이행정 트렁크형디이젤 기관에 있어서 중유사용으로 인한 시린더마모의 소고

강 봉 수

Influence of Intermediate Fuel on Cylinder Wear in a Two Stroke Trunk Engine

Kahng Bongsoo

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초 록

최근에 와서 박용디이젤기관에서는 거의 중유를 연료로서 사용하고 있다. 이에따라 종전의 경유기관보다 시린더마모율이 훨씬 크지고있는 실정이다. 여기에 필자는 착안하여 아래와같은 세가지 운전조건상태에서 실험하여 마모율을 훨씬적게 할수있으며 따라서 유지비절감에 기여하고저 하는바이다.

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2. 냉각수의온도
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1. Summary

This report covers an investigation on cylinder wear in a two stroke trunk type diesel engine, using intermediate fuel oils.

Radioactive tracer techniques have been employed to measure the wear rate of the upper piston rings.

The test have shown that, if appropriate measures are not taken, cylinder wear may increase by more than 130% when engine is running on intermediate fuels, compared with gas oil fuel.

The wear rate may be reduced to a more acceptable figure of not more than 50% above

that of gas oil, by merely adjusting the fuel preheating temperature. The engine should be stopped and started on gas oil due to the abnormally high injection pressures which would result from cold intermediate fuel.

A 10% reduction of the wear rate was obtained by increasing the fuel oil temperature from 70℃ to 90℃.

Rising the cooling water temperature from 60℃ to 80℃ caused another 30% wear reduction.

An optimum cylinder oil feed rate of 0.5 g/bhp/h has been determined under this test conditions when the engine is running on a fuel whose viscocity is 100 Redwood at 100 F. This refers to a load of 125 bhp/cyl.

Injection characteristics have been recorded for different fuels and preheating temperatures.

No evident defects have been recorded.

2. Introduction

This test series has been concerned with the effect of intermediate fuel oil on the cylinder wear of the engine and also studied the effect of operational parameters when using this fuel.

The investigation has had a somewhat practical slant, as I have attempted to answer questions like;

How large cylinder oil feed rate should be used?
How large an effect does increased fuel oil viscosity have?
To which temperature should the fuel be preheated?
etc.

I hope that the result of this investigation will prove of value to engine builders and users, and help pave the way for increased use of intermediate fuels on appropriate trunk diesel installation. These tests have also given us further experience in the use of radioactive tracer techniques. probably the largest source of erro in my tests-the settling of radioactive particles in the lubricating oil system-has been kept under control by changing the system oil more often, thereby keeping the contamination at a low level. I therefore regard the measuring technique employed in these tests to give acceptable reliability.

The Wichman AC type engine employed for this test has not previously been run on intermediate fuels. The test show that such fuels can successfully be used on this engine type withought suffering unacceptable liner wear, if operational parameters are adjusted accordingly.

I have also gained some practical experience with running the engine on intermediate fuels, experience which can prove of value for later experiments or be helpful when designing or operating engine installation where such fuel are used.

3. Test Installation

The test were carried out on a Wichman 2 ACAT diesel engine in the laboratory at the

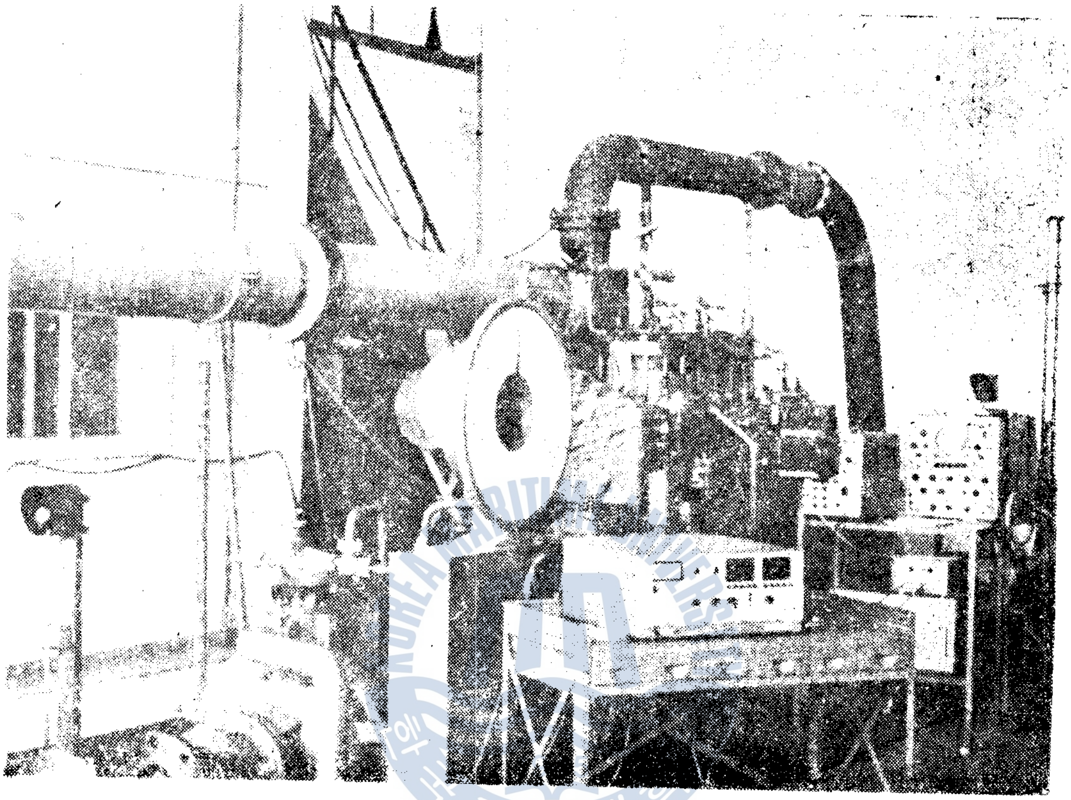


Fig. 1 Picture of the engine test setup

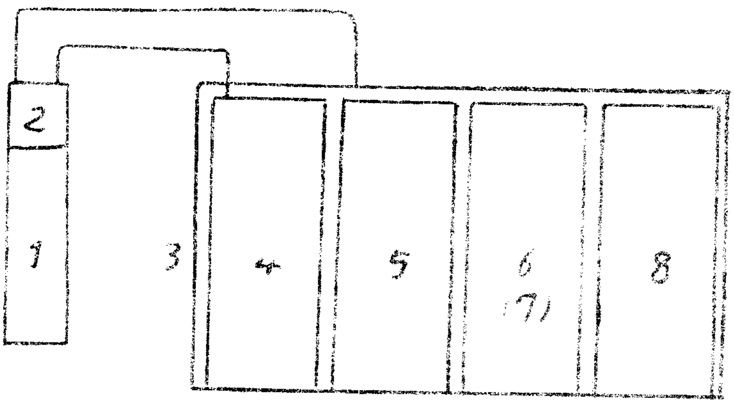


Fig. 1 Setup of registration instruments

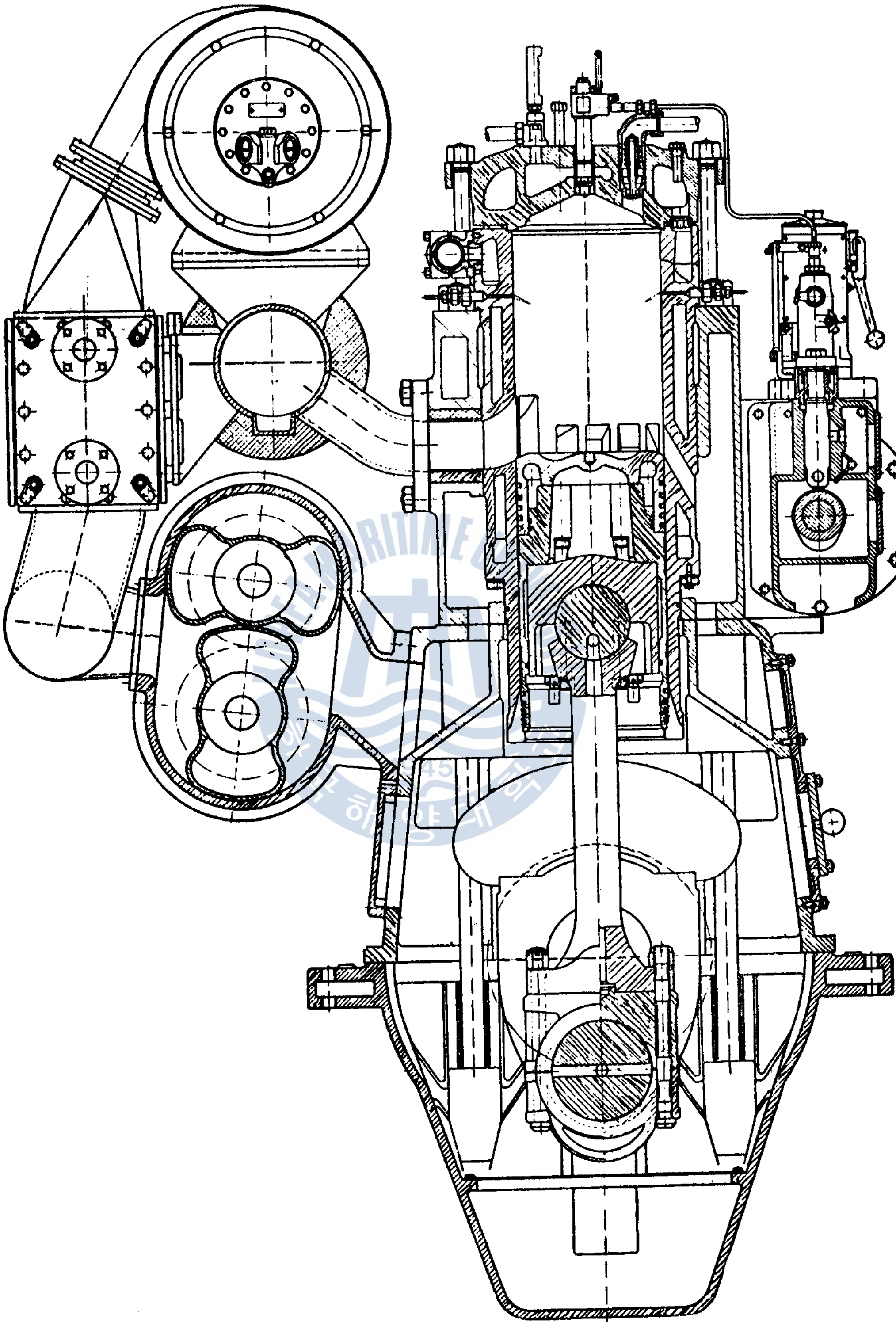


Fig. 2 Cross section of Wichmann ACAT engine.

Institute of Internal Combustion Engine at the Technical University of Norway, Trondheim.

This two stroke, turbocharged, trunk piston engine run at a moderate speed, and is intended for marine propulsion duties directly coupled to a controllable pitch propeller.

It has a hydraulically operated friction clutch and servometor assembly built integral with the engine inside the bed plate.

The laboratory engine has two cylinders.

The test setup is shown in fig. 1, and cross section of a Wichman Acat engine is shown in Fig. 2.

The uppermost piston ring in cylinder number one(front end of engine) has been activated in a nuclear reactor, and the ring wear monitored by resisting the increase in radioactivity of the lubricating oil system. Due to the large air content of the system oil on this engine, a sampling method is employed, and samples allowed to deaerate before the radioactivity is measured by a scintillation counter. Figure 3 shows the lubricating oil system and measuring circuit. The following modifications were made for these tests;

a) The normal cylinder liner for the engine was reinstalled in cylinder number one. This has a total of two lubricating quills in the upper part of the liner(130mm from the top placed in the port and starboard posisitions. The drilled passage from the quill(check valve) to the liner surface has a throttling screw filler(snubber insert) and normal grooves in the liner.

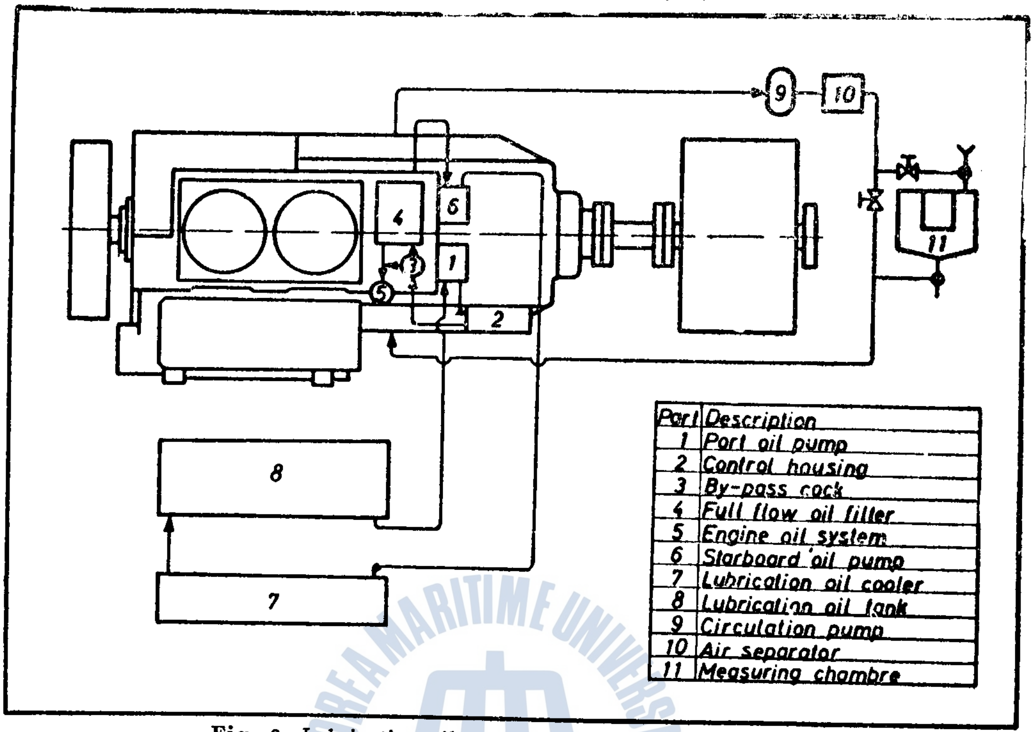
(b The standard fuel injection nozzles and holders were changed to the dirrectly cooled type having the same orifice specifications as the standard nozzles. The new nozzle holders had a shaft length of 200mm(instead of the standard 80mm), as this was available of the shelf. The following particulars apply;

Nozzle:	Bosch	5 holes
Holder:	〃	KBF200 T 2/20 602
Coolant:	Water	42-48℃ out

c) The fuel injection pumps, originally bosch PF I C 160 BS 29 with 16 mm plungers, were altered to Bosch PF I C 180 BS 1640 with 17mm plungers and with leak off drain to the suction side of the pump.

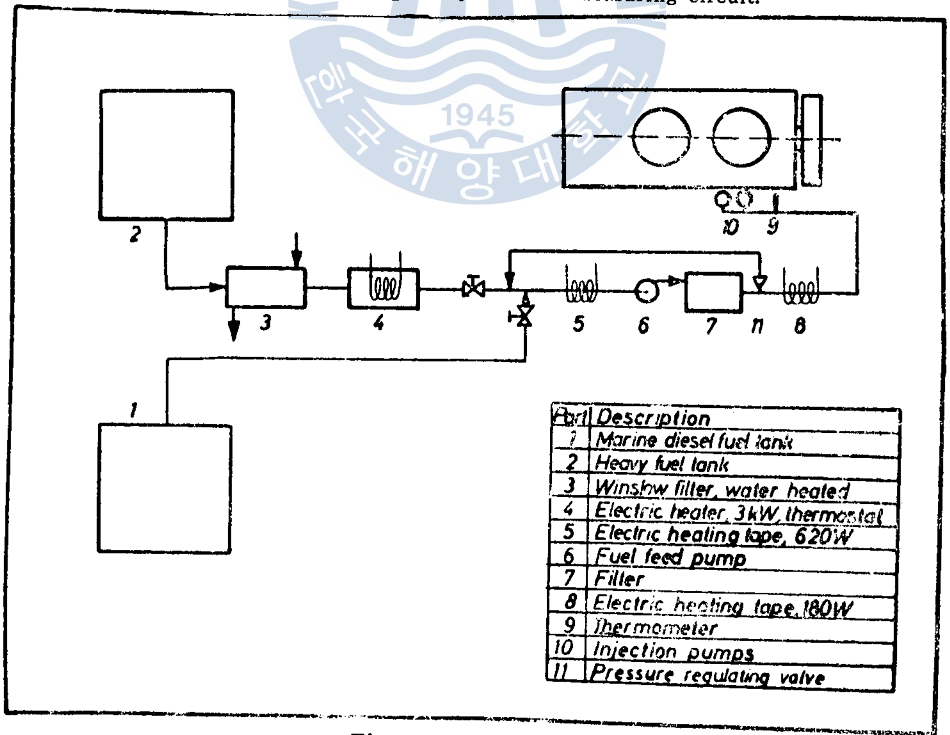
d) The injection cams were changed to the latest profile with increased plunger speed at the start of injection.

The fuel injection equipment now corresponds to the specification presently used for the turbocharged Wichmann ACAT engine. The combustion on this laboratory engine with a special(smaller) turbocharger had previotrusly not been satisfactory, even though standard production engines with similar injection equipment(but with the standard turbocharger)had given very satisfactory results under test. When a heavier fuel was used on the laboratory engine, however, the combustion became unacceptable for our purpose, giving a dark exhaust and a very dirty engine. Tests showed that the original injection equipment(for the 125 bhp/cylinder output)did not match well the special configuration of this laboratory



Part	Description
1	Port oil pump
2	Control housing
3	By-pass cock
4	Full flow oil filter
5	Engine oil system
6	Starboard oil pump
7	Lubrication oil cooler
8	Lubrication oil tank
9	Circulation pump
10	Air separator
11	Measuring chambre

Fig. 3 Lubricating oil system and measuring circuit.



Part	Description
1	Marine diesel fuel tank
2	Heavy fuel tank
3	Winslow filter, water heated
4	Electric heater, 3kW, thermostat
5	Electric heating tape, 620W
6	Fuel feed pump
7	Filter
8	Electric heating tape, 180W
9	Thermometer
10	Injection pumps
11	Pressure regulating valve

Fig. 4 Fuel oil system.

engine, and the present specification of injection equipment was envoled. As a result the conbustion improved considerably, the smoke readings now being very satisfactory for all fuels and operating conditions tested. It was also gratifying that the injection equipment coincided with the present standard(for 150 bhp/cylinder rating) so that the results are more readily applicable to production engines. The intermediate fuel, even the 100sec. Redwood 1, was heated in order to attain an acceptably low viscosity. The various equipment for heating of the fuel oil was installed piece by piece as the viscosity was increased and the tests progressed.

Figure 4 shows schematically the final arrangement of the fuel oil system.

The fuel oil was originally heated by circulating hot water through the heating jacket on the Winslow filter number 3 in the figure.

Later a tank with a 1 KW heating element was installed after the filter.

In both cases the return flow from the engine feed pump went back to the service tank.

The heating ability of this system proved insufficient for later test requirements. Furthermore, it did not allow the engine to be started on gas oil fuel. The fuel oil system was therefore altered to the configuration shown in figure 4, whereby the heated fuel oil is also fed back to the engine. After the alteration the piping was lagged up to the injection pumps, and the final double filter partially lagged.

However, the heating tape shown in the figure was not installed until the tests required a fuel oil temperature of 90°C.

Mesuring method is as follows. Newtron radiation method was taken undercorporation with IFA in Norway. The rings lay 3 weeks in the reactor in a neutron flux of $2 \cdot 10^{12}$ n/cm²/sec. The measuring chamber is shielded by about a 5cm thickness of lead in order to reduce the background radiation. The registration apparatus was as follows, See figure 1′

1. scintillation detector Harshaw NaJ (Ta)type 7SF8/E
2. Pre-amplifier NOR4194
3. Cabinet with power supply NOR4171
4. High voltage supply NOR4165
5. Pulse height analyser NOR4115
6. Scaler NOR4101
7. Ratemeter NOR4145
8. Timer NOR4120

Calculation of the wear:

1) change of volume per unit time, $S_v = \Delta V/t$
2) change of mass per unit time, $S_m = \Delta m/t$
3) change of dimension per unit time, $S_L = \Delta L/t$

$$S_m = \rho S_v$$

$$S_L = S_v/A \quad (A \text{ is wear area})$$

S_m having the dimenson [mg/h]

S_L having the dimension [mm/10.000h]

$$S_L = 0.182 \cdot S_m \quad \varDelta m = \frac{dI}{dt} / I_{cal} \text{ [mg/h]}$$

The loss of piston ring, or wear, is thus:

$$S_m = q \cdot \varDelta m = \cdot q \frac{dI}{dt} / I_{cal} \text{ [mg/h]}$$

The radial wear of the piston ring is:

$$S_L = 0.182 \cdot S_m = 0.182 \cdot q \cdot \frac{dI}{dt} / I_{cal} \left[\frac{mm}{10.000} h \right]$$

$$q = 77$$

4. Test Condition

This project has as its primary objective an investigation of the liner wear associated with the use of fuel oils up to approx. 200 seconds Redwood 1. and also an investigation of the influence on the wear rate of various operating parameters when using such fuels, Typical parameters studied were:

a) Cylinder oil feed rate.
b) Preheat temperature of fuel oil
c) Temperature of cooling water.

The method followed was to investigate the influence of changing one single operating parameter at a time. All other conditions were accordingly held as constant as possible.

The main operational data are:

Bore:	280mm
Stroke:	420mm
Output per cylinder:	125bhp
Mean effective pressure:	6,22kp/cm²
Engine speed:	350rpm
Oil temperature before cooler:	40-45°C
Piston cooling oil:	65-68°C
Cooling water from nozzles:	42-48°C
Exhaust temperature to turbine	350~360°C
Scavenging air Pressure:	0,5kp/cm²
Scavenging air temperature:	30-32°C

All tests except no. 20-22 were carried out with a cooling water temperature of 60-62°C at the outlet from the cylinder heads.

The engine manufacturer concered suggested Mobilgard 393 be used for these tests. In accordance with his wishes Mobilgard 393 was used for both cylinder oil and system oil during all tests. The lubricant was generously supplied free of charge by Mobil Oil A/S

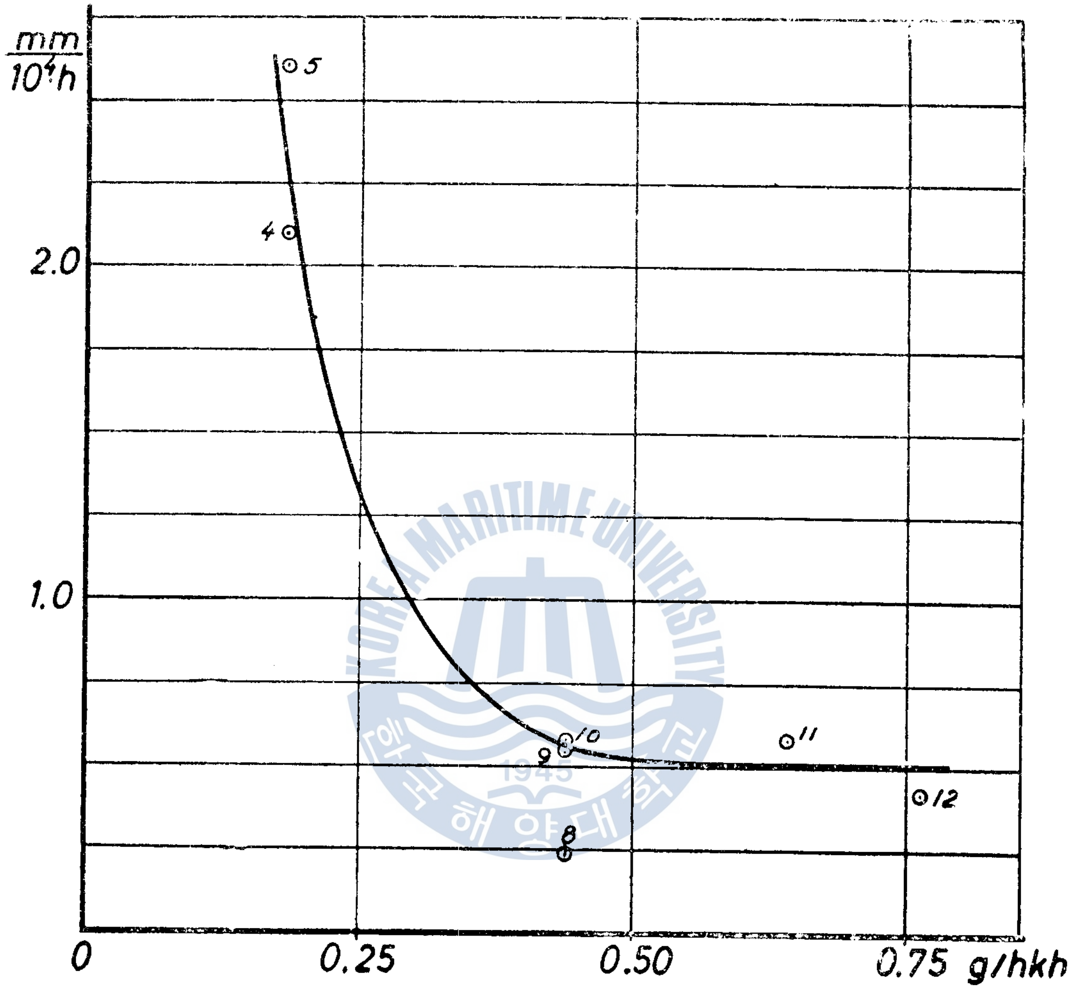


Fig. 5

Oslo.

Three types of fuel oil were employed during these tests, namely:

a) Gas oil.

b) A mixture of a) and c) giving a viscosity of about 100 sec. RI at 100oF.

c) Light Marine Fuel Oil, delivered by BP in Trondheim, with a viscosity not exceeding 220"RI.

The facilities limited the fuel to small batches. The following Table 1 gives the batches used.

TABLE 1

Fuel Oil Blend No.	Fuel Oil Type	Test No.	Viscosity sec Red. 1	Sulphur Content
1	b	4, 5	100	1.64
2	b	8, 9	90	1.67
3	b	10	105	1.64
4	b	11, 12	102	1.64
5	c	13	202	1.9
6	c	14, 15, 16	206	1.9
7	c	17, 18, 19	203	1.9
8	c	20, 21, 22	194	1.9

The sulphur content is obtained on the basis of analysis data for blend no. 5 and that for gas oil.

The tests have been run with the following preheat temperatures. of the fuel oil, measured at the inlet of injection pumps:

1) 35−40°C. This temperature was employed before sufficient heating capacity was inserted in the circuit. (Tests 4−12). The viscosity when using fuel "b" (100 s RI) was therefore somewhat higher than what is often regarded as suitable for fuel injection equipment of this type, namely around 75 "RI.
2) 70°C. Used in connection with fuel oil c) this temperature gives a viscosity lying just under the arbitrary of 75 "R1 at atmospheric pressure.
3) 90°C. This was the highest preheat temperature employed, and was used under the investigation of the effect of fuel oil·viscosity on the wear.

5. Test Result

The present investigation included in all 22 individual test runs.

Five of these have not given satisfactory results, mainly due to various minor operational difficultes with the fuel injection system in the preliminary stages.

The individual tests are recorded in Table 3 and shown in the figures given in the table.

In this report the wear of the upper piston ring is given as mean radial wear. Note, however, that it is also common, especially for liner wear, to refer the wear figures to the diameter.

In the following the effect of various operating parameters is discussed.

5·1 Cylinder oil feed rate

Figure 5 shows the ring wear as a function of the cylinder oil feed rate from the lubricator. The measurements are based upon tests with fuel oil b) (100 "R1). It will be seen that the wear rate increases quite rapidly when the feed rate in these tests was reduced below about 0. 1 g/bhph. The large increase in wear is thought to be due to the inability to maintain a satisfactory oil film over critical areas of the liner with a reduced oil supply. These measurements do not indicate that an increase of the feed rate beyond 0. 5 g/bhph will significantly reduce the wear when fuel type b) is employed and load is 125 bhp/cy1.

5·2 Type of fuel oil

Figure 6 shows the wear for the three fuel oils a), b) and c). The tests were run with the same cylinder oil feed rate on the lubricator(3 drops). The fuel oil quality is in the figure given by the sulphur content.

These test show that the wear is definitely larger for the two heavier fuels b) and c) than for gas oil (type a). This may be due to the heavier oils having a much higher content of sulphur, asphaltenes, ash, etc. The sulphur is conductive to corrosive effects, while other impurities may from solid particles in the combustion gases which in turn contribute to the wear. The combustion itself seems to be similar and acceptable under all tests. The smoke density has been registered around 5% with a Hartridge Smokemeter on all runs. No marked difference in fuel oil consumption has been registered apart from the normal discrepancy due to different calorific values.

5·3 Preheating of the fuel oil

Figure 7 shows the wear rate when the fuel oil is heated to 70° and 90°C. This tests have been run only with fuel oil type c) (200 "R1). The tests indicate that the wear dimishes somewhat with increased preheat. The higher temperatures give a lower viscosity and thereby a change in the fuel injection(atomisation) characeristics. This seems to improve the combustion somewhat as far as wear is concerned. The present setup does not allow a preheat temperature much above, 90°C. One should not, however, expect to find a similar decrease in wear rate when the temperature is increased further.

5·4 Cooling water temperature

Figure 8 shows the relationship between the wear measured and the corresponding cooling water temperature. The tendency under these tests is evident: the wear decreases with increasing cooling water temperature. Other tests (2) have shown that the wear decreases with increasing temperature until a water temperature of about 75°C is reached. Further increase had no marked effect on the wear. Our data is not sufficient to define a break point on the curve.

A temperature limit of this nature is believed to be connected with the formation of acids on the inner surface of the liner. It is evident that the configuration of the cooling spaces, etc. will have direct bearing on the temperature of the liner surface, and thereby on the limiting cooling water temperature for a particular design. Liner surface temperature at a suitable point would perhaps have given a more valid basis for presentation of results.

The results are strictly speaking also only applicable to the rating of 125 bhp/cyl (pe= 6.22kp/cm^2) employed for these tests. A higher engine load, for example the 150 bhp/cyl rating, would of course increase the metal temperatures somewhat and thereby give slightly different results. It is not inconceivable that with higher ratings and higher water temperatures, a point would be reached where metal temperatures in the upper portion of the liner exceeded a limiting value, whereby increased water temperatures would give an increase in wear.

6. Evaluation of Results

The wear has been calculated on the basis of the radioactivity in the lubricating oil system. This wear figure will be lower than the actual wear of the piston ring, partly due to the fact that some of the wear particles are deposited in the ports, and partly due to wear particles leaving the cylinder with the exhaust gases.

Some of the tests (no. 18, 19) show that a part of the activity is deposited in the engine during th[illegible] 1−3 hours after starting. When evaluating the data one has therefore ignored [illegible] period. A certain amount of activity is probably constantly being deposited in the system throu[illegible]tire test, but the rate of this deposit is, as far as we can see, very small after 3−4 hours running time. On the whole, these deliberations indicate that the figures for wear rates should be regarded more in the nature of relative values for purposes of comparison rather than quantitatively correct numerical

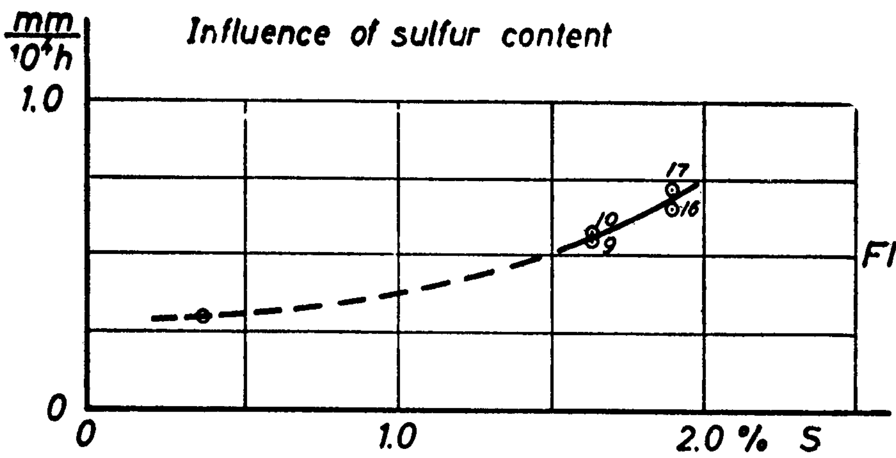


FIG. 6

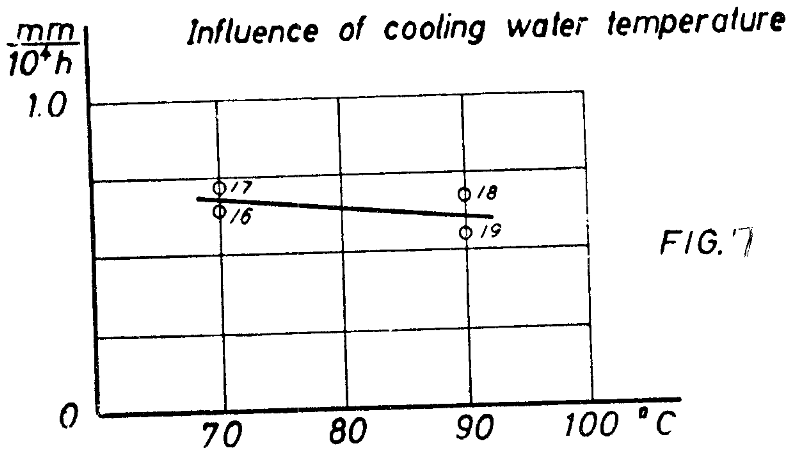


FIG. 7

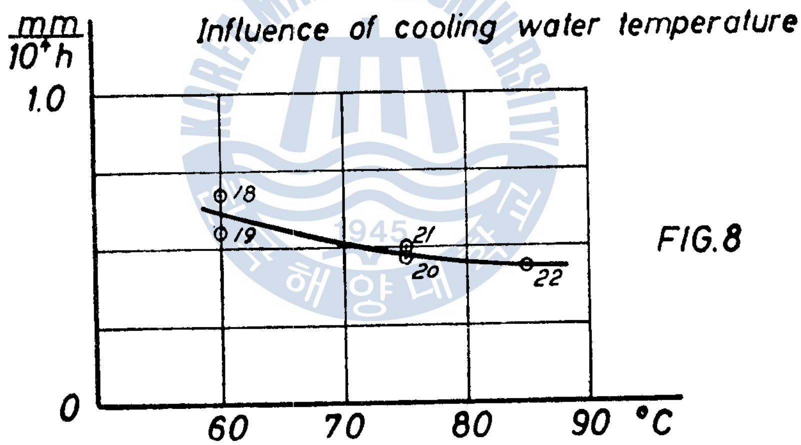


FIG. 8

values.

The tests have only attempted to measure wear of the upper piston ring. The ring wear is, however, often used as a measure of liner wear, as one then assumes the ratio of ring wear to be constant.

This assumption appears to be reasonable when gas oil is employed. When heavier fuels are used, however, the assumption is not necessarily valid. The reason for this may lie in the fact that ring and liner are not of exactly the same material, and that they may therefore not be affected in quite the same manner by corrosion. One should therefore strictly speaking not directly assume relative wear rates given here to be also valid in all cases for liner wear.

The cylinder oil feedrate has in this report been calculated from the oil supplied to the lubricator. The true cylinder oil consumption may be somewhat higher, as we can assume the oil scraper rings are not always capable of scraping down all the oil impinging upon the liner surface. The consumption from the oil sump is very small and has not been measured. Make-up oil required would not be significant, and it is difficult to determine how much of this slight loss finds its way to the upper part of the liner in question.

The quality of the fuel oil has been somewhat variable from delivery to delivery. The last tests(20, 21, 22) have thus been run with a fuel of lower viscosity than the previous runs. These tests probably give slightly lower wear rates than would otherwise have been found with the heavier fuel. This effect is, however, not assumed to be of any signifficant magnitude.

7. Practical Experience Uting Heavier Fuels

It is absolutely necessary to switch over to gas oil when starting and stopping the engine. Even though the heavier fuel is preheated before start, the fuel standing in the injection system and the final filter (item 7, fig. 4) will be cold. (This applies to the normal arrangement expected for this class of fuel where the injection system and piping are not heated When the engine is started on the cold viscous fuel, the injection pressures rise to very high values. Even though the fuel temperature rises quickly, large forces are for a short while imposed upon the injection system, and one runs the risk of bursting the jerk pumps when starting. This may happen at as low a fuel oil viscosity as 100 "R1(fuel b).

It did happen with 18mm plungers in our PFIC180B S1640 pumps(cracked pump liner).

The higher injection pressures caused considerable fuel oil leakage(though timing window, etc.) on our original Bosch pumps. New pumps with leakage drain back to the suction side proved very helpful in this respect. Strengthened pumps seem advisable, and one should at least try to avoid standard injection pumps bored to take the max. plunger diameter available. It should be noted when changing over from gas oil to heavier fuel that it takes time to heat the filter. It may therefore be an advantage to blend some gas oil with the heavier fuel until the temperatures reach a satisfactory level. The engine should be switched over to gas oil sufficiently early before stopping so as to allow the fuel blend in the injection pumps and piping to consist mainly of gas oil for the next start.

The fuel oil ought to be heated the minimum requirements, if possible. For fuel oil type c) (200 "R1), the preheat temperature should at least be 70°C, corresponding to a viscosity just under 75 "R1 at atmospheric pressure. In order to improve injection characteristics and reduce wear, the tests indicate that the temperature for 200 „R1 fuel should preferably lie 90°C or higher. Note that the viscosity of such fuels increases with pressure, and higher temperatures lower viscosity at atmospheric pressure should be employed in order to maintain a suitable viscosity at injection pressure.

The specific fuel oil consumption was about 3 larger with fuel type b) (100 "R1). This is mainly due to the lower calorific value of the fuel, but slightly poorer injection conditions may also have some part in this. The fuel consumption has not been recorded for fuel oil typec(200 "R1.)

8. Injection Characteristics

In order to check the pressures and injection characteristics of the fuel injection system, recordings were made with various fuels and preheat temperatures. The injection pressure was registered in the piping just above the Bosch pump and also just before the nozzle connection, in both cases by means of GAV transducers sufficiently small to give insignificant extra volume. These transducers are of the strain gauge type. The signals were fed to a four-trace Tektronix oscilloscope via Hottinger bridge.

The needle lift was monitored via an extension rod out through the top of the nozzle holder and measured by a Disa capacitive transducer and tuning plug. The signal was fed via a Disa oscilloscope to the third channel on the Tektronix oscilloscope. The degree markings were generated by a Disa unit attached to the engine.

The recordings were made at n=350 rpm and Ne=250 bhp, first with gas oil just after the engine reached operating condition, and then with a 200 "R1 fuel as the fuel oil temperature before the injection pumps gradually increased. The pressure basis was determined by benting the system(screw on nozzle holder) when running. During these tests the traces were recorded on an ABEM Ultralette UV recorder. The pressure diagrams are shown in figure 9.

It will be seen that the course of the recordings is approximately the same in all cases, while the maximum pressure changes somewhat with the viscosity of the fuel oil. With intermediate fuels the needle executes two distinct oscillations when closing. During the second of these the pressure at the nozzle is sufficient high to create the possibility of some slight dribbling. No pronounced effects were evident.

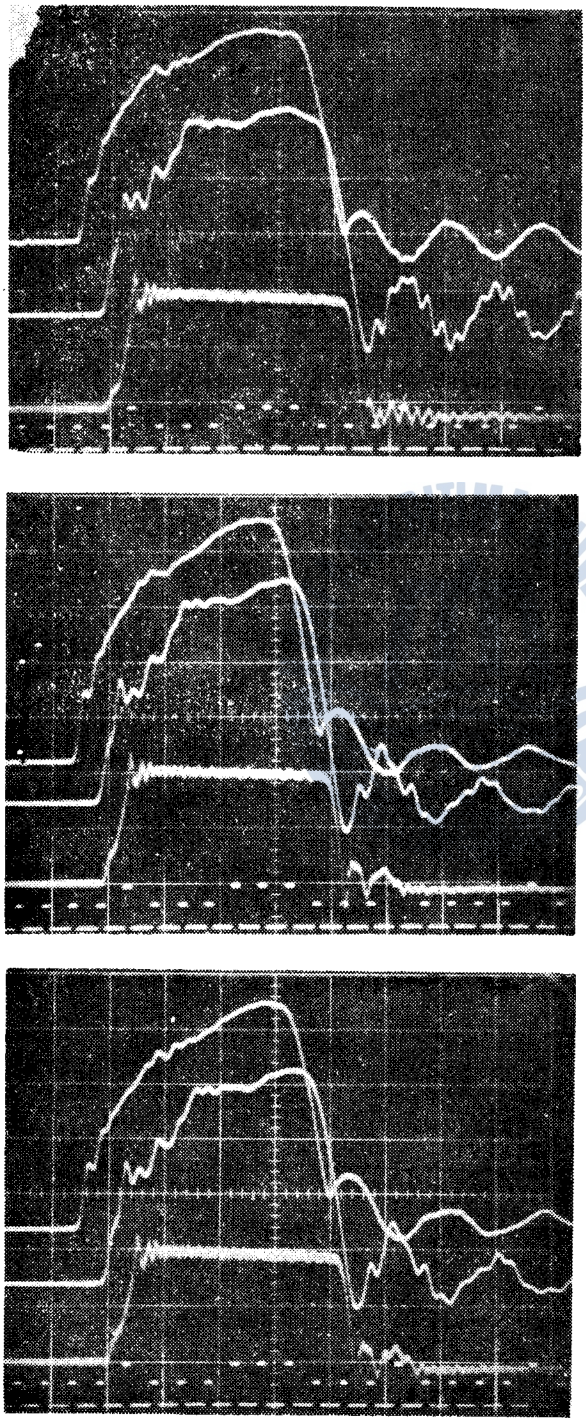
The maximum injection pressures(rounded values) are given in table 2. It will be seen that the pressure increases with viscosity, as one would expect.

TIABLE 2

Maximum injection pressure for gas oil and intermediate fuel of 200 s R1 at 100° F, Based on a pressure drop of 180 kp/cm² when venting system

	Fuel oil temperature	at pump kp/cm²	at nozzle holder kp/cm²
gas oil intermediate fuel	20°C	560	550
(200 s R1)	90°C	590	570
〃	80°C	600	580
〃	70°C	610	590
〃	60°C	620	590

Fig. 9 Injection pressure diagrams.



From top of the pictures the curves are:

1. Pressure at injection pumps
2. Pressure at nozzle holder
3. The needle lift
4. Crank angle marking

Fuel: Gas oil

Fuel viscosity: 200 "R1 at 100° F
Temperature: 60°C

Fuel viscosity; 200 "R1 at 100°F
Temperature; 90°C

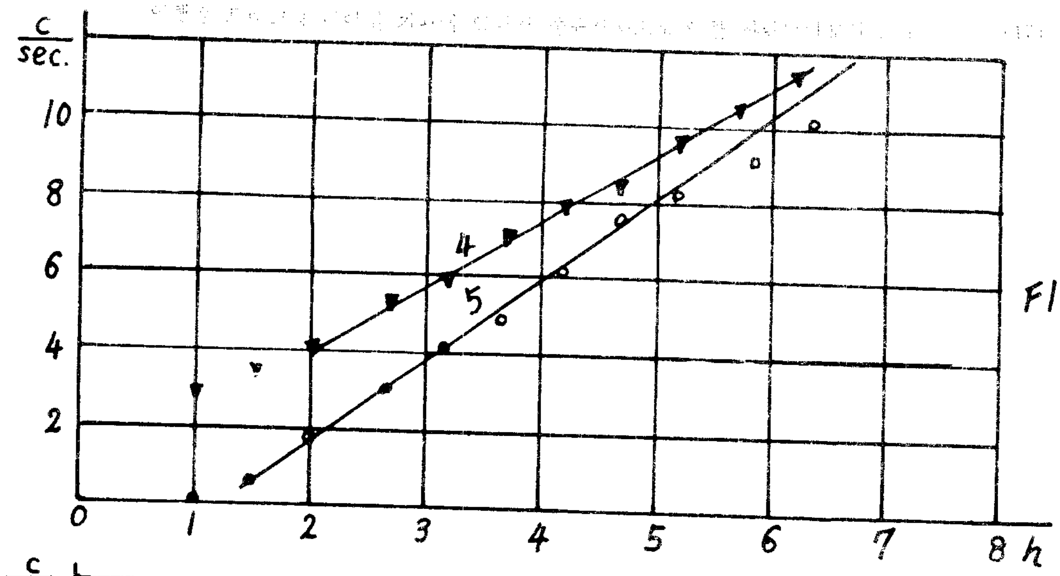


FIG.10

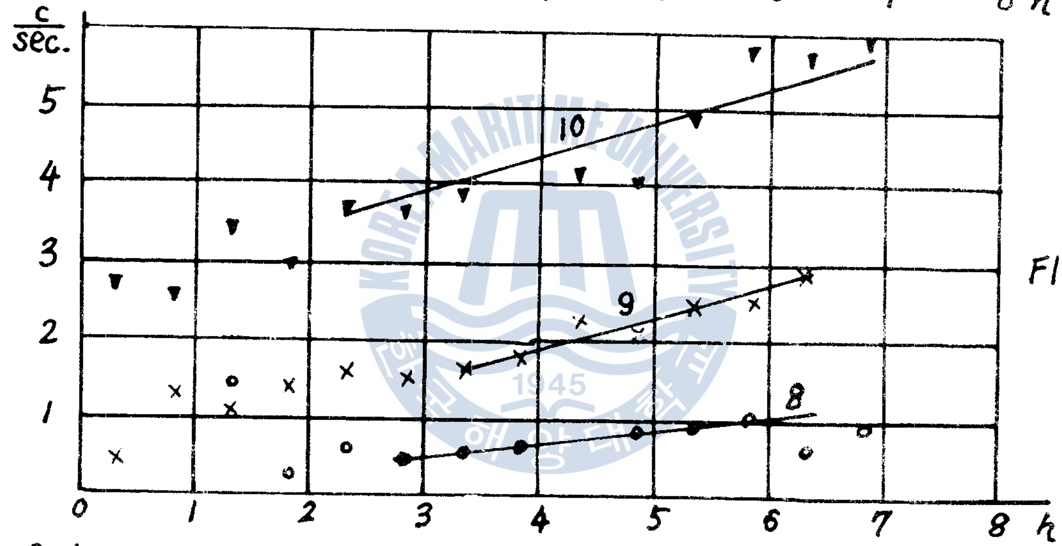


FIG.11

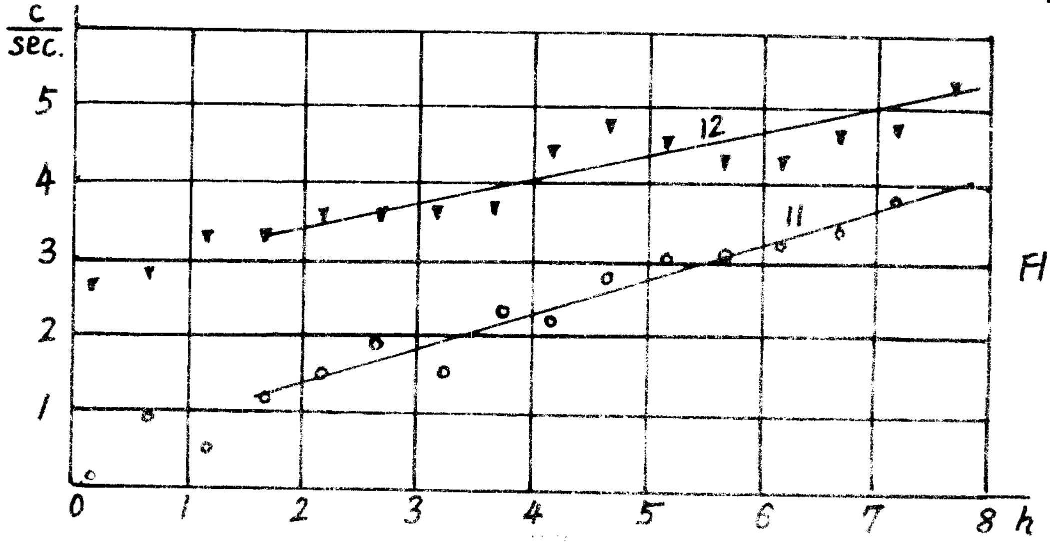


FIG.12

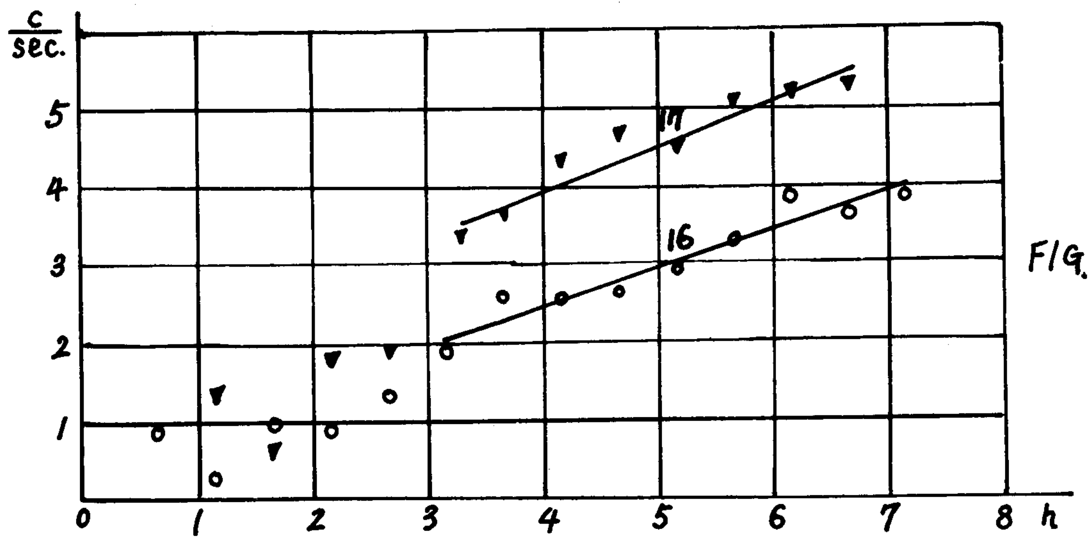


FIG.13

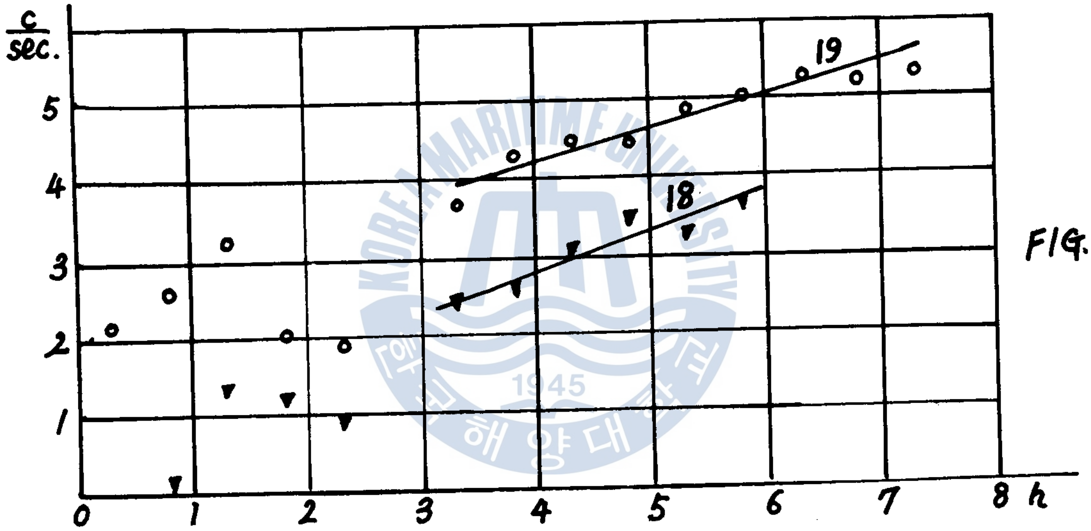


FIG.14

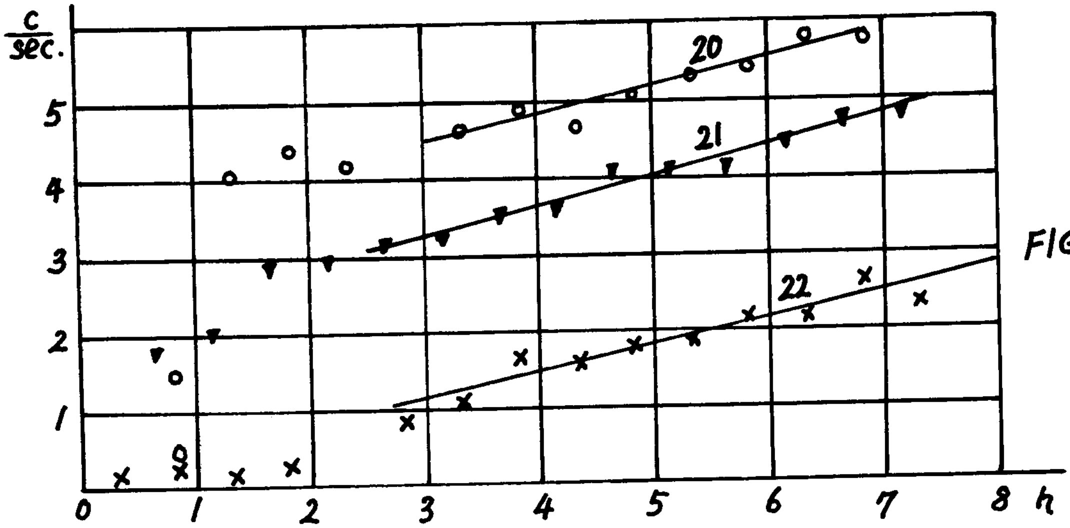


FIG.15

APPENDIX A RESULTS OF INDIVIDUAL TESTS.

The individual tests are listed below and shown in figures on the following pages. The engine operational data are listed on page 8. Lubricating oil quality is:

Mobilgard 393

TABLE 3

TEST NO.	ACTIVITY OF FIRST SAMPLE (c/s)	MEAN ACTIVITY INCREASE (c/sh)	EQUIVALENT RING WEAR (mm/10 000 h)	CYLINDER OIL FEED RATE (g/hph)	FUEL OIL TYPE	FUEL OIL TEMPERATURE (°C)	COOLING WATER TEMPERATURE (°C)	FIGURE NO.
1	11. 4	0. 25	0. 30					
2	13. 2	0. 25	0. 30		A		60	
3	13. 2	0. 15	0. 18					
4	15. 9	1. 75	2. 1	0. 18	B	35	60	10
5	27. 2	2. 15	2. 6					
8	42	0. 19	0. 24					
9	44	0. 42	0. 55	0. 43	B	35	60	11
10	46	0. 45	0. 58					
11	50	0. 45	0. 58	0. 76	B	35	60	12
12	52	0. 32	0. 42	0. 76				
16	22	0. 50	0. 65	0. 44	C	70	60	13
17	26	0. 55	0. 72					
18	25	0. 52	0. 52	0. 44		90	60	14
19	30	0. 425	0. 425					
20	27	0. 35	0. 35	0. 44	C	90	75	15
21	38	0. 385	3. 885					
22	44	0. 34	0. 34	0. 44	C	90	85	15

APPEENGIX B ANALYSIS OF FUEL OIL

The average import analysis as supplied by the oil company Norsk Brandselolje A/S is as follows:

Type:	Light Marine Fuel
Oil Specifc gravity:	0. 92−0. 93 at 15°C
Viscosity:	max. 220 "R1 at 100°F

Flamepoint:	80−90°C
Pourpoint:	−30°C
Gross calorific value:	10450 kcal/kg
Carbon residue, Conradsontest:	7−10%
Sulphur:	1.5−2.5% (weight)
Ashes:	0.01−0.02%
Asphaltenes:	1.0−2.5%
Vanadium pentoxide:	122−150 ppm
Vanadium:	68
Sodium:	20
Water and sediments:	traces.

Our analysis on one particular batch (no. 5, table 1) has give the following result:

Sulphur:	1.9%

APPENDIX C REFERENCES

1. H. Valland Influence of lubricating quill location on cylinder wear in a two stroke engine. Report no. IF/RI, The Institute of Internal Combustion Engines, NTH, Trondheim, 1967.
2. W.G. Payne, W.F. Joachim investigation no cylinder liner wear. SAE Trasactions Vol. 3 no. 1 1949 P. 51—8
3. J.W. Schrakamp; Discussion at CTMAC—1962 Conference, Copenhagen, p. 723—4

APPENDIX D LIST OF FIGURE