these two oils at 100°F.

For example, when we want to know the VI of some oil whose viscosities at 210°F and 100°F are 60 and 600, respectively, we can easily find out that L and (L-H) are 781 and 355, from number 60 which shows the viscosity at 210°F in Table 12-3.

$$\text{Viscosity Index (VI)} = 100(L - x)/(L - H)$$

$$= 100\ (781-600)/355$$

$$\doteqdot 51.0$$

Table 12-3

S210°F	L	L-H	S210°F	L	L-H	S210°F	L	L-H
40	138	31	81	1,674	864	122	3,966	2,301
41	161	41	82	1,721	892	123	4,031	2,343
42	185	52	83	1,769	920	124	4,097	1,387
43	210	63	84	1,817	949	125	4,163	2,430
44	237	76	85	1,865	977	126	4,229	2,470
45	265	89	86	1,914	1,007	127	4,296	2,517
46	293	102	87	1,964	1,037	128	4,363	2,561
47	322	116	88	2,014	1,067	129	4,430	2,605
48	353	131	89	2,064	1,098	130	4,498	2,650
49	386	147	90	2,115	1,129	131	4,567	2,696
50	422	165	91	2,116	1,160	132	4,636	2,742
51	456	184	92	2,217	1,191	133	4,705	2,787
52	491	203	93	2,270	1,224	134	4,755	2,834
53	525	220	94	2,322	1,256	135	4,845	2,880
54	561	239	95	2,375	1,288	136	4,915	2,927
55	596	259	96	2,428	1,321	137	4,986	2,974
56	632	276	97	2,481	1,353	138	5,058	3,022
57	669	295	98	2,536	1,388	139	5,130	3,070
58	706	315	99	2,591	1,423	140	5,202	3,118
59	743	335	100	2,646	1,457	141	5,275	3,167
60	781	355	101	2,701	1,491	142	5,348	3,216
61	819	376	102	2,757	1,526	143	5,422	3,266
62	857	396	103	2,814	1,562	144	5,496	3,316
63	897	419	104	2,870	1,597	145	5,570	3,365
64	936	440	105	2,928	1,634	146	5,645	3,416
65	976	462	106	2,985	1,670	147	5,721	3,467
66	1,016	484	107	3,043	1,706	148	5,796	3,518
67	1,057	507	108	3,102	1,744	149	5,873	3,570
68	1,098	530	109	3,161	1,782	150	5,949	3,621
69	1,140	555	110	3,220	1,819	151	6,020	3,673
70	1,182	578	111	3,280	1,858	152	6,104	3,726
71	1,225	602	112	3,340	1,896	153	6,182	3,279
72	1,268	627	113	3,400	1,934	154	6,260	3,832
73	1,311	651	114	3,462	1,974	155	6,339	3,886
74	1,355	677	115	3,524	2,014	156	6,418	3,940
75	1,939	702	116	3,585	2,053	157	6,498	3,995
76	1,444	728	117	3,648	2,094	158	6,578	4,049
77	1,489	755	118	3,711	2,135	159	6,659	4,105
78	1,534	781	119	3,774	2,171	160	6,740	4,160
79	1,580	808	120	3,838	2,218			
80	1,627	836	121	3,902	2,259			

(4) Low and high VI oils

Additives have been developed that improve the VI of engine oils until now lubricants are available with VI values ranging from about 0 to well over 100. Three oils having the same low-temperature viscosity for equal cold starting characteristics may have the minimum desirable viscosity at different operating temperatures. Thus, the low VI oils are as satisfactory as high VI oils from a viscosity standpoint, under milder operating conditions. The low VI oils might be unsatisfactory at severe operating conditions because of lower viscosity.

The development and use of VI improvers makes possible the double branding of a given oil which has a viscosity in the upper part of the range at a low temperature and in the lower part of the range at a high temperature for the minimum VI. This indicates the possibility of an all-season motor oil such as an SAE 10W-20 oil to satisfy viscosity requirements.

Synthetic (Prestone) motor oils with VI values above 140 have been developed, and tests in internal-combustion engines indicate that these lubricants are satisfactory. These oils are synthesized from hydrocarbon gases.

12.4.4 Additives for engine lubricating oil

Additives which lower the pour point of lubricants are used to obtain oil flow in engines under low-temperature starting conditions. These additives apparently prevent the formation of wax at low temperatures.

(1) Viscosity index improver

The range in temperature from cold starting to normal operating temperature results in a large decrease in viscosity of the lubricant. This change reduces the film thickness in bearings and may result in bearing surfaces breaking through the thin film under high-load conditions. Additives are used which reduce the change in viscosity with increase in temperature so that an oil may be more satisfactory under both cold-starting and high-load operating conditions in so far as viscosity effects are concerned.

(2) Antioxidation additives

Engine operation subjects the lubricating oil to high temperatures, particularly at the upper part of the piston-ring belt and under the piston head, and results in oxidation and decomposition of the oil. Oxygen is present in the blowby gases, which consist of unburned charge and products compressed in the clearance spaces between the piston and rings and the cylinder walls. Also, the gases in the crankcase usually contain an appreciable amount of air. Thus, oil oxidation occurs at varying rates depending principally on operating conditions, and in combination with unburned products it produces a deposit which may be baked to a hardness in the ring grooves that will cause ring sticking and plugging of oil slots. In mixture with water vapor the oxidation products form a sludge that may clog the screen and other parts of the lubrication system as well as form a soft sludge deposit on the various surfaces the oil contacts.

Antioxidation additives are used to decrease the oxidation of the oil. These additives apparently have a greater affinity for the oxygen than does the oil. Obviously, the longer the oil is used, the lower its resistance to oxidation becomes, since the additive's capacity for oxygen decreases as it reacts with oxygen.

(3) Detergents

The addition of *detergents* to lubricating oils improves the tendency of the oil to wash or cleanse the surface where oxidation products form, thereby resulting in a marked reduction in ring sticking, particularly in heavy-duty service, and a marked improvement in cleanliness of pistons. Detergent oils usually disperse or keep the oxidation products in suspension and reduce or prevent the accumulation of sludge on the various engine surfaces. Thus, these oils usually become discolored quickly because of the cleaning and dispersion effects.

(4) Anticorrosion additives

These additives reduce or prevent the chemical action of acids (fromed by the oxidation of the oil) which destroy some bearing materials, such as lead in the lead-copper bearings, used for crankshafts and connecting rods. These additives usually coat the bearing surfaces and protect the metal from attack by the acids.

(5) Other common lubricant additives

During the past 30 years various types of lubricant or oil additives have been developed. Those commercially available today are classified, according to their function, as in Table 12-4.

Table 12-4

Additive used	*Purpose of additive*
Oxidation-inhibitor	Increases oil and machine life; decreases varnish and sludge on metal parts
Corrosion inhibitor	Protects against chemical attack of alloy bearings and metal surfaces
Antiwear improvers	Protect rubbing surfaces operating with thin films; boundary lubrication
Detergent	Cleanliness of lubricated surfaces
Dispersant	Keeps insoluble combustion and oxidation products in suspension and dispersed
Alkaline agent	Neutralizes acid from oxidation of oil so it cannot react with oil or engine
Rust inhibitor	Eliminates rusting in presence of water or moisture
Pour depressant	Lowers low-temperature fluidity
Viscosity improver (VI)	Lowers rate of change of viscosity with temperature change
Oiliness agent	Reduces friction, seizure, wear; increases lubricity
Extreme pressure (EP)	Increases film strength and load-carrying capacity
Antifoam agent	Prevents stable foam formation
Tackiness agent	For greater cohesion, nondrip property
Emulsifier	Reduces interfacial tension so oil can disperse in water
Fatty oils	For greater wetting for moisture conditions
Solid lubricants (fillers)	Withstand high temperatures and/or pressures
Thickening agent	Converts oil into a solid or semisolid lubricant
Water repellents	Impart water-resistant properties to grease or other components of lubricants
Metal deactivators	Pacify, prevent, or counteract catalytic effect of metals
Silver pacifier	Noncorrosive to silver bearings
Color stabilizer	Standardizes desirable color and prevents formation of undesirable color
Odor-control agent	Provides distinctive or pleasant odor or masks undesirable odors
Antiseptic	Prevents emulsion breakdown or odor from growth of bacteria

12.5 Lubricating Oil for Diesel Engine

12.5.1 Recommendation from engine producer

Each model of diesel engine has a suitable lubricating oil, which was used when the engine was developed and tested. Other oils may serve well but for best result one should use the oil recommended by the engine maker. Alternatively, you should use the higher class lubricating oil (in API classification) than the recommended one. Recently, a new engine lubricating oil, called a "highly refined lubricating oil" has become available. This oil is considered to be one of the best of the heavy duty lubricating oils.

Table 12-5 shows some properties of highly refined lubricating oil for marine diesel engine which is recommended by an engine producer.

Table 12-5

Item \ SAE No.		20 W	30	40
Specific gravity 15/4°C		0.899	0.903	0.908
Flash point	(°C)	230	240	250
Viscosity at 100°F	(CST)	66	117	169
Viscosity at 200°F	(CST)	8	12	14
Viscosity index (VI)		95	94	91
Pour point	(°C)	-20	-15	-15
Alkalinity number	(mg KOH/g)	8	8	8

12.5.2 Highly refined oil and starting ability

Highly refined lubricating oil has a high VI (viscosity-index number), it means that this oil has rather high fluidity in low temperatures.

This lubricating oil makes it easy to start the engine manually, by cell motor or pneumatically. When starting an engine, the temperature of the lubricating oil in the engine usually falls to the atmospheric temperature. So, if a common lubricant is used, the engine is difficult to start because of an increase in oil viscosity. Some types of oil are composed of the highly refined base oil, which shows less increase of viscosity even at low temperature. This allows an easy engine start. For reference, the relation between the viscosity and the temperature of lubricating oil is given in Fig. 12-22.

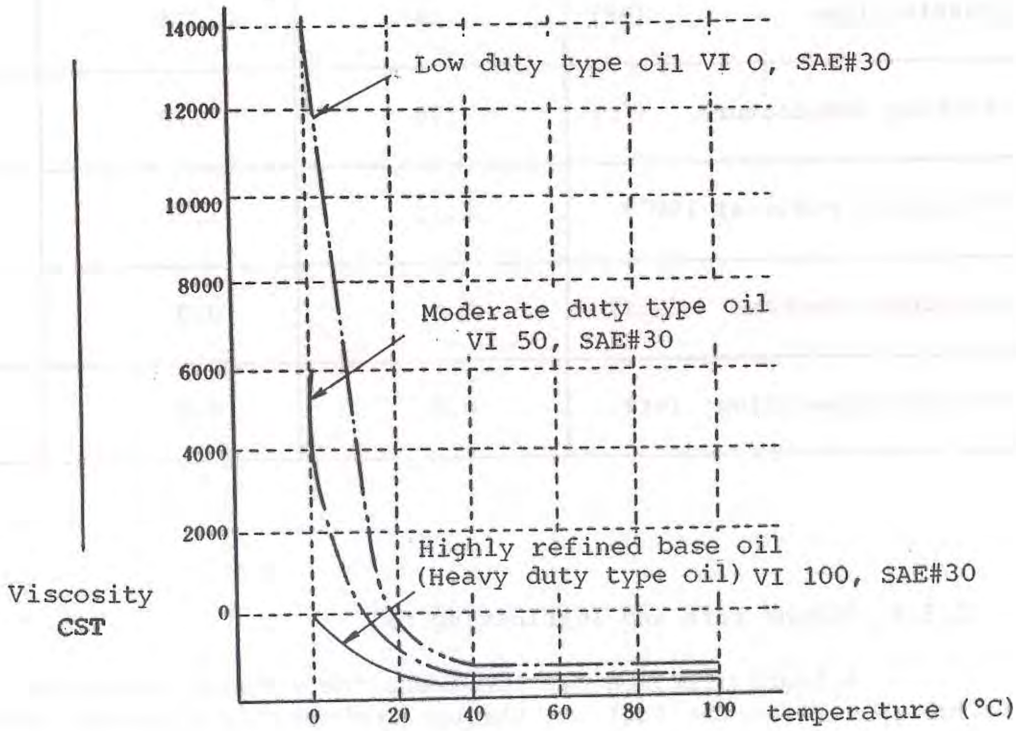


Fig. 12-22

12.5.3 Quality of lubricant and its life

Highly refined base oil is expensive, but it is an economical engine oil as it is durable for the long use. Being highly refined, it shows far less deterioration, and is of a higher cleaning faculty and durable for a long time of use, compared with other heavy duty engine oils on the market. The results of the Indiana Oxidation Test applied to several types of engine oils is shown in the Table 12-6.

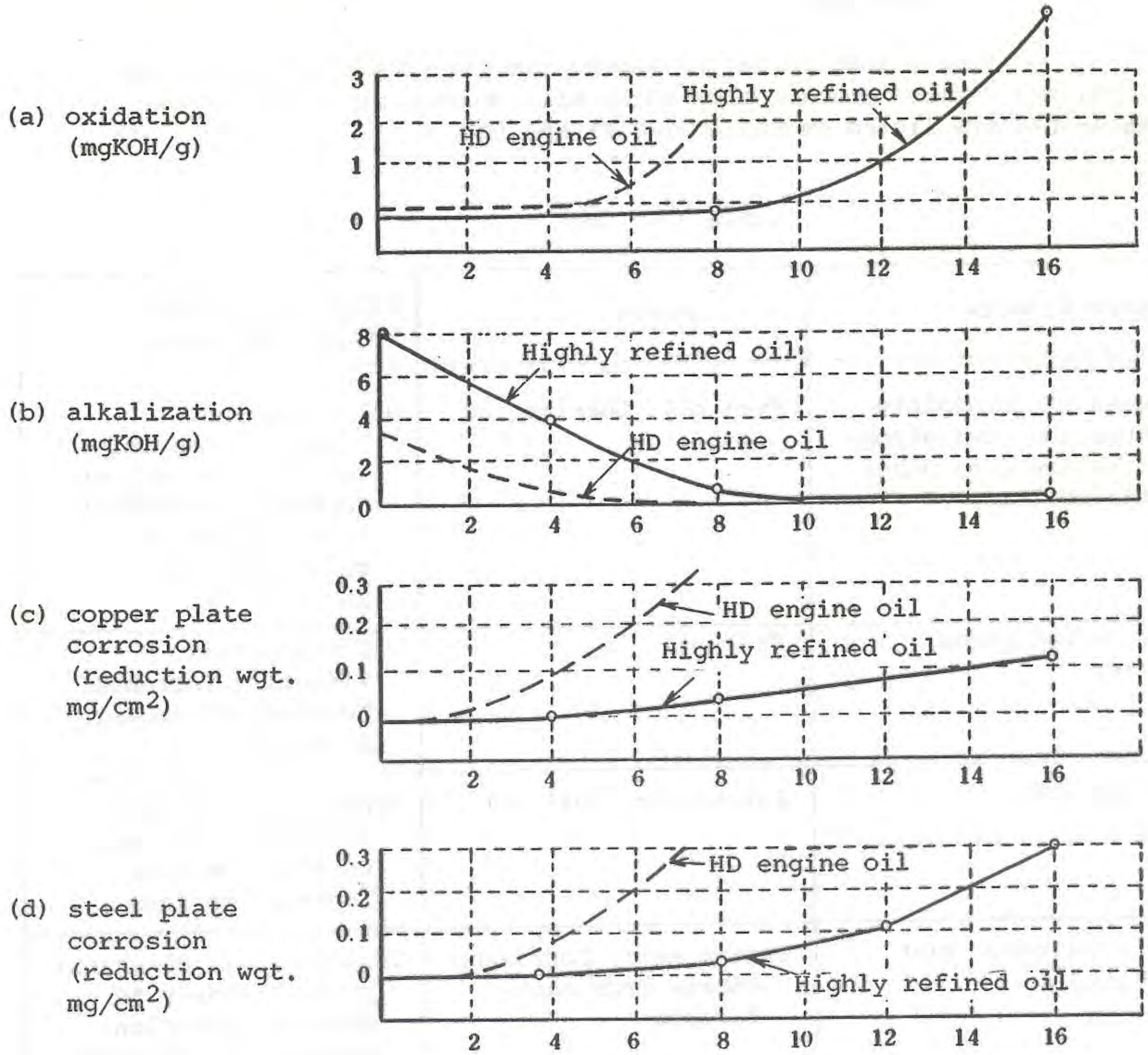
Table 12-6

Item \ Oil		Highly refined oil	HD type oil on the market	Premium type oil on the market
	Testing time (hr)	24	24	24
	Testing temperature (°C)	170	170	170
oil condition after testing	Viscosity ratio at 100°F	1.15	1.30	1.33
	Insoluble heptone (wt%)	0.0	0.9	1.3
	Varnish-deposition (wt%)	0.0	0.0	4.6

12.5.4 Engine life and lubricating oil

A highly refined oil lengthens the life of the engine. The sulfur content in the fuel is, through combustion, converted into the surfuric acid, which shortens considerably the life of the engine. This tendency is greatly increased when using heavy oil, and "milky spot" corrosions is produced on chromium-plated liners.

A highly refined oil has three times greater ability to make sulfuric acid harmless than any conventional heavy duty engine oil on the market. It therefore lengthens the life of an engine, even in the case of engine with chromium-plated liners. There is a testing method to evaluate the capability of an oil to neutralize the sulfuric acid produced by the combustion of sulfur. By this evaluation method, sulfuric acid is added to a highly refined oil and to other H.D. type engine oil to examine the production of oxidizing product (corrosive acid) and their corrosive rate in metal plate. The result of this test is shown in Fig. 12-23. (Testing conditions: temperature 50°C and reacting time 1 hr).



Ordinate shows the quantity of sulfuric acid added to oil: (mgH_2SO_4/g)

Fig. 12-23

12.6 Replacement Time for Lubricating Oil

12.6.1 Factors affecting oil durability

The lubricating oil should be replaced in accordance with the oil manufacturer's instructions and engine operation manual. The time for replacement, however, varies according to the following three factors:

a) Class of lubricant regular, premium or heavy duty (in API)

b) Type of fuel (light oil or a heavy oil, weight % of sulfur content)

c) Work load of engine (severe, ordinary or light working condition)

After a long period of engine running, the lubricating oil will collect various contaminants which will shorten its life. These contaminants are listed in Table 12-7(a) and (b).

Table 12-7 (a)

Contaminants	Source	Effects on diesel engine lubricant
Products of incomplete combustion. Oxidation, polymerization. Other deterioration.	Fuel oil. Lubricating oil.	Oil darkening. Viscosity increase. Tendency to lacquer formation--sedimentation. Oxidation susceptibility increases.
Sooty carbonaceous matter.	Fuel oil	Oil blackening. Viscosity increase. Tendency to deposit formation.
Dust and dirt.	Intake air, fuel oil	Tendency to deposit formation. Tendency to abrasive wear. Tendency to foam.
Metal particles. Rust and scale.	Engine wear. Corrosion. Debris from manufacture.	Catalyzes oil deterioration. Tendency to deposit formation. Tendency to abrasive wear.
Fuel oil	Leaks (pump, piping). Injection dribble. Blow-by Incomplete combustion.	Oil stability lowered. Oil viscosity reduced. Tendency to lacquer formation.
Water	Condensation of combustion gases. Leaky jackets. Leaky cooler.	Oil stability lowered. Tendency to form sludge. Tendency to rust and corrode metal.
Acids	Blow-by of combustion gases--especially from high sulfur fuel	Tendency to corrode metal. Oil stability lowered. Tendency to lacquer formation.

Table 12-7 (b)

Contaminants	Source	Effect on gasoline engine lubricant
Products of incomplete combustion. Oxidation, polymerization. Other deterioration.	Gasoline. Lubricating oil	Gray discoloration due to lead from fuel. Tendency to deposit formation. Tendency to lacquer formation. Viscosity increase. Oxidation susceptibility increases
Carbon soot	Gasoline	Oil blackening. Tendency to deposit formation. Viscosity increases.
Road dust	Intake air. Air cleaner and crankcase breather.	Tendency to abrasive wear. Tendency to deposit formation.
Metal particles	Engine wear. Rust scale. Engine debris from manufacture or overhaul.	Tendency to deposit formation. Catalyzes oil deterioration. Tendency to abrasive wear.
Dilution	Fuel. Frequent starts, cold idling, excessive choking.	Viscosity reduced. Oil stability lowered. Tendency to lacquer formation.
Water and antifreeze materials	Condensation. Head gasket leaks.	Tendency to lacquer formation (permanent-type anti-freeze). Tendency to rust and corrode metal parts Tendency to emulsify. Oxidation susceptibility increases. Tendency to form deposits.
Acids	Combustion gases.	Tendency to corrode metals. Tendency to sludge formation. Tendency to lacquer formation.

Table 12-8 lists typical tests, results and interpretation of used-oil condition from an automotive engine and a large diesel engine.

Table 12-8

	Sample A: Detergent		Sample B: Nondetergent	
	new	After use in 6-cylinder gasoline engine	New	After use in 2,400-hp diesel 2-stroke cycle
Use period	0	2,300 miles	0	2,100 hr
Viscosity at 210°F, SUS	68	68	62	59
Flash point °F	....	....	420	380
Dilution %	Nil	4.0	Nil	2-4
Neutralization No	....	*	0.05	0.56
Pentane insoluble %	Nil	1.9	Nil	0.2
Benzene insoluble %	Nil	1.7	Nil	
Water %	Nil	Nil	Nil	Nil
Ash %	0.60	0.93	Nil	0.02

* Neutral number not significant with detergent oil.

Sample A has reached the acceptable limit with respect to insolubles and ash-forming contents. Most of the insoluble contaminants are lead compounds derived from fuel and sooty matter from the combustion chamber. Some fuel dilution is present. Oil viscosity appears normal, however. This is due to the compensating effect of fuel dilution and insolubles contamination. The former tends to thin the oil, while the latter thickens it. Oil in the condition shown by Sample A is unsatisfactory for use in an engine and drainage is advisable.

Sample B with its low flash point indicates that some fuel dilution is present, reducing viscosity somewhat. From viscosity decrease, the amount of dilution is estimated at 2 to 4 per cent. The results of other tests are satisfactory. Neutralization number is within acceptable limits, indicating that no excessive amount of internal change has occurred in the oil. No water is present. Insolubles and ash contents are satisfactorily low, the latter showing a small amount of iron from normal wear of engine parts. Based on these results, this oil is considered suitable for further service.

12.6.2 Used oil inspection

Used oil inspections can be useful in the engine maintenance program. Changes in used oil inspections can give early indications of problems or malfunctions in the engine. It is not technically feasible to set forth hard-and-fast rules or limits on used oil inspections. The most useful information is obtained from sudden deviations from used oil inspection patterns which have been previously established for an engine in a given service with a specific oil. Table 12-9 indicates the kind of information that can be obtained from used oil inspections. Reliable oil companies are always ready to discuss the significance and interpretation of used oil inspections as they apply to a particular oil. In addition they may be able to perform more sophisticated tests in cases of actual or suspected trouble.

Used oil analyses are of little use in comparing oils. This is because oils differ widely in type of additives used, and each additive system gives a different pattern of used oil inspections. What may indicate good performance with one oil can mean poor performance with another oil.

Appearance of the oil is no longer a good guide to the further usefulness of modern oils. Many oils are highly dispersant and will darken very quickly with use. The pentane isolubels test (ASTM D893 and D893B) is often substantially in error on such oils, which are designed to suspend large amounts of material and still perform well.

Table 12-9

Test	Significance
Viscosity change (ASTM D88 or D445)	A decrease can indicate fuel dilution An increase can indicate oil oxidation or insolubles
Flash point (ASTM D92)	Drop in flash point can indicate fuel dilution
Fuel dilution (ASTM D322)	Indicates fuel dilution
pH (ASTM D664)	Indicates corrosive acids from oil oxidation or condensation from blowby
Total acid No. (TAN) Total base No. (TBN) (ASTM D664)	Of limited significance

Table 12-9 (Cont'd)

Test	Significance
Water (ASTM D95)	Cooling-system leaks or low-temperature operation. Water from blowby accumulates at sump temperatures below 110°F
Pentane insolubles (ASTM D893)	Indicates build-up of sludge, soot, carbon, or resins in oil
Benzene insoluble (ASTM D893)	In diesel engines shows carbon and soot from poor combustion
Elemental analysis	Can be used to show abnormal wear or corrosion or oil contamination

12.6.3 When to replace lubricating oil : the spot test

In order to know the present condition of the system oil being used and to determine whether purification, make-up or total replacement is necessary, a used oil sample must be analyzed. However, ordinary test methods require complex apparatuses and time consuming operations and, therefore, they are not suitable for a quick checkup in the field. For this purpose, "Spot Test" is offered.

(1) Objective

The "Spot Test" can only be used for HD type engine oils. HD engine oil contains detergent-dispersant additives which prevent ring sticking and suspend carbon and sludge from cylinder drip oil in very fine particles, thereby preventing deposit formation. Also engine oil is added with alkalinity to neutralize acidic material formed by combustion, which may cause corrosion of cylinder liner and bearings. Therefore, where HD engine oil is used, detergency-dispersancy and alkalinity should be checked to determine condition of the oil being used.

The additives which impart detergency-dispersancy and alkalinity are gradually consumed during service. The "Spot Test" checks degrees of decreased effectiveness of additives.

(2) Method

The "Spot Test" will give results in a few minutes using a simple apparatus, including test paper, indicator dropper and sample oil dropper. This method adopts paper chromatography techniques. The detergency and dispersancy is judged by appearance of diffusing droplet of the sample put on the filter paper and alkalinity by color change of indicator.

(3) Judgement of contamination and lispersancy

Using a dropper, put a drop of the sample oil on the test paper. Observe the degree of shade of color and diffusion. Figure 12-24 shows two examples representing good and poor dispersancy. When detergent dispersant additive in the engine oil is still active and well maintained, carbon and sludge are suspended in the oil in very fine particles. Therefore, they are carried by the oil which is diffusing on the paper and the edge of the circle is blurred. On the other hand, if the additive is depleted and its dispersive property is lost, carbon and sludge are separated and thereby diffusion of oil drop is obstructed showing a contrasing edge of circle. A darker oil spot shows heavy contamination and lighter spot less contamination.

Fig. 12-24

Figure 12-25 shows examples to judge dispersancy and degree of contamination.

(a) indicates oil condition of less contamination and good dispersive properties. It is inferred that less deterioration of the engine oil, less cylinder liner and ring wear, less blow-by and less contamination of cylinder drip etc. has occured. Furthermore it is an indication that effective purification of the engine oil is being carried out.

(b) indicates oil having still active dispersancy but relatively heavy contamination. When contamination of cylinder drip is excessive or the operating time is prolonged, the contaminants are accumulated to show this appearance. To prevent sharp decrease of effectiveness of additive and to remove contaminants, it is recommended to prolong purification period and/or to use a fine mesh filter.

(c) indicates that the oil has lost most of its dispersancy but is less contaminated. This may be interpreted that the system oil is being purified thoroughly. It is also an indication of the additive depletion that the system oil has been used for a fairy long period of time. It may be considered an example of a carefully controlled lubrication. When this condition is first observed, oil drain or partial replacement with new oil should be considered. However, it is also permissible to continuously use the present oil without draining or make up if the system oil is being maintained under the similar lubrication control.

(d) indicates that oil is severely contaminated and has completely lost dispersancy. Oil contains more contaminants, such as carbonaceous matter and sludge. Also deterioration of oil has progressed considerably. By reason of complete depletion of additive and also reaching limit of use, the system oil should be drained. Generally, color of the ring outside the black spot turns darker as oil deterioration proceeds.

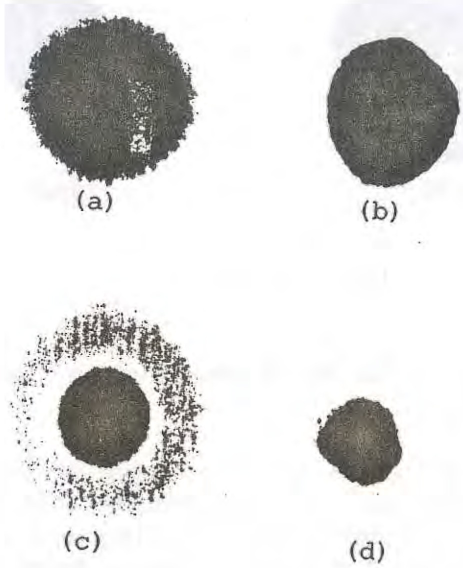


Fig. 12-25

(4) Judgement of Alkalinity

Put a few drops of indicator using a dropper, on the test paper and wait for 10-30 seconds until indicator permeates the paper. Then put a drop of the sample oil on it. In 1-2 minutes, sample oil diffuses and color of the outer ring of black spot turns blue depending on alkalinity of the oil. If color of the indicator does not turn to blue, it means that the oil has lost alkalinity and is neutral or acidic. When alkalinity is lost, the outer ring will appear to be yellow or yellowish-brown but gradation of color does not necessarily relate to the degree of acidity.

Figure 12-26 indicates color changes against alkalinity.

(a) and (b): Alkalinity still remains in the oil and effective detergent-dispersant and acid neutralization properties can be expected, but in (c) it disappears completely. Appearance of (b) may be considered to be the limit for further use. In such a case, whether partial renewal or complete drain should be done, depends on the degree of contamination in oil, the vessel's operation schedule, etc.

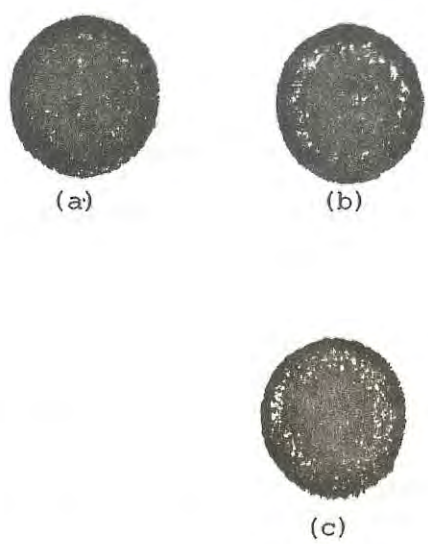


Fig. 12-26

(5) Others

Contamination of oil also includes fuel dilution and the presence of foreign matter, such as dust and metal particles. However, such contamination cannot be detected by the "Spot Test" but by analytical methods. Water contamination may not be detected precisely, but it can be estimated by a peculiar phenomenon.

When an oil sample contaminated with water is tested by the "Spot Test", the edge of back circle becomes very black by contrast and there is no diffusion outside of the circle. Also the center part of the circle becomes lighter in color. This phenomenon may be explained by the fact that emulsified oil, by contamination of water and agglomeration of carbon particles by water contamination, obstructs diffusion of the oil.

(6) Conclusion

Although the "Spot Test" is a simple test method, its results correlate with analytical test results and will indicate oil conditions fairly accurately. It is a very effective method to determine periodically the degree of contamination and/or deterioration of the used oil.

12.7 Quantity of Lubricating Oil

The volume of the lubricating oil supplied to the crank case should be checked by the oil level dip gauge.

a) Less oil .... insufficient lubrication, causes sticks or seizures.

b) Excessive oil .... causes a rise in oil temperature because of the excessive oil supply to piston and liners. Thus, excessive oil is bad for the engine.

Diesel engines for ship are inclined. This inclination varies according to the installation in the ship and the requirements of ship operation. Therefore, if the oil is supplied only judging by the oil level dip gauge, and excessive supply is often made. To avoid excessive supply, pour the oil into the engine after the installation is completed, check the oil level, and record it on the dip gauge. Thus, you will know the correct oil level at any time.

A) Splash Oiling

As shown in Fig. 12-27, the highest oil level must be kept horizontal with the large end of the connecting rod, while the lowest shall be in the position when the oil splasher dips in oil by 5 mm from the oil hole at its end.

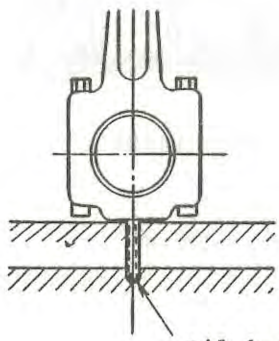


Fig. 12-27

B) Forced Oiling

The highest oil level should be parallel with the large end of the connecting rod. In case of a multi-cylinder engine, the oil level is based on the large end of the connecting rod that takes the lowest position (such a connecting rod is on the clutch side in case of the main engine for ship). The lowest oil level should be kept 20 mm higher than the oil pump inlet and be kept horizontal. Any oscillation, such as rolling and pitching are to be considered, and the supply of oil must be kept sufficient to avoid air being sucked in.

C) In case of the auxiliary engine which is installed keeping its fly-wheel side lower, it is necessary to change the inlet hole of the oil pump by fitting a pipe, as shown in Fig. 12-28.

12.8 Temperature of Lubricating Oil

A) The temperature of lubricating oil is greatly influenced by the temperature of cooling water, that of atmosphere and the work load of the engine, and it differs according to the season. The temperature of oil is usually 40-60°C in winter and 50-70°C in summer. And is higher in small engines than in large ones. Generally, 40°C is considered the lowest and 70°C as the highest temperature.

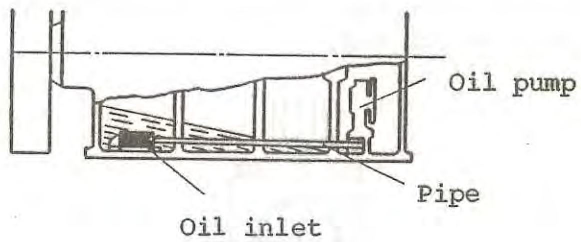


Fig. 12-28

B) The heat of the piston escapes to the cooling water through rings. In addition, it is carried away by the lubricating oil. This is the greatest factor that raises the temperature of the lubricating oil. In an abnormal case, such as sticking, a great amount of heat is produced which raises the temperature of oil. When the temperature rises over 150°C, the oil film of the lubricating oil is broken.

Although it is not correct to think that high temperature of lubricating oil always causes sticks, it is sure that it causes rapid deterioration. The lubricating oil should therefore be cooled by means of some oil cooler.

C) The maximum permissible temperature in each part of the engine depends upon the material used. The inspection of fishing boat in accordance with Fishing Boat Law specifies the maximum permissible temperature as in the following table:

Position measured	Highest permitted temperature (°C)	
	Spark ignition engine hot bulb engine, low speed 4-cycle diesel engine	2-cycle diesel engine, medium and high speed 4-cycle diesel engine
Main metal	60	70
Crank pin metal	65	75
Piston pin	80	90
Reverse gear housing bearing	50	60
Thrust bearing	60	70

12.9 Additives against Corrosion

All fuel oils contain sulfur which cannot be completely removed on an economical basis. So, as explained in Chapter 11, users of engine fuel must try to choose the fuel which contains the least amount of sulfur (less than 1%).

Sulfur in the fuel changes into sulfurous acid gas. A part of this sulfurous acid gas is converted into sulfuric acid gas. When the combustion is complete in the cylinder of high temperature, a greater part of the sulfuric acid gas goes out through the exhaust hole together with the exhaust gas, leaving no trouble at all. However, a part of the sulfuric acid gas usually combines with water in combustion chamber to produce the sulfuric acid (H_2SO_4) which condenses on the wall of the cylinder liner.

Especially when the cylinder liner is of low temperature, the sulfuric acid is remarkably easy to condense on the cylinder, accelerating its wear, and causes even stripping and surface oxidation corrosion (milky spots) on chromium-plated cylinder linens.

Therefore it is recommended to use the lubricating oil of high alkalinity number. However, its alkalinity is decreased little by little during the engine operation and disappears before long. Further use of the lubricating oil without any neutralization faculty results in corrosion, wear and stripping of the liner and piston rings. So, if you find the alkalinity number decreasing, you should use an additive for increasing the alkalinity number of the lubricating oil. Such an additive available on the market is PRECOA, manufactured by Teikoku Piston Ring Company. PRECOA is the abbreviation of "PREVENTIVE CORROSION ADDITION". The actual test has proven that PRECOA has a great effect in preventing the surface oxidation. Its effect is remarkable when a straight lubricating oil without any additive is used for the engine. But, some attention is required in using PRECOA; I shall explain how to use it according to the instructions given by the producer.

(1) PRECOA should be used as follows:

(A) When the lubricating oil is replaced:

a) Although there is no need for disassembling, completely clean the interior of the engine before the lubricating oil is supplied to the crank case. Let the engine idle for 10-20 minutes with the use of flashing oil or straight oil of low viscosity.

b) Supply the required volume of lubricating oil (even a cheap straight oil is good enough, although a high quality lubricating oil is better to the crank case).

c) Add the required volume of "PRECOA F" to the lubricating oil in the crank case. (It is recommended to add PRECOA F after it has been mixed with a small amount of lubricating oil).

d) 10-20 minutes of idling uniformly diffuses "PRECOA F", producing the alkaline lubricating oil which has a cleaning faculty.

e) A further addition of a dose of "PRECOA D" completes the proparations for running.

(B) When the lubricating oil is replenished:

Add PRECOA at the rate of 60g to 1L of the lubricating oil to be replenished.

The addition of PRECOA F to the lubricating oil to be replaced for the crank case of engine should be made at the rate of 60g of "PRECOA F" to 1 L of lubricating oil.

(2) "PRECOA D" is added to supplement the consumed alkalinity in the lubricating oil. The consumption of alkalinity is in proportion to the sulfur contained in the fuel, that is, in the long run to the consumed quantity of the fuel. Therefore it is advisable to add

"PRECOA" every time when a drum or one ton of fuel is consumed corresponding to the horsepower of engine as well as to the conditions of ship operation. In case of the engine which is put in operation under a constant power, it is convenient to add PRECOA D" after regular intervals of time, such as 10 hours, a day or one sailing trip. It can be added, in case of small engine, at the same time when the crank case is supplied with the lubricating oil.

A standard volume of "PRECOA D" to be added can be calculated from the following formula:

Volume of PRECOA D (g) = 0.7 × S × X × Quantity of fuel consumed (kg)
Where, 0.7 is a constant, S is a constant corresponding to the type of fuel (its sulfur content) and X a constant corresponding to the type of engine (production rate of sulfuric acid).

Type of heavy oil for fuel	S
A Heavy oil	1.0
B Heavy oil	2.5

Engine (Dia. of cylinder)	X
110 mm and less	0.35
150 mm and less	0.30

LESSON 13 COOLING SYSTEM

13.1 General

13.1.1 Necessity of cooling

A part of the heat developed during combustion flows from the gases to the cylinder walls, raising their temperature. If the wall temperature is allowed to rise above a certain limit, about 150°C, the oil that lubricates the piston begins to evaporate rapidly, and both piston and cylinder may be damaged. At the same time high local temperatures in certain parts of the engine, e.g., cylinder head and piston, may cause excessive stresses and cracking of these parts. In gas and gasoline engines hot spots in the combustion space may cause preignition.

For all these reasons, since heat cannot be prevented from flowing to the cylinder walls, it must be carried away. The quantity of heat generated in an engine cylinder varies from about 1,800 Kcal to 3,000 Kcal/PS-hr. Tests show that from 25 to 35 per cent of this heat in stationary and automotive and about 12 to 15 per cent in aircooled aircraft engines finds its way into the cylinder walls and must be carried away. This heat is often called *jacket loss*.

13.1.2 Cooling methods and classification

All heat carried away from an engine in the final count is conveyed to the atmosphere. However, the methods of cooling may be divided into two main groups; air and liquid cooling. These two methods differ in construction details and also in operating conditions. The air cooled engines are simple in construction and light in weight. If there is a fault in the engine design, there may be hot spots on some parts of combustion chamber, which will cause precombustion of engine. However, air cooling is applied to many engines, from small gasoline engine for agriculture to large multi-cylinder airplane engines.

There are three methods of cooling by liquid. One of them is sea water cooling method. The sea water, or sometimes river water, is discarded after it cools the engine. Another method is to use a circulating coolant which is cooled by sea water or by air, with the help of some heat exchanging equipment. The third method is to use fresh water, which is mixed with some antirust and antifreeze additive.

- Engine cooling
  - By air
  - By liquid
    - Sea water
    - Circulating coolant cooled by sea water
    - Circulating coolant, cooled by air

13.2 Principle of Heat Transfer

13.2.1 Heat transfer through cylinder wall

(1) Of the three means of heat transfer, conduction, convection, and radiation, in cooling engine cylinders, conduction plays an important part in carrying the heat through the thin layers of stagnant gas and water in contact with the walls. The rest of the heat exchange is done partly by radiation but chiefly by convection.

The heat flow from one fluid to another, separated by a metal wall, can be best explained using the diagram in Fig. 13-1. The temperature t_a of the gas at point a in the interior of the cylinder gradually falls to the value t_b at the surface of the inert gas film. The resistance of this film is very great, a great temperature head, $t_b - t_c$, being required to pass through it by conduction. The temperature head required to cause the flow through the metal wall is $t_c - t_d$. The temperature head required to pass through the outside film is $t_d - t_c$.

Its value is comparatively small if the cooling fluid is water and large if it is air. Finally, the temperature drops to tf of the cooling liquid at some distance away from the wall.

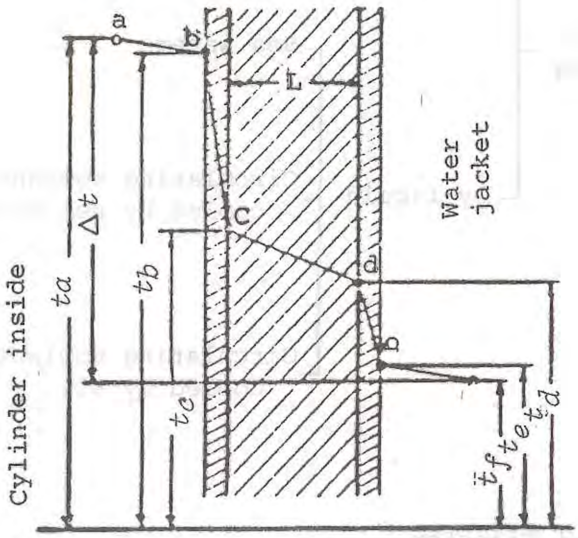


Fig. 13-1

(2) The amount of heat transferred to the walls may be computed from the general expression.

$$Q = H_m A(t_g - t_w)r \qquad (13\text{-}1)$$

where h_m is the mean heat-transfer coefficient, Kcal/m².hr.°C,

A is the area of the walls exposed, m²

t_g is the average temperature of the gases, °C,

t_w is the average temperature of the walls, °C,

r is the time, hr.

The heat exchange between the gases and metal walls is affected by radiation and convection combined with conduction. These factors together with the temperature difference determine the value of the coefficient h_m.

(3) Using the overall heat-transfer coefficient U, instead of h_m, the general equation (13-1) per hour becomes

$$Q = UA(t_1 - t_2) \qquad (13\text{-}2)$$

and U can be presented as

$$U = 1/(1/h_1 + L/k + 1/h_2) \qquad (13\text{-}3)$$

where h_1 is the inside surface coefficient, Kcal/m²·hr·°C,

h_2 is the outside surface coefficient, Kcal/m²·hr·°C,

L is the thickness of cylinder wall, m,

k is the thermal conductivity of the metal, Kcal/m²·hr·°C per meter thickness.

For a cylinder the area A in equation (13-2) is the area corresponding to the mean diameter, $A = 0.5(A_o + A_i)$, where A_o is the area corresponding to the outside diameter and A_i to the inside diameter. The difference in the areas absorbing heat from the hot gases and releasing it to the cooling medium can be taken into account by introducing the weighted values for h_1 and h_2, which changes expression (13-3) to

$$U = \frac{1}{A/(A_i h_1) + L/k + A/(A_0 h_2)} \qquad (13\text{-}4)$$

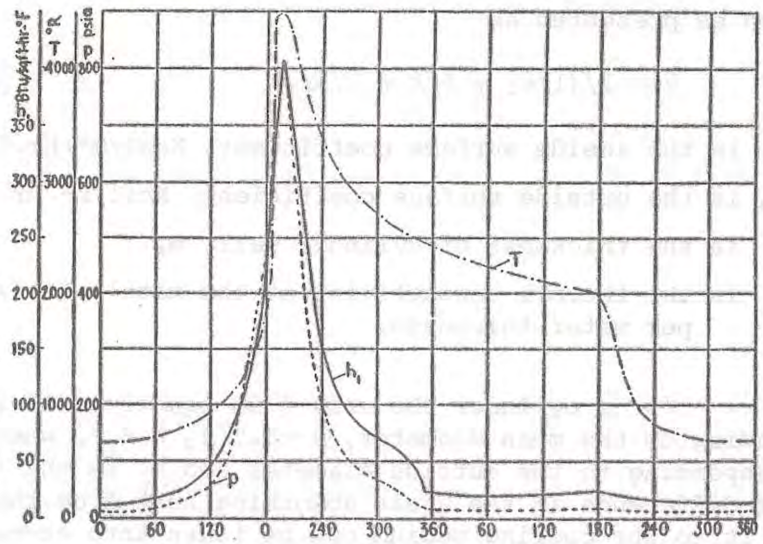
The coefficient h_1 varies considerably during the cycle, because of the change of gas pressure and turbulence. Its value can be computed from the expression (13-5). The mean value of h_1 varies from about 250 to 500 and can be estimated if the rates of heat loss and temperature are known.

(4) The surface coefficient h_1 depends upon the movement of the gas, which is a function of piston speed c, in meters per minute, of the density of the gas which is a function of pressure p, in kilograms per square centimeter, and its temperature $Tg = tg + 273$. According to Nusselt's investigations, the mean value for an engine cycle is

$$h_1 = b(49 + c)\sqrt[3]{p^2 Tg} \qquad (13\text{-}5)$$

where the coefficient $b = 0.00065$ in engines with low turbulence, and may go up to 0.00130 in engines having a high turbulence.

(5) Figure 13-2 gives the values of the gas film h_1 for the whole cycle of an airless-injection compression-ignition oil engine operating at 1600 rpm for full-load conditions. The pressure and temperature curves complete the picture. The mean value of h_1 = 300 Kcal/m²·°C·hr, or h_1 = 62 Btu/sq ft·°F·hr.



Crank angle degrees from top dead center

Fig. 13-2

The value of h_2 depends in the first place on whether the outside surface of the cylinder is cooled directly by air or by a liquid.

For the heat exchange between a metal wall and air, h_2 can be computed from the equation

$$h_2 = mv^{0.89} \qquad (13\text{-}6)$$

where v is velocity of air in meters per minute and the constant $m = 0.14$ in normal conditions and may reach 0.35 if the heat exchange is very efficient, as in automobile radiators.

For the heat exchange between a metal wall and water, h_2 can be determined form the equation

$$h_2 = 290 + b\sqrt{v} \qquad (13\text{-}7)$$

where v is the water velocity in meters per minute and b is a constant. For a quiet flow $b = 260$, sharp turns and sudden changes of velocity in the water jackets, by creating turbulences, may increase the value of b up to 500 and even 750.

(6) The thermal conductivity k of metals can be taken from Table 13-1. As can be seen from expression (13-3), the coefficient of conductivity k affects the overall coefficient U only very little.

Table 13-1

Alloy	Specific heat Kcal/kg·°C	Average conductivity Kcal/m·hr·°C	Coefficient of linear expansion per °C a x 10^6
Aluminum	0.207	149	25.4
Brass	0.090	92	18.7
Bronze	0.102	88	17.6
Gray cast iron	0.130	40	11.7
Monel metal	0.127	22	14.9
Mild steel	0.110	38	11.7

13.2.2 Movement of cylinder temperature

When we calculate the heat transfer coefficient U from expression (13-3), it is not very accurate, about ± 5 per cent.

However, the method of splitting the overall coefficient in three components is valuable for determining the average temperatures t_2 of the inside surface of the cylinder and t_2 of its outside surface.

The gas, which has a temperature t_1, loses to the inside surface of the wall the heat amount

$$Q = h_1 A_i (t_1 - t_2) r \quad (13\text{-}8)$$

which gives

$$t_2 = t_1 - Q/h_1 A_i r \quad (13\text{-}9)$$

The same quantity of heat travels through the wall with a temperature drop of $t_2 - t_3$; therefore

$$Q = (k/L) A (t_2 - t_3) r \quad (13\text{-}10)$$

This gives for the temperature drop

$$t_2 - t_3 = QL/kAr \quad (13\text{-}11)$$

where A is the mean between the inside surface A_i, and outside one A_0. Thus the inside temperature t_2 found by expression 13-9 will permit t_3 to be determined.

Another approach to finding the cylinder-wall temperature is first to find t_3 from the equation

$$Q = h_2 A_0 (t_3 - t_4) r \qquad (13\text{-}12)$$

which gives

$$t_3 = t_4 + Q/h_2 A_0 r \qquad (13\text{-}14)$$

and then to determine t_2 by means of equation (13-11)

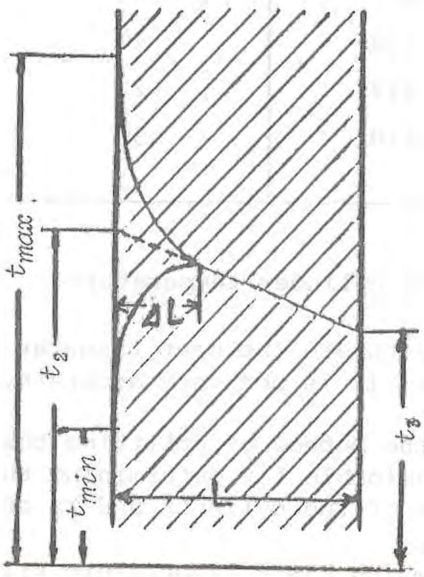


Fig. 13-3

The temperature t_2 of the inside surface of the cylinder wall is not constant during a cycle but fluctuates following the variation of the gas temperature. Figure 13-3 gives a diagram of the temperature distribution through the wall. The temperature of the inner surface goes up to t_{max} during combustion, and drops to t_{min} toward the end of the suction stroke. Eichelberg found for a two-stroke oil engine that at full load the fluctuation above the average value, $t_{max} - t_2$, is about 14°C and below it, $t_2 - t_{min}$, is about 8°C.

In a four-stroke engine the downward fluctuation will be about the same as upward, meaning a temperature range of about 28°C.

The temperature fluctuation does not penetrate deeply; 9 mm from the surface the range of fluctuation is less than 0.5°C.

13.2.3 Heat flow from combustion chamber wall

When the rotative speed of an engine is increased, the duration in seconds of all events, during each cycle, is decreased, but the increased piston speed creates a greater turbulence, slightly increasing U; and as a result the percentage of heat rejected to the jacket slightly increases with the engine speed.

Tests have shown that the percentage of jacket loss is nearly independent of the engine load and slightly decreases with an increase of the cylinder diameter.

Since water-cooled engines constitute the majority, they will be discussed first.

Figure 13-4 shows the distribution of heat flow in four-stroke engines of different stroke-bore ratio l/d. The left ends of the curves, for $l/d < 1$, were drawn to show the general direction of the curves. An average figure of 30 per cent of the total heat generated is assumed as the basis for the distribution.

Figure 13-5 gives a similar picture for two-stroke engines.

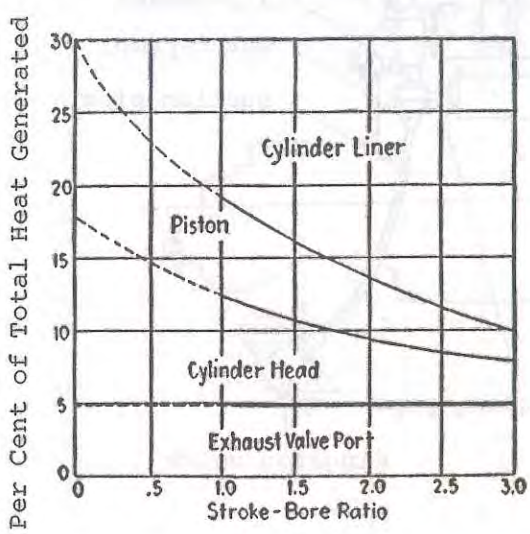


Fig. 13-4

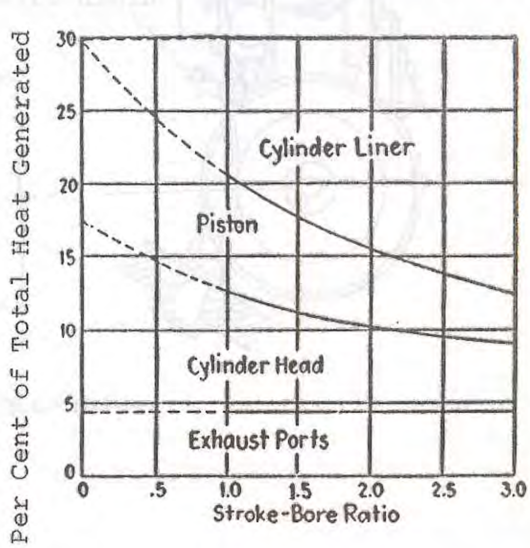


Fig. 13-5

13.3 Cooling Water System

1) Many inner parts of the internal combustion engine are in contact with combustion gas. These parts get hot if there is no cooling. The parts surrounded by high temperature are cooled by water or air from opposite side, and also cooled by lubricating oil directly. We use sea water or fresh water as cooling water. Some types of engine use a circulating coolant; air, fresh water or sea water. Lubricating oil also must be cooled by sea water or fresh water. If the lubricating oil is not cool enough, it may lose its lubricating property because of the high temperature.

2) In most cases the sea water is used as cooling water for the marine engines. As shown in Fig. 13-6, the sea water is sent through the cooling water system in the following order: Kingston valve → suction pipe → water pump → water supply connecting pipe → oil cooler → cylinder → cylinder head → connecting pipe → water adjusting valve → silencer → drain pipe. The plunger type water pump is provided with an air vessel, and thermometers are used at some places (small engines are rarely equipped with oil cooler, or thermometer).

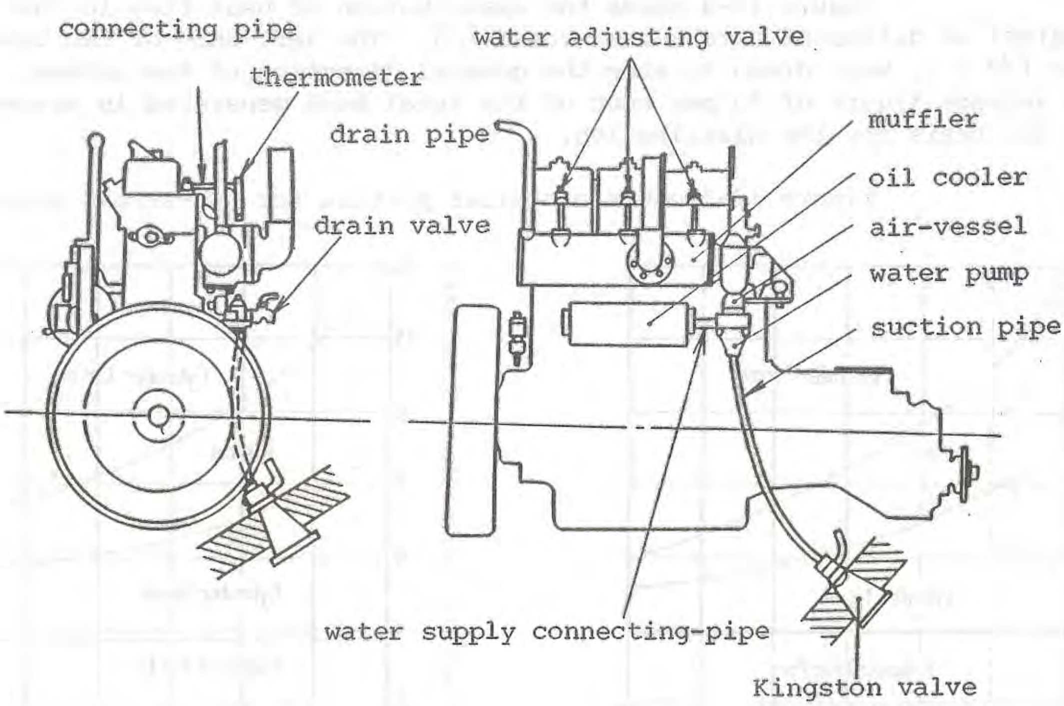


Fig. 13-6

13.4 Cooling Water Equipment for Marine Engines

13.4.1 Kingston valve

The sea water enters through the Kingston valve (Fig.13-7) which is made of gun metal (Cu 80-90, Sn 20-10). The water has to pass through a sieve (filter) at the mouth of Kingston valve, thus preventing refuse and sand from entering and damaging the cooling water pump.

The volume of the sea water entering depends the position of the Kingston valve, as well as the direction toward which it faces. The engine operators frequently complain that the amount of sea water drawn through the Kingston valve is insufficient. In most cases, it has been found that insufficient suction is due to the improper fitting position for the Kingston valve.

A) Figure 13-8 shows the example where the Kingston valve is fitted very near to the sea level. In this case with the rolling of the ship, air is sucked through the Kingston valve while the suction of sea water is reduced.

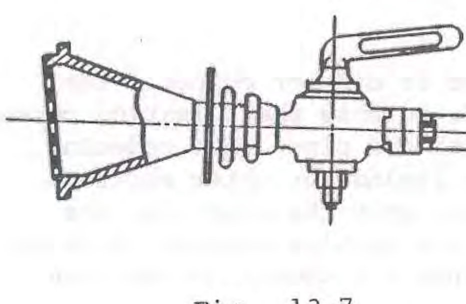


Fig. 13-7

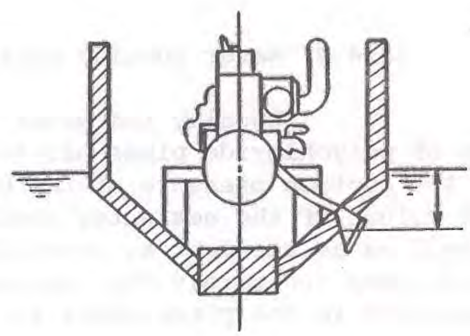


Fig. 13-8

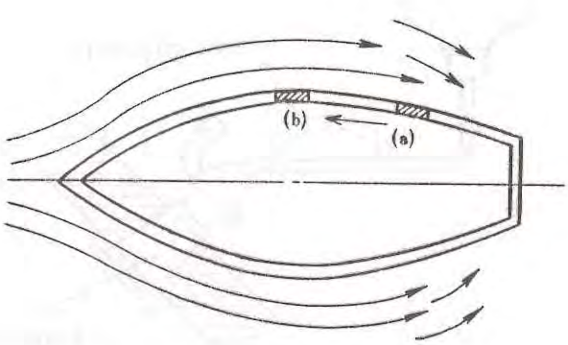


Fig. 13-9

B) It should be avoided to fit the Kingston valve in the position where the eddy current and wake of the ship prevents the sea water from being sucked in, as shown in Fig. 13-9.

C) As shown in Fig. 13-10, the Kingston valve should be fitted facing the stream of water so that the sea water is easily sucked in.

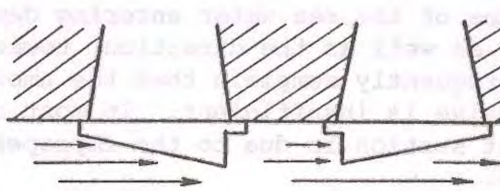


Fig. 13-10 Kingston valve fitting

13.4.2 Water suction pipe

Usually the water suction pipe is made of copper. The use of polychloride pipes has to be avoided, because the pulsation caused by the suction pressure of the pump distorts the pipe, thus reducing the volume of the sea water sucked in. The piping for water suction should be as straight as possible. Depending upon the capacity, the water pump can rarely draw up sea water at its initial suction if it is installed in the place where it is higher than 3 m above the sea level.

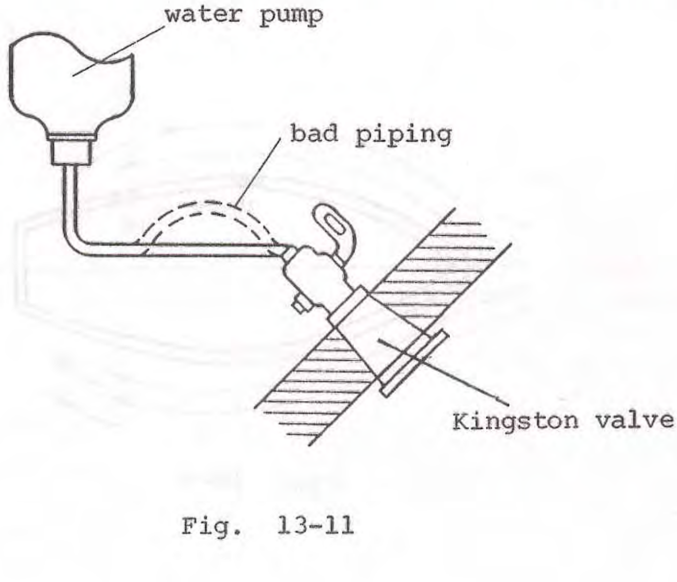


Fig. 13-11

13.4.3 Water pump

There are several kinds of water pumps; centrifugal, rotary plunger, and piston type. For large marine diesel engines the centrifugal pump is used; plunger and piston type have for a long time been used for small and high speed engines. Recently, owing to the development of chemical rubber, the rotary type has been adopted for many engines.

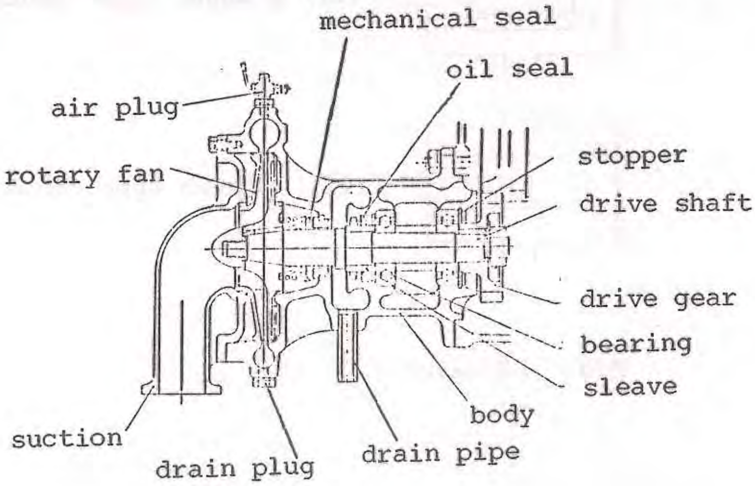


Fig. 13-12(a) centrifugal water pump.

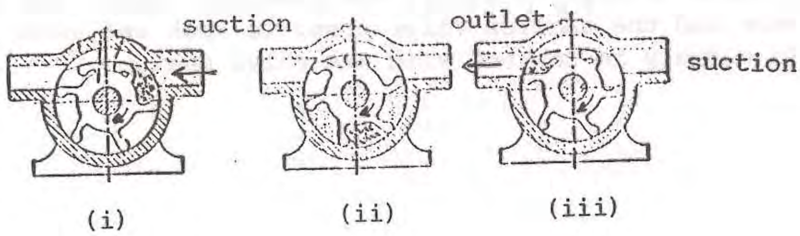


Fig. 13-12(b) rotary water pump

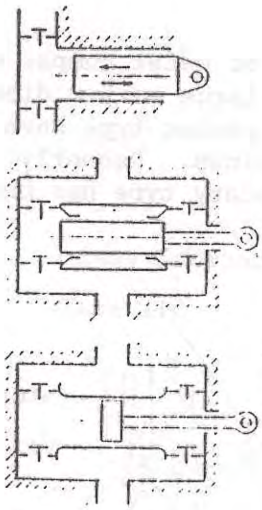


Fig. 13-12(c)

In this lesson, we shall learn about the function, driving method, and discharge volume of plunger type cooling water pump.

A) Function of water pump

When the plunger in the pump body moves toward the arrow mark as shown in Fig. 13-13(b), the pressure inside the water pump falls below zero and the suction valve opens to suck sea water, while the discharge valve is closely in contact with its valve seat.

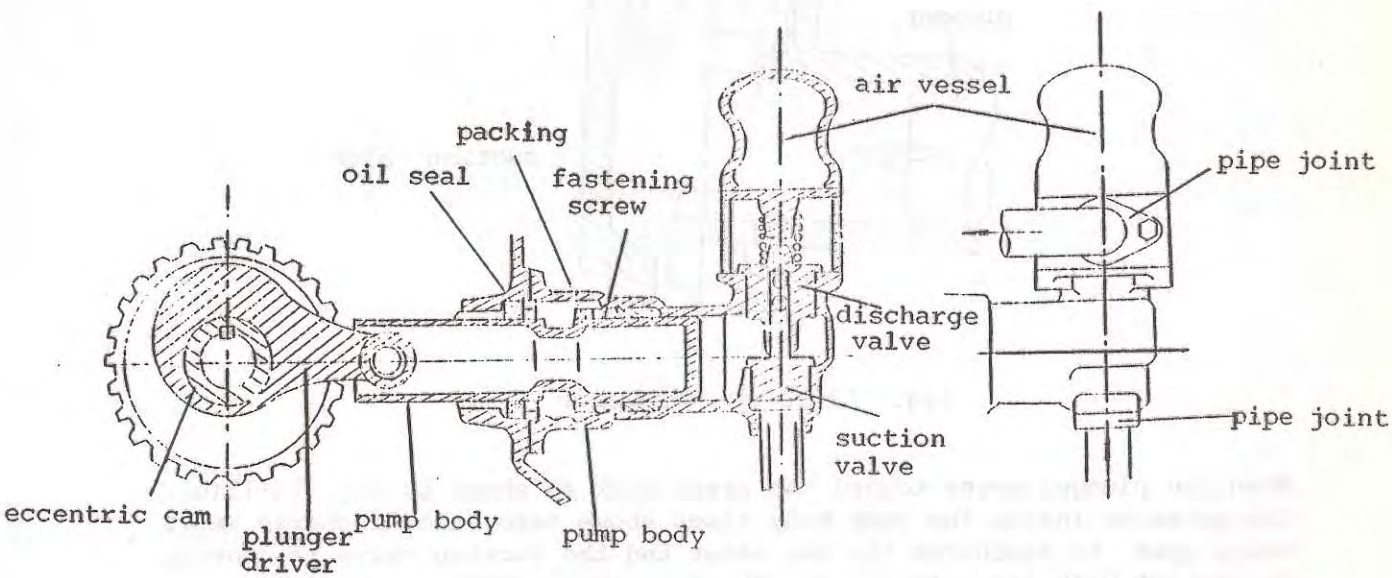


Fig. 13-13(a)

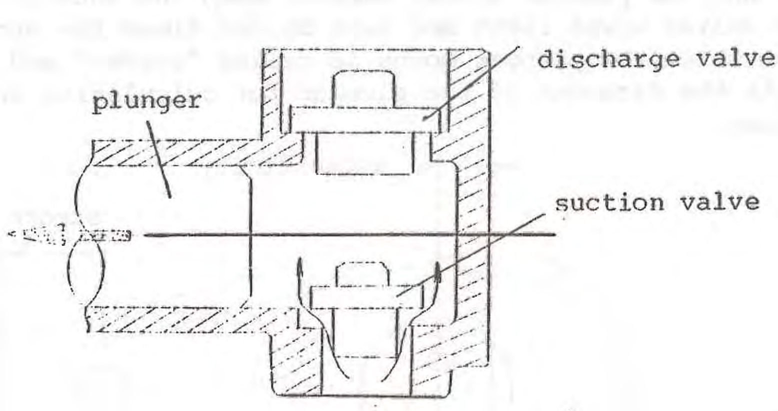


Fig. 13-13(b) suction

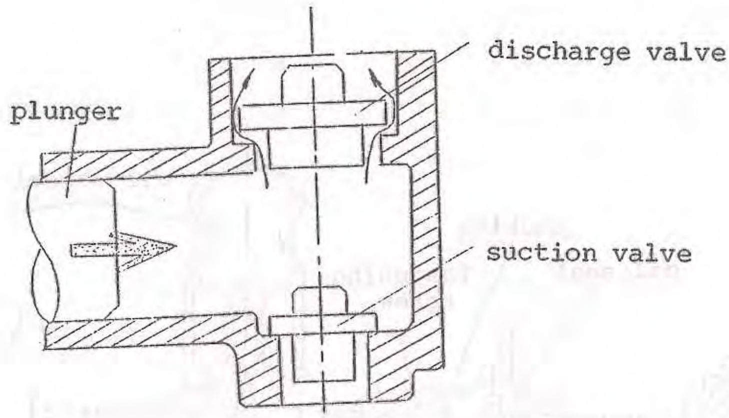


Fig. 13-13(c) discharge

When the plunger moves toward the arrow mark as shown in Fig. 13-13(c), the pressure inside the pump body rises above zero, the discharge valve being open to discharge the sea water and the suction valve is closely in contact with its valve seat. The repetition of these operations ensures the delivery of sea water.

B) Driving method of plunger

As shown in Fig. 13-14, the plunger driver is connected to the eccentric cam which is positioned to the centers of both the drive shaft and the plunger driver shaft. When the eccentric cam revolves, the plunger driver moves right and left by two times the eccentricity. The distance which the plunger moves is called "stroke" and is used together with the diameter of the plunger for calculating the capacity of the water pump.

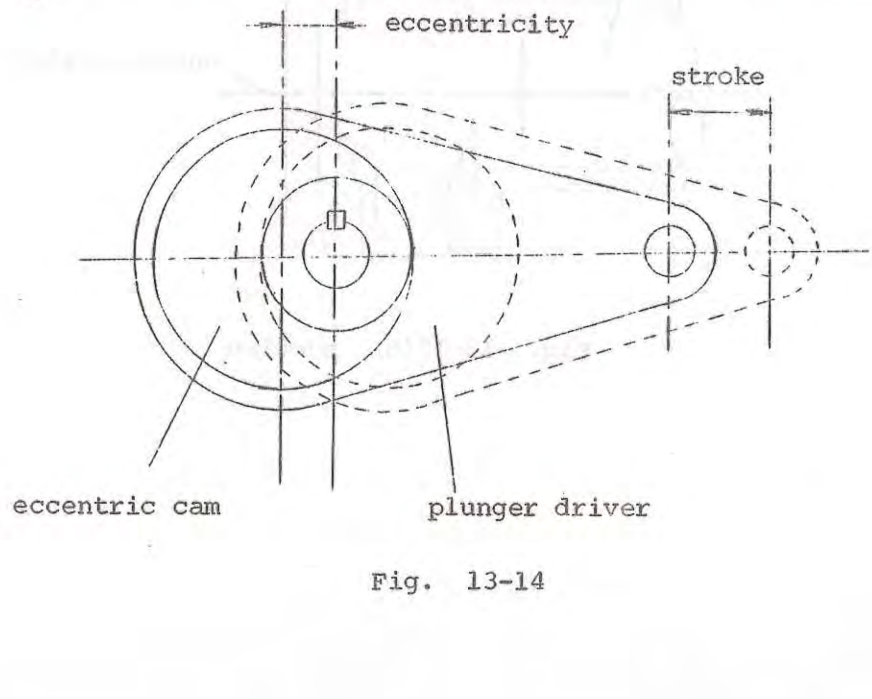


Fig. 13-14

C) Discharge volume of water pump

The following table show the actual measured discharge volume of some types of marine engine.

Table 13-2

Model	Horse-power	Actual discharge volume l/min (rpm)	
1A	3	1.2 (700 rpm)	1.35 (1,800 rpm)
1B	5	4.2 (800 rpm)	5.2 (1,000 rpm)
1C	10	11.5 (950 rpm)	17.5 (1,200 rpm)
2B	10	21.0 (1,600 rpm)	24 (2,000 rpm)
3D	25	27.1 (950 rpm)	29.1 (1,200 rpm)
3E	60	34.0 (750 rpm)	40 (950 rpm)
6E	100	52.0 (750 rpm)	60 (950 rpm)

Although the actual discharge volumes are as in the above table, they are different from the calculated discharge volumes. Figures 13-15 and 13-16 show the actually measured data of the relationship between the engine revolutions of some diesel engine, and their discharge volumes.

As seen from this chart, the actual measured discharge volume is far more than the calculated one. In case of Fig. 13-15 the actual discharge volume is double the calculated one at the normal revolution (about 1,400 rpm.). However, when the revolution is further increased, the discharge volume gets less without being preportional to the revolution (acting frequencies of the pump), which is a common characteristic of the plunger pump.

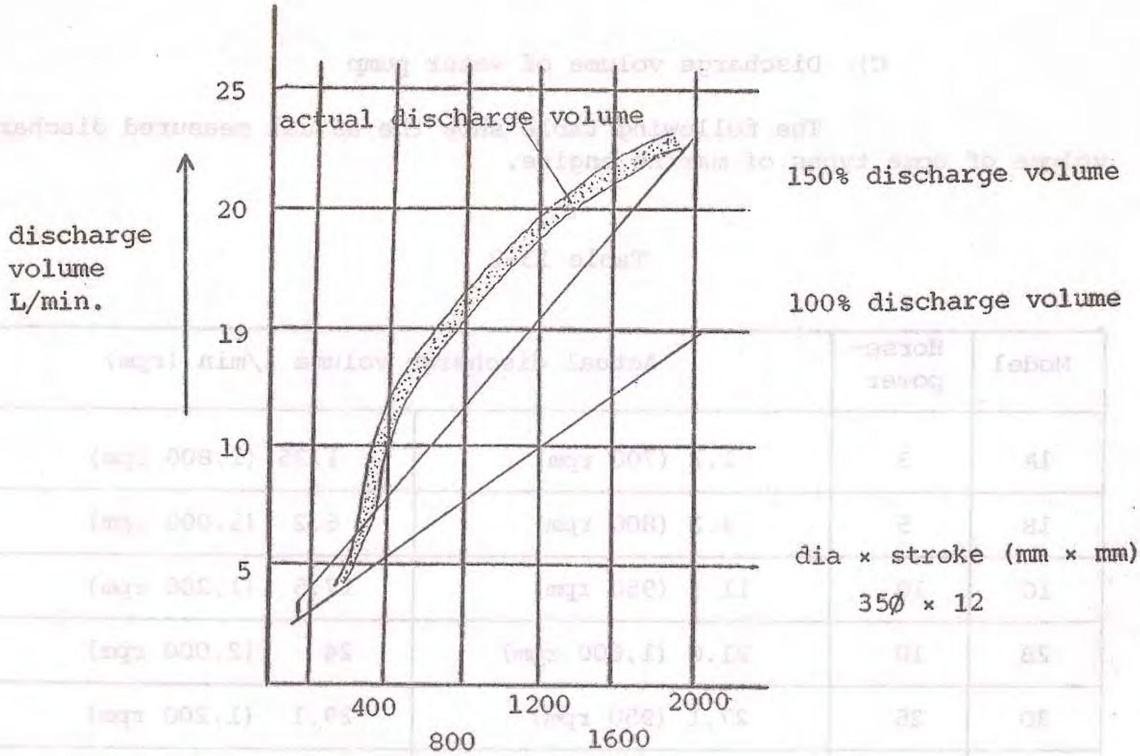


Fig. 13-15 Engine revolution (rpm)

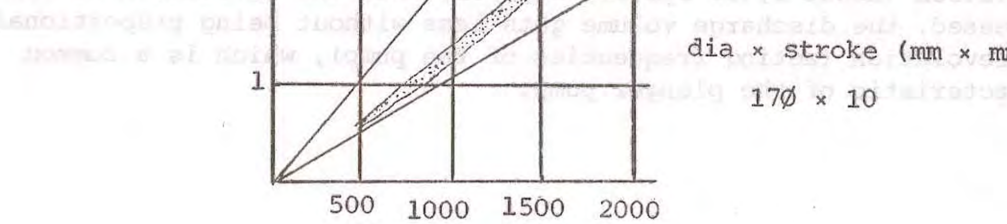


Fig. 13-16 Engine revolution (rpm)

This is supposed to be due to the following reason; when the speed of the plunger is decreased near the end of the suction stroke, the water opens the discharge valve to flow to the discharge side because of the strong water current force inside the suction pipe, that is, a part of the suction stroke precess acts to the effect of discharge.

Moreover, "valve lift" is also an important factor for decreasing the water volume in the pump. The standard valve lifts of some marine diesel engines is shown in the table below:

Model	Valve lift mm		Model	Valve lift mm	
	Suction	Discharge		Suction	Discharge
1A	2	1.5	3D	3	5
1B	2	1.5	3E	4	5
1C	3	3	6E	5	6
2B	3	3			

13.4.4 Air vessel

In the plunger type pump, pulsation is caused by its operation. In order to minimize its influence, an air vessel is provided at the outlet of the water pump.

13.4.5 Water adjusting valve and water thermometer

Figure 13-17 shows a water adjusting valve and water thermometer which can be used for the marine diesel engine. In this case, the water adjusting valve body is fitted at the water outlet on the cylinder head side, while the thermometer joint is fitted on the silencer side. These two are connected by a pipe. The water adjusting valve which is set in the water adjusting valve body is rotated to change the section area of water flow, thus regulating the discharge volume of water.

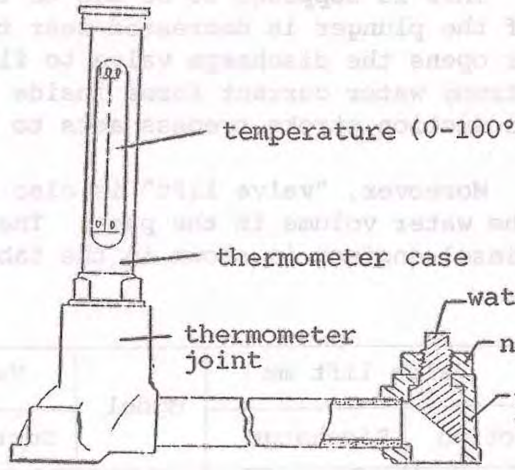


Fig. 13-17

13.5 Calculation Formula of Cooling Water Volume

The volume of the water which is necessary for cooling the internal combustion engine depends upon what percentage of the total calorific value generated by the combustion of fuel is to be removed. The quantity of heat to be removed by the cooling water "Q" and the circulating water volume "W" are calculated by the following formulas;

$$Q = X \times b \times BPS \times H$$

$$W = \frac{Q}{t_2 - t_1} \quad (L/hr)$$

where: X ..... Percentage of heat to be removed, out of the total calorific value of fuel

b ..... Specific fuel consumption (kg/PS.hr)

BPS..... Break horsepower (PS)

H ..... Total calorific value of fuel (Kcal/kg)

t_2..... Temperature of discharged cooling water (°C)

t_1..... Temperature of sucked cooling water (°C)

$$X = 25\% = 0.25$$

$$b = 180g/PS.hr$$

$$H = 10{,}000\ Kcal/kg$$

$$t_2 = 45°C$$

$$t_1 = 32°C$$

Then, $$Q = \frac{0.25 \times 180 \times 10{,}000 \times BPS}{1{,}000}$$

$$= 450\ .\ BPS\ (Kcal/hr)$$

$$W = \frac{450\ .\ BPS}{45 - 32} = 34.6\ BPS\ (L/hr) = 34.6\ (L/PS.hr)$$

The circulating water volume is 34.6 L/PS.hr. If the circulating water volume really adopted is 40 L/PS.hr, it has 15% margin because 40/35 is about 1.15.

13.6 Causes of Reduction of Discharge Pressure in Water Pump

1) When the filter of the Kingston valve is clogged with dust or sand;

2) When the water leakage through the clearance, produced by the wear of the plunger, is considerably increased;

3) When the lifts of both the suction and discharge valves are too big;

4) When damage of either suction valve seat or discharge valve seat causes water leakage;

5) When the valve spring is damaged or broken;

6) When the valve is broken;

7) When the lack of air in the air vessel causes a water hammer action and a decreased pump efficiency;

8) When air is sucked through the joints of the water suction pipe.

In each of the cases mentioned above the discharge pressure is decreased, for which you are required to take appropriate action.

13.7 Temperature of Cooling Water

The internal combustion engine is cooled for the reasons given below. The discharge temperature should be kept at 40-60°C.

A) The lubricating oil in the inner wall of the cylinder deteriorates due to high temperture and loses its lubricating properties.

B) With high temperature, the cylinder liner and piston expand, and this may cause burning stick.

C) Cylinder head, piston, exhaust valve, etc., are damaged by excessive heating.

D) High temperature causes pre-ignition in the injected fuel.

E) High temperature of cylinder causes insufficient suction and thereby a decrease in capacity efficiency.

If the temperature of the cooling water is too high, scale forms on the cooling surfaces of cylinder and liners. Scale is a non-conductor of heat; cooling efficiency is decreased and there is overheating which causes seperation of the lubricating oil film and sticking of piston and liner.

High temperature accelerates the corrosion caused by the electrolysis effect. If the temperature of the cooling water is too low, it lowers the temperature of the cylinder and liner which causes an incomplete combustion during operation. The result of this is knocking and bad discharge gas of the engine and increased fuel consumption. Furthermore, the sulfuric acid gas produced by the combustion of sulfur content is cooled and converted into the sulfuric acid, which corrodes the interior of the engine including liners and rings.

13.8 Replacement of Packings in Water Pump

1) When the packing for the plunger is worn, the leakage of water is increased. In such a case, tighten the clamping screw which is shown in Fig. 13-18. However, it should not be screwed in too tightly, because the water which leaks a little from there acts as cooling and lubricating agent for the plunger where no lubricant is applied.

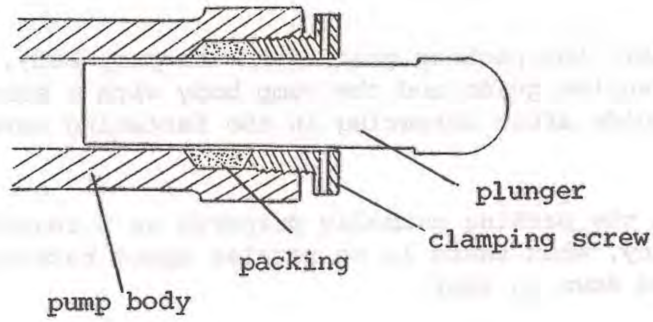


Fig. 13-18

If no water is allowed to leak from the packing the following troubles are caused:

a) A stick is brought to the packing which is sometimes burnt black;

b) Resistance is increased to break the plunger driver pin;

c) There will be an increase in wear of the plunger.

2) The replacement of the packing should be done as shown in Fig. 13-19:

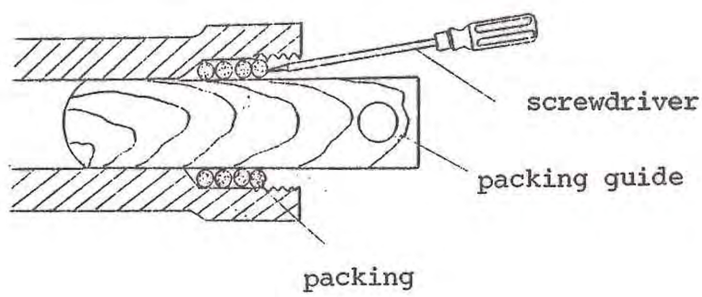


Fig. 13-19

Insert the packing guide into the pump body, stuff the wound packing between the guide and the pump body with a screwdriver, pull off the plunger guide after screwring in the fastening screw, and then fit the plunger.

3) Use the packing normally prepared as a regular spare part. In an emergency, when there is no regular spare packing, use the cotton packing boiled down in suet.