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ENGINEERING DEPARTMENT – DROOGBAK 1 A – AMSTERDAM

INFLUENCE OF PISTON TEMPERATURE ON PISTON FOULING AND PISTON-RING WEAR IN DIESEL ENGINES USING RESIDUAL FUELS

(DE INVLOED VAN DE ZUIGERTEMPERATUUR OP DE VERVUILING EN DE ZUIGERVEERSLIJTAGE VAN DIESELMOTOREN BIJ GEBRUIK VAN RESIDUALE BRANDSTOFFEN)

BY

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Issued by the Council

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CONTENTS

SUMMARY

A description is given of tests on a two-stroke, two-cylinder engine running for 50 hours on heavy fuel to ascertain the fouling of piston, piston rings, etc. and piston-ring wear, with three piston-cooling arrangements, giving high (600 °C), medium (350 °C) and low (290 °C) piston temperatures.
The results show that as regards fuel consumption, exhaust smoke density, piston-ring wear, piston and piston-ring fouling, conditions are more favourable with the high-temperature piston than with the low-temperature and medium-temperature pistons.

Introduction

To obtain information on the conditions favourable for the combustion of residual fuels in the combustion chamber of a diesel engine as regards piston fouling and piston-ring wear, tests have been made in the Internal Combustion Engine Laboratories, of the Technological University Delft, with a view to ascertaining the influence of piston temperature on these two phenomena.
Three types of pistons were made for these tests, the piston-head temperatures under full-load conditions being between 290° and 600 °C.
After a running period of 50 hours with these three types of piston, the fouling of pistons, piston rings and other parts was ascertained. An attempt was also made to determine the degree of fouling by photographing the parts concerned. In the assessment of fouling, it was necessary to bear in mind that the cylinder lubrication oil consumption of the test engine was high (4.5 gram/eff. hp hour).

Test engine

The test engine, placed at our disposal by the Stichting Motorontwikkeling, Delft, was a two-cylinder, two-stroke diesel engine, with trunk piston and longitudinal scavenging. Some details of this engine are given below:

Cylinder diameter:	190 mm
Stroke:	350 mm
Effective power:	100 eff. hp at 430 rpm
Mean effective pressure:	5.2 kg/cm²

Nozzle orifice diameter: 0.55 mm
Cylinder 1 has a loose lining
Cylinder 2 is made without loose lining

Fuels

The test fuel used was a residal fuel with the analysis as shown below. Since the wear values obtained with this fuel were compared with the values when gas oil was used, the analysis of the gas oil employed is also given (Analyses by Chem.-Techn. Lab. Dr. Lobry de Bruyn).

	Residual fuel	*Gas oil*
Specific gravity 60 °F	0.9530	0.8249
Viscosity 100 °F	539 Redw. I	1.15 °E
210 °F	59.3 Redw. I	–
Congealing point	2 °C	–
Flash point P.M. (closed cup)	115 °C	75 °C
Carbon residue (Conradson)	7.27%	0.01%
Sulphur	2.8%	0.67%
Ash	0.016%	0.004%
Water	trace	trace
Sediment (extract)	0.013%	0.007%
Hard asphalt	4.82%	nil
ASTM distillation		
commencement	200 °C	185 °C
to 204 °C	-	5% vol.
232	0.5% vol.	29.5% vol.
260	2.5	62.0
288	8.5	84.0
315	17.0	96.0
340	–	99.0
343	29.5	–
371 (oil cracks)	43.0	–
Residue above 371 °C	67.0% vol.	
Cetane number (approx.)	32	53
C content	85.22%	85.69%
H content	11.02%	13.77%
Lower calorific value	9746 kcal/kg	10226 kcal/kg

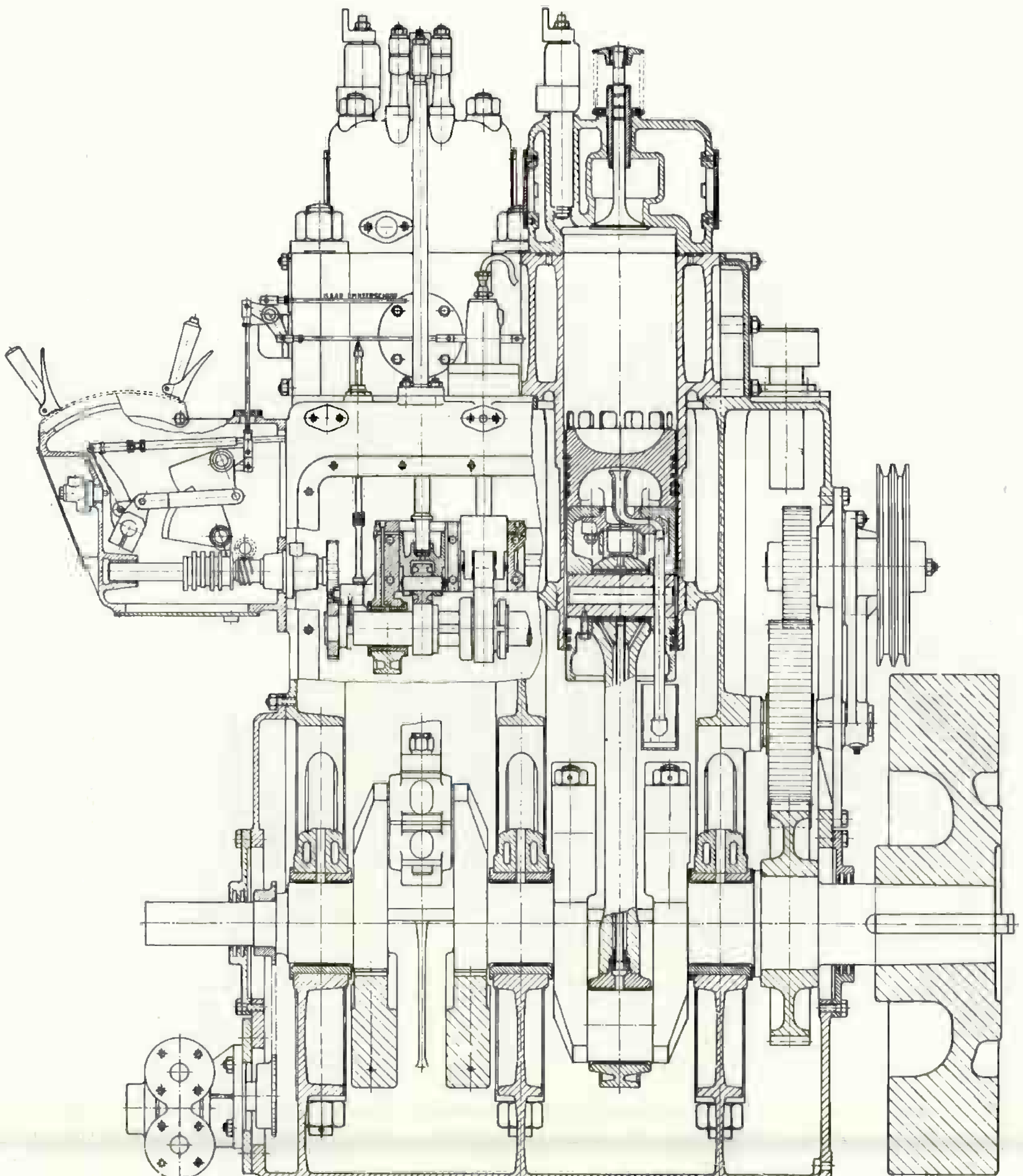


Fig. 1a. Longitudinal section of test engine.

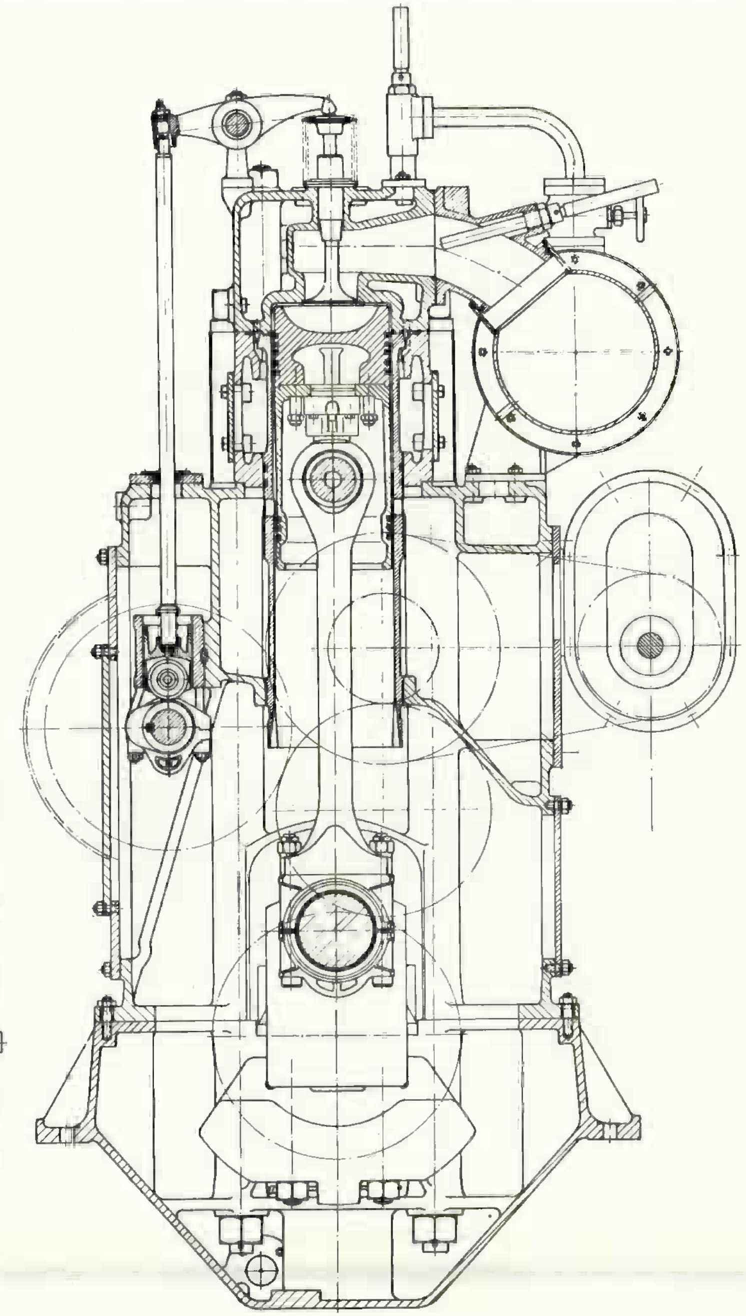


Fig. 1b. Cross section of test engine.

To ensure good atomization of the residual fuel, the latter was heated, so that its temperature on reaching the injector nozzles was 80 to 90 °C. At this temperature, the viscosity is 90–70 sec. Redw. I (3–2.7 °E).

The fuel was heated electrically, the heating elements (7200 W per tank) being situated in the day tank (temperature of fuel in day tank 100–110 °C).

The standard Bosch fuel pump (PF type) was provided with a second pipe connection to permit circulation of the heated fuel.

To prevent blocking of the fuel pipe by cooling, the pipe system was arranged so that it could be washed out with hot gas oil (70 °C).

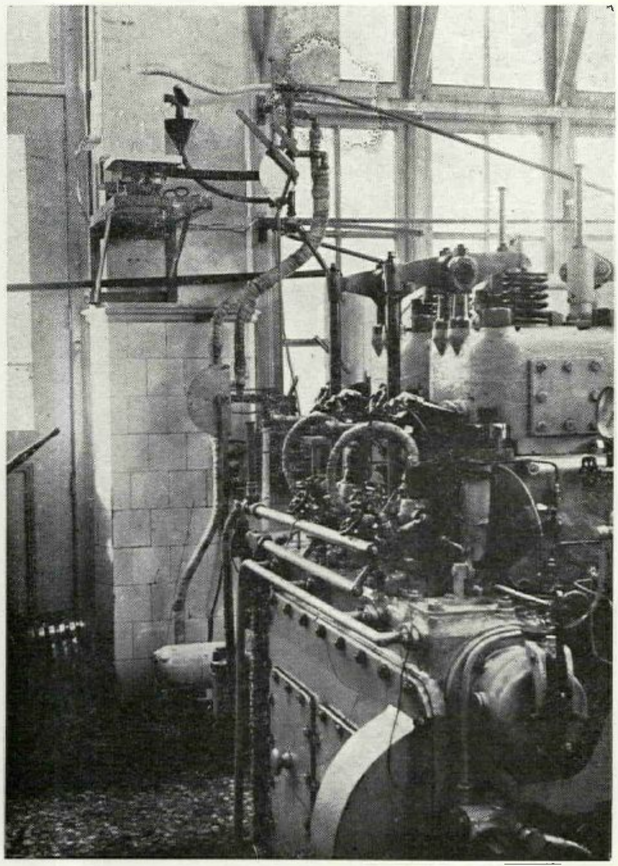


Fig. 2. Engine fuel supply and return pipes.

Nozzles

The nozzles (Bosch type DLF, orifice diameter 0.55 mm) were cooled with gas oil. A thermocouple was provided for checking the fuel temperature on entry into the nozzle holder.

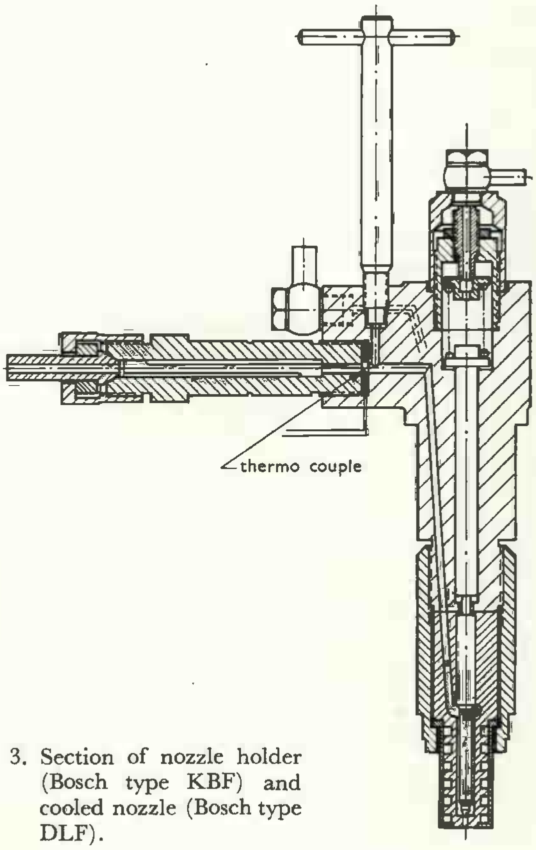


Fig. 3. Section of nozzle holder (Bosch type KBF) and cooled nozzle (Bosch type DLF).

Lubricating oil

The lubricating oil used was Shell Talpa 30. This oil was also used for cylinder lubrication. The consumption of cylinder lubricating oil was 4.5 gram/eff. hp hour. This is rather on the high side, but the engine seizes if the lubricating oil pumps are adjusted to give a lower supply.

The crankcase oil was renewed before each test (crankcase capacity 50 litres). After test, samples of lubricating oil were taken. The sample from the first test was analysed, but the result was not of interest.

Pistons

To obtain a variation in piston temperature, three types of pearlitic cast iron pistons were made; in what follows, these will be referred to as H, M and C pistons (hot, medium and cold).

The M and C pistons were cooled by lubricating oil. In the case of the C piston, the transition from piston head to skirt is recessed rather more, so that the temperature of the C piston head is lower than that of the M piston head.

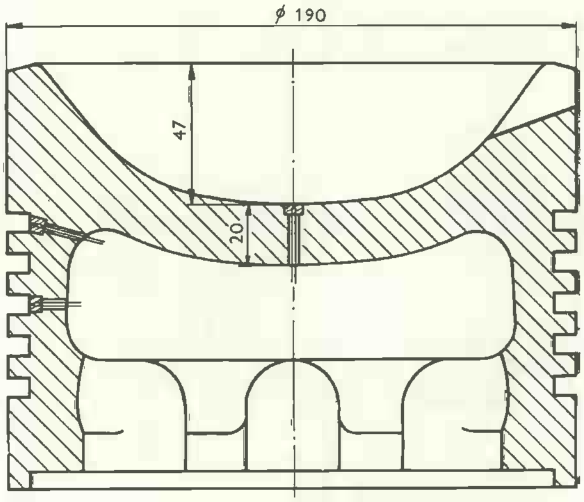


Fig. 4. C piston head.

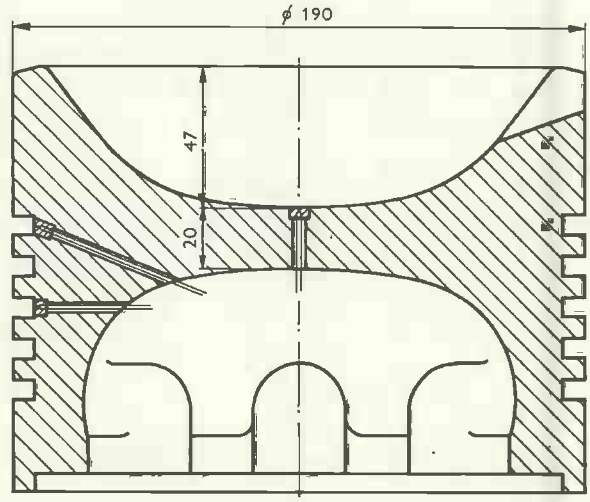


Fig. 5. M piston head.

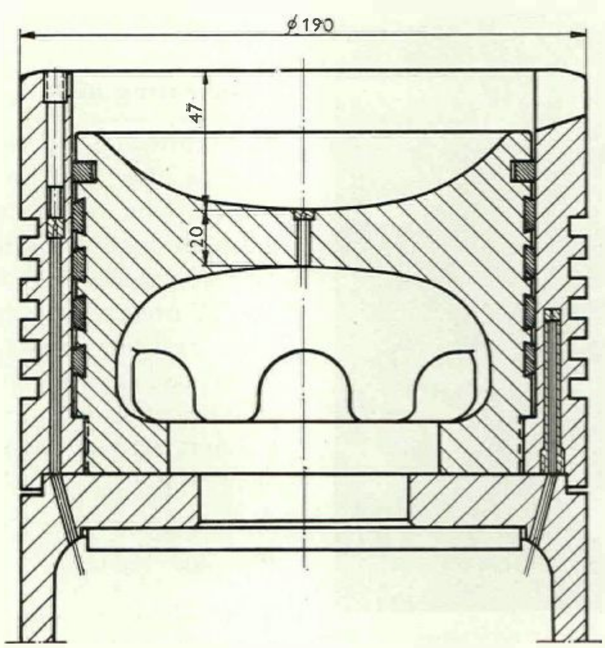


Fig. 6. H piston head.

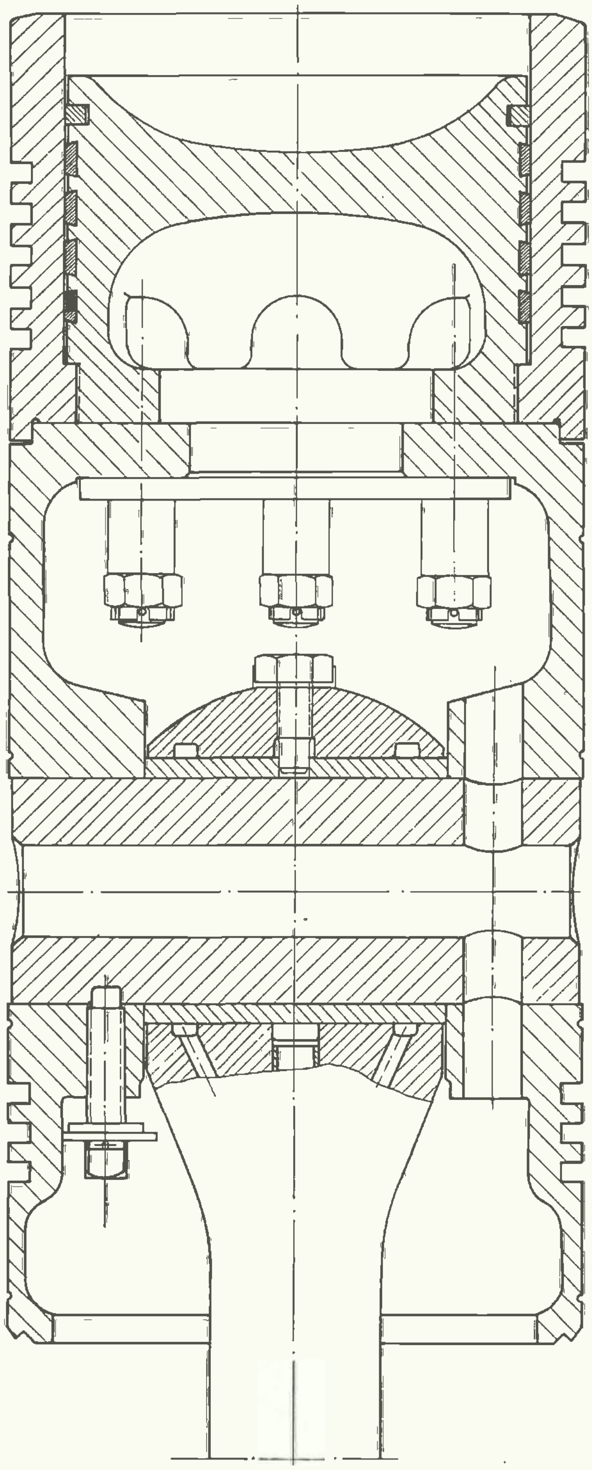


Fig. 7. Construction of H piston.

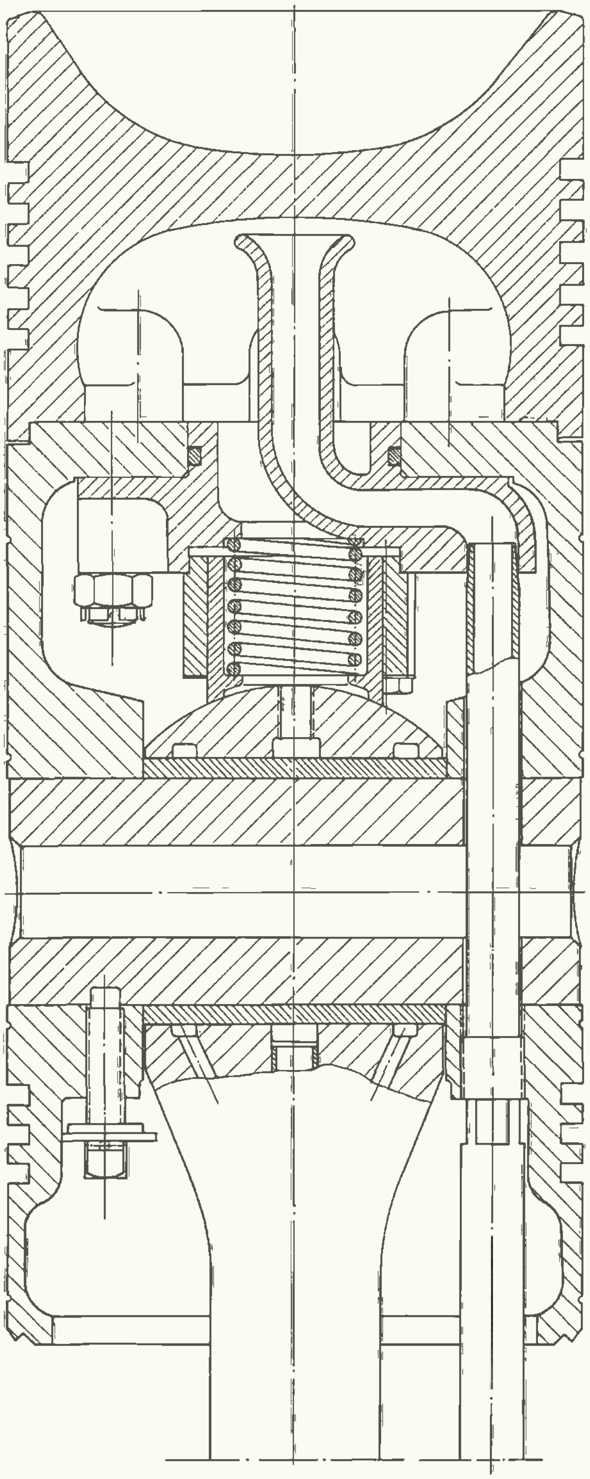


Fig. 8. Construction of M piston.

The H piston was not cooled. To ensure a high pistonhead temperature at low loads, the H piston was made of a special form. The piston head consists of a pot, the side wall of which is provided with copper bands. When the pot reaches a high temperature, these bands are pressed against the piston wall which comprises the piston rings; this results in a better dissipation of heat. At low loads, the pot contracts, the bands yield and the transmission of heat to the piston rings is reduced. The purpose of this special H piston construction was to obtain constant pistonhead temperature at all loads.

Measurement showed that the temperature was by no means constant. Furthermore, under full load, the pot temperature was found to be low in comparison with the temperatures of the C and M pistons. The bands were therefore machined flush with the side of the pot.

Some time was spent in endeavouring to obtain a different temperature-regulating effect of these bands by altering the dimensions, but these attempts failed to produce any results.

The temperature distribution in these three pistons was determined by means of fusible plugs under full load, using gas oil. Fig. 9 shows the results of these measurements.

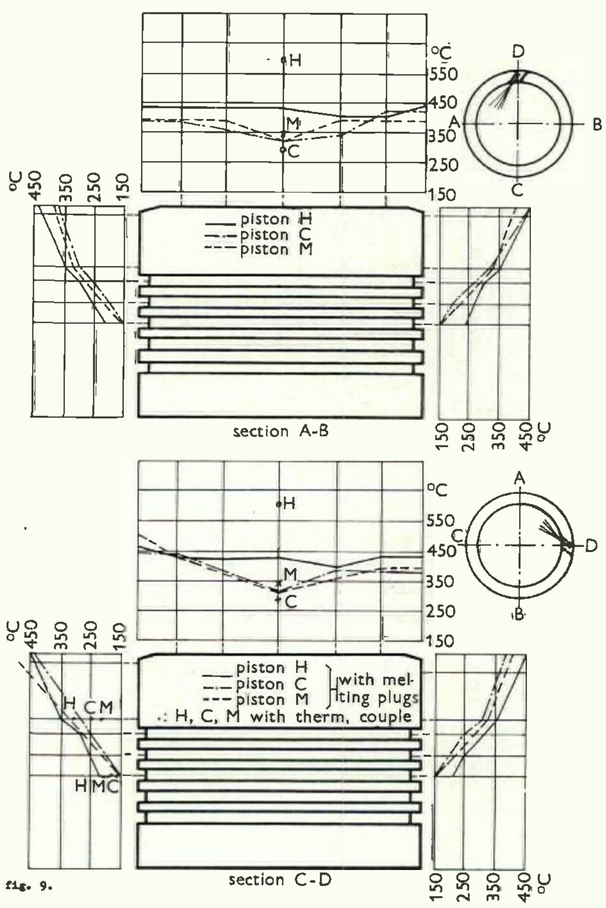


Fig. 9. Mean temperature variation determined by means of fusible plugs (fuel: gas oil).
100 bhp; n = 430 rpm.

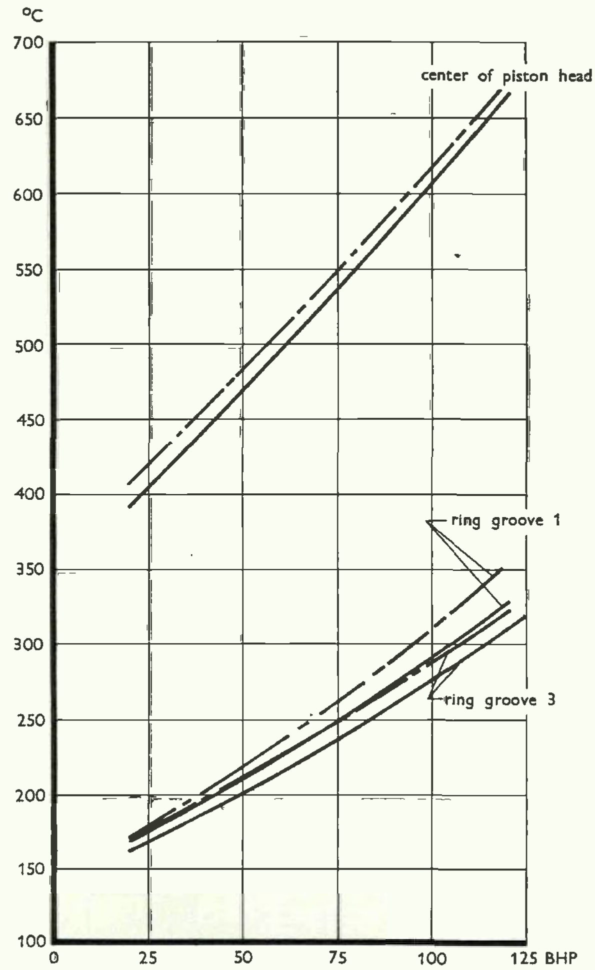


Fig. 10. Measurement of H piston temperature; n = 430 rpm.
——— gas oil —.—. residual fuel

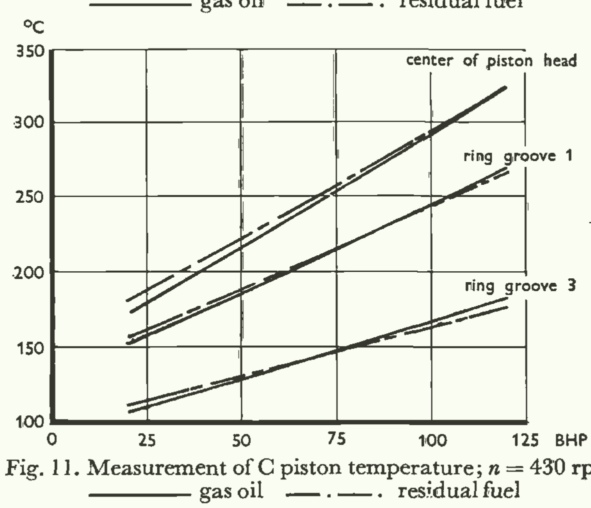


Fig. 11. Measurement of C piston temperature; n = 430 rpm.
——— gas oil —.—. residual fuel

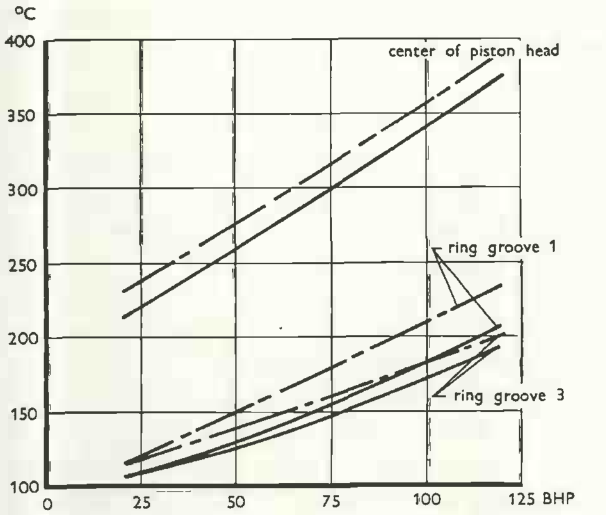


Fig. 12. Measurement of M piston temperature; $n = 430$ rpm.
——— gas oil —·— residual fuel

Since it is not always possible to ascertain exactly the melting of the plugs, and this method of measurement requires much time for assembly, thermocouples were mounted in the centre of the piston head on a level with the 1st and 3rd piston rings (see Figs. 4, 5 and 6). The thermocouple voltage was measured by means of contacts under the piston, when the piston was in the lower dead centre position.

A feature of the results of these measurements which is at once noticeable is that at full load, the temperature in the centre of the H piston head is 150 °C higher than when measured with fusible plugs.

With the fusible plugs, after warming up for 15 min. under 25% load, the engine was run for 30 min. on full load. Measurements with the thermocouples showed that 15 min. are necessary to reach temperature equilibrium, the load being increased or reduced by 25%. At the high temperatures of the H piston, the measuring time with the fusible plugs was apparently too short. The very short measuring time and indefinite melting of the high-temperature fusible plugs are the causes of the difference between the two methods of measurement.

In the low-temperature range, the two temperature-measuring methods were in agreement (difference $\pm$ 40 °C, see Fig. 9).

Test programme

General experience shows that the minimum running time required for obtaining a reliable picture of fouling and wear can be considered to be 50 hours. In the circumstances, it was not possible to run the engine continuously for 50 hours.

The period of 50 hours was run in 6 days. The same procedure was adopted as far as possible when starting and stopping the engine, in addition, the endeavour in all the tests was to have the same cooling water temperature and lubricating oil inlet temperature.

Before the commencement of each endurance test, the combustion chambers, pistons, exhaust ports and piston rings were cleaned, new exhaust valves were mounted to afford better comparison and the crankcase was replenished with 50 litres of Talpa 30 lubricating oil.

The following endurance tests were made:

	H piston	M piston	C piston
Residual fuel	100 hp/430 rpm	100 hp/430 rpm	100 hp/430 rpm
	30 hp/430 rpm	–	30 hp/430 rpm
	100 hp/430 rpm	–	–
Gas oil	–	–	100 hp/430 rpm

In the 100% load tests, the best results as regards wear and fouling were obtained with the H piston. To obtain evidence of the fact that delayed fuel injection is particularly detrimental when heavy fuels are used, an endurance test was repeated with delayed injection, produced by using a pressure valve with a small relief volume.

As basis for comparison, an endurance test under 100% load with the H piston was originally on the programme for gas oil. In the preparatory activities, one of the H pistons was burnt through. A 100% endurance test with gas oil was then made with the other extreme, the C piston.

Engine during the tests

Preparatory to a 50 hour endurance test, the fuel consumption, scavenging pressure, exhaust temperature and smoke curves were determined for different loads at 430 rpm with gas oil and residual fuel. In changing over from gas oil to heavy fuel, no alteration was made in the engine or its adjustments, so as to express only the influence of the different fuels on wear and fouling.

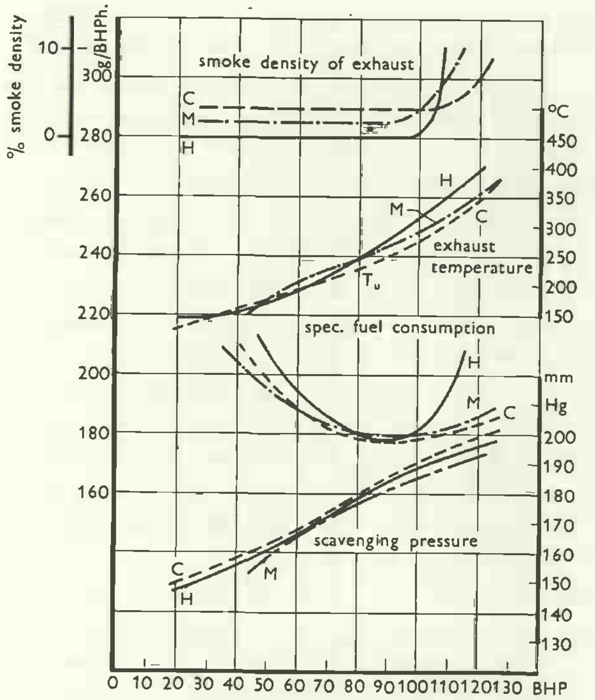


Fig. 13. Curves showing comsumption, scavenging pressure and exhaust temperature using gas oil.

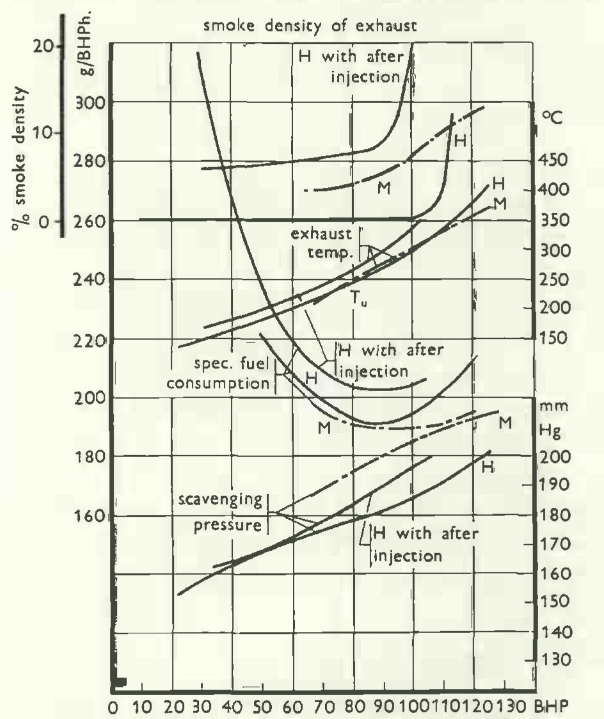


Fig. 14. Curves showing consumption, scavenging pressure and exhaust temperature using heavy fuel.

Gas oil

With gas oil, the minimum consumption was practically the same for the three types of piston (178 gram/eff. hp hour). The measurements also showed that the range of satisfactory consumption diminishes considerably when the H piston is used.

Heavy fuel

With the use of heavy fuel, the minimum consumption was higher, being 190 gram/eff. hp hour. In view of the lower calorific values of the residual fuel, a minimum consumption of 187 gram/eff. hp hour was to be expected.
Due to circumstances, the consumption curve for the C piston was not determined.
Fig. 14 also shows that the smoke values are better with the H piston than with the M piston. With delayed injection, the smoke figures are very unsatisfactory and consumption is high, indicating poor combustion. Smoke figures of 0–7% indicate a colourless exhaust.
The scavenging pressure appears to be considerably increased when running on heavy oil with the M piston, due to more sluggish combustion of the fuel.
The above consumption curves also show the values of the consumption, exhaust temperature, etc. recorded during the endurance tests.
The temperatures were measured by means of thermocouples at different points in the engine, viz:

- temperature in the body of the valve disc (at a depth of 1 mm in the material);
- temperature of injector washer;
- temperature of cylinder wall (injector side) (Fig. 15);
- temperature of cylinder cover wall (exhaust side) (Fig. 15).

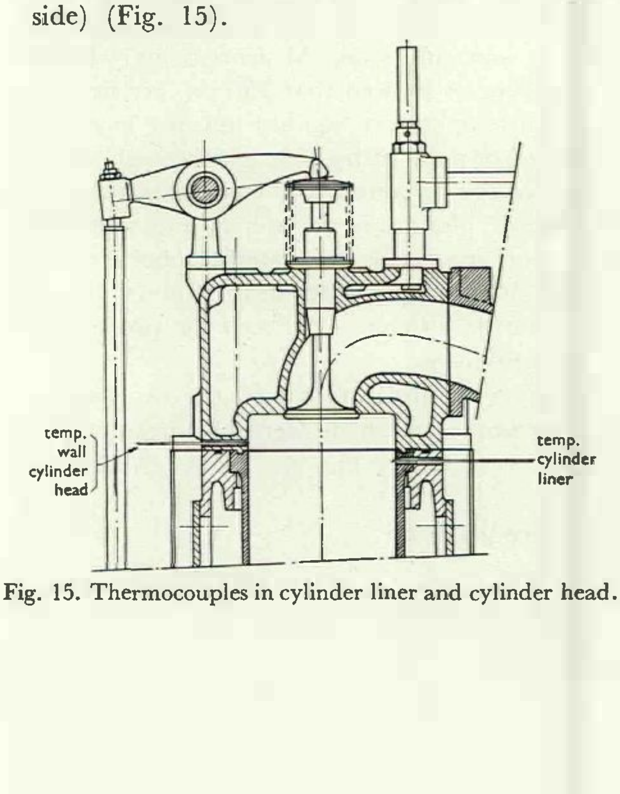


Fig. 15. Thermocouples in cylinder liner and cylinder head.

In the first-mentioned measurements, the thermocouple wires were supported by a strip of spring steel, see Fig. 18.
Figs. 16 and 17 show the temperatures measured for different loads at 430 rpm.

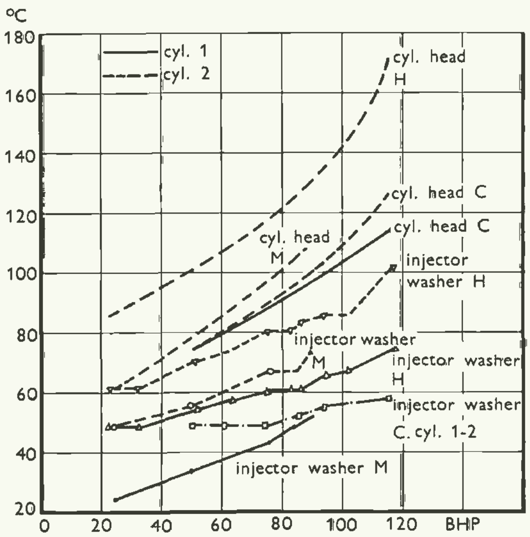


Fig. 16. Temperature of cylinder head and injector washer.

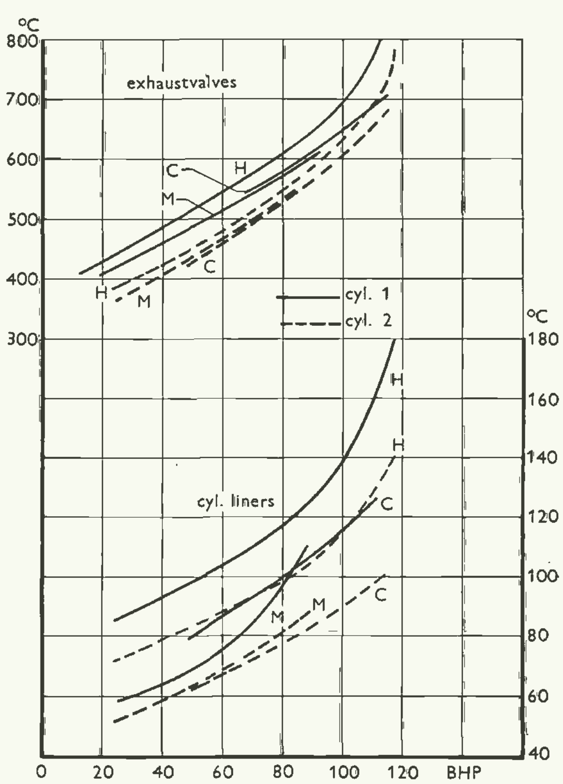


Fig. 17. Temperature of exhaust valves and liners.

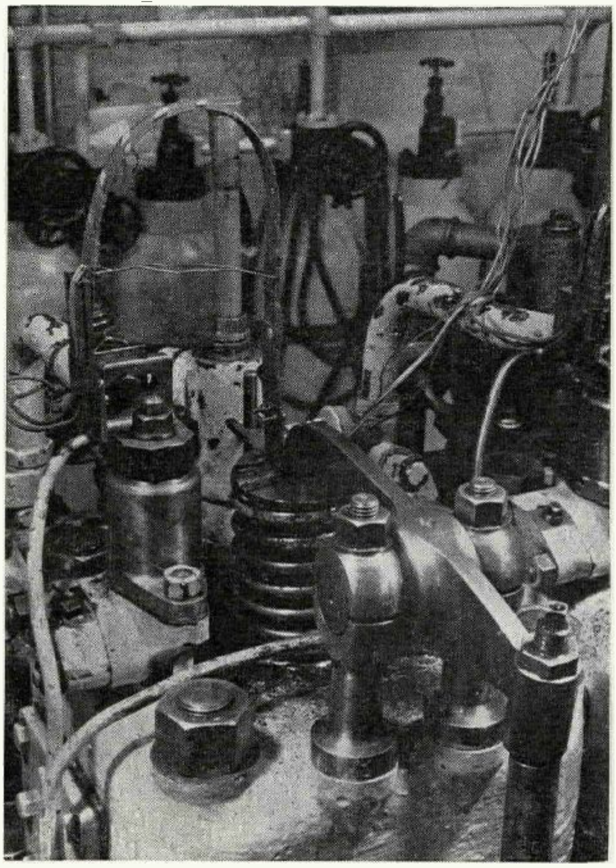


Fig. 18. Spring steel strip support of thermocouple wires to exhaust valve.

Test results

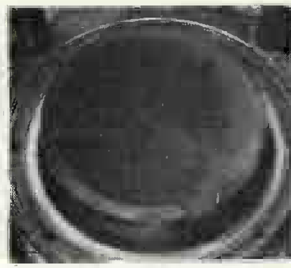
Fouling: By photographing the principal parts exposed to fouling, an endeavour has been made to give an impression of the degree of fouling after the tests. The parts photographed are:

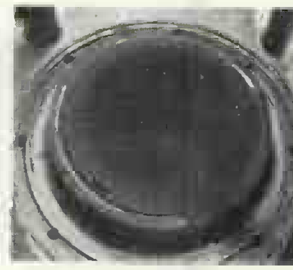
Fig. 19: a. Piston head;
- b. Piston, front view;
- c. Cylinder head;
- d. Exhaust valve;
- e. Exhaust valve seat;
- f. View of piston in cylinder head.

Fouling after 50 hours run at 100 bhp and 430 rpm
Examination of the complete series of photographs (Fig. 19, a to f) brings out the striking feature that for *pistons and piston ring assembly*, fouling after the endurance test with the H piston using heavy fuel can be assimilated to that after the endurance test with gas oil using the C piston. Fouling after the endurance tests on heavy fuel using the M and C pistons is decidedly much greater. There is no difference in fouling between the H pistons with and without delayed injection.

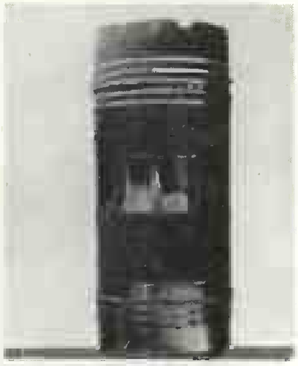

H piston with heavy fuel


M piston with heavy fuel


C piston with heavy fuel

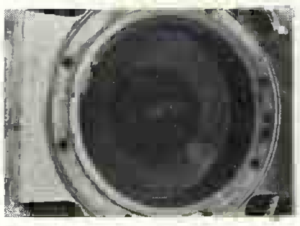

C piston with gas oil


H piston with heavy fuel


M piston with heavy fuel

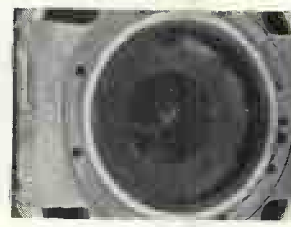

C piston with heavy fuel


C piston with gas oil


H piston with heavy fuel


M piston with heavy fuel


C piston with heavy fuel

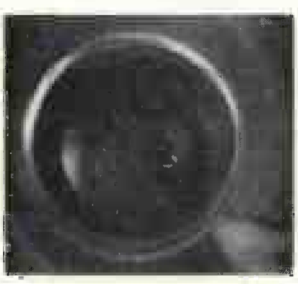

C piston with gas oil

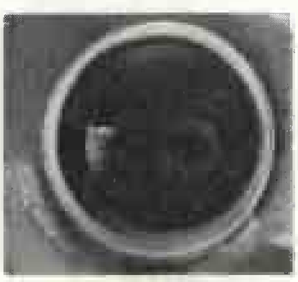

H piston with heavy fuel

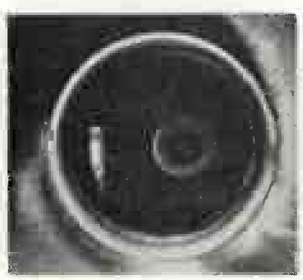

M piston with heavy fuel


C piston with heavy fuel


C piston with gas oil


H piston with heavy fuel


M piston with heavy fuel


C piston with heavy fuel


C piston with gas oil


H piston with heavy fuel

Not photographed


C piston with heavy fuel

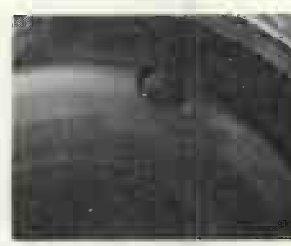

C piston with gas oil

Fig. 19

a. Fouling of piston head after 50 hrs run at 100 bhp/430 rpm.

b. Fouling of piston and piston rings after 50 hrs run at 100 bhp/430 rpm.

c. Fouling of cylinder head.

d. Fouling of exhaust valve.

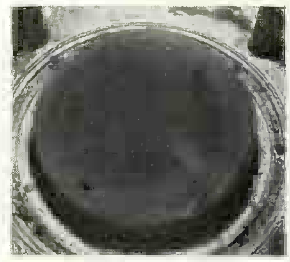
e. Fouling of exhaust valve seat.

f. Fouling of injector in cylinder head.

Fig. 20

a. Fouling of piston head after 50 hrs run at 30 bhp/430 rpm.


H piston with heavy fuel

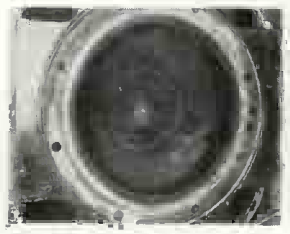

C piston with heavy fuel

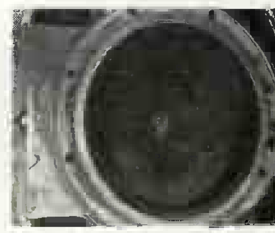
b. Fouling of piston and piston rings after 50 hrs run at 30 bhp/430 rpm.


H piston with heavy fuel


C piston with heavy fuel

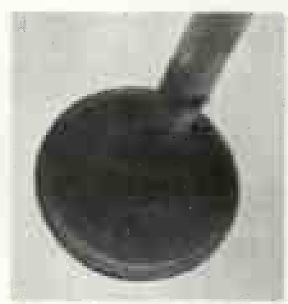
c. Fouling of cylinder head.


H piston with heavy fuel

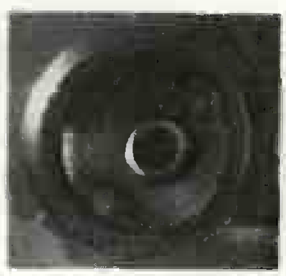

C piston with heavy fuel

d. Fouling of exhaust valve.


H piston with heavy fuel


C piston with heavy fuel

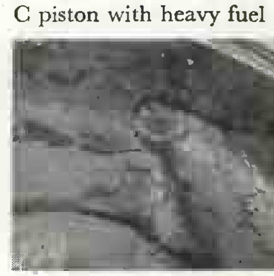
e. Fouling of exhaust valve seat.


H piston with heavy fuel


C piston with heavy fuel

f. Fouling of injector in cylinder head.


H piston with heavy fuel


C piston with heavy fuel

On dismantling after the tests, complete and partial sticking of the following piston rings was found. In this table, the part of the piston ring found to be stuck is expressed as a percentage of the circumference.

this impression is quite certainly due to vanadium pentoxide. Apparently, this effect does not occur during combustion at the high temperatures associated with the H piston.

Fouling of the other parts does not provide

	H piston delayed injection	Residual fuel			Gas oil
		H piston	M piston	C piston	C piston
Cylinder 1, ring 1	100%	25%	100% (fracture after dismantling)	100% (fracture after dismantling)	50%
ring 2	100%	25%	100% (fracture after dismantling)	–	–
ring 3	50%				
ring 4	100%				
ring 5	25%				
Cylinder 2, ring 1	100%	100% (fracture after dismantling)	100% (fracture after dismantling)	100% (fracture after dismantling)	75%

The other piston rings did not show any ring sticking.

Ring sticking with the H piston after test with residual oil is worse than after test with gas oil, but not so bad as after tests with the M and C pistons.

Cylinder heads: The appearance of the exhaust valves creates the impression that they had been very hot in the tests on heavy fuel with the M and C pistons. This impression is not gained from the test on heavy fuel with the H piston and the test on gas oil. The whitish grey deposit which produces

sufficient difference to permit reliable comparison.

Fouling after 50 hours run at 30 bhp and 430 rpm. Fouling after this low power is greater than after the tests on full power (see Fig. 20, e to f). Fouling after tests with the H piston is rather less than after the test with the C piston.

For both pistons, fouling of the piston ring assembly is considerable; the piston-ring temperatures were too low, however, to give rise to ring sticking.

Wear. Wear of the piston rings was obtained from the loss in weight of the rings during the 50 hours test.

Wear after full-power tests. The following graphs show the loss in weight of the piston rings after the tests:

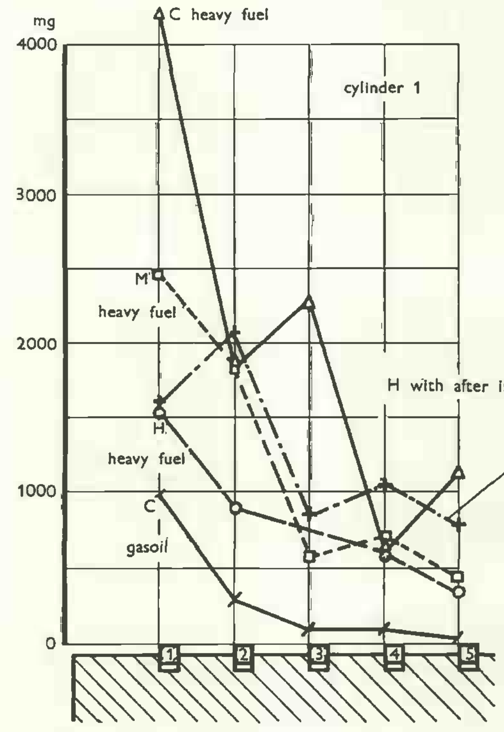


Fig. 21. Piston-ring wear, cyl. 1; 100 bhp, 430 rpm.

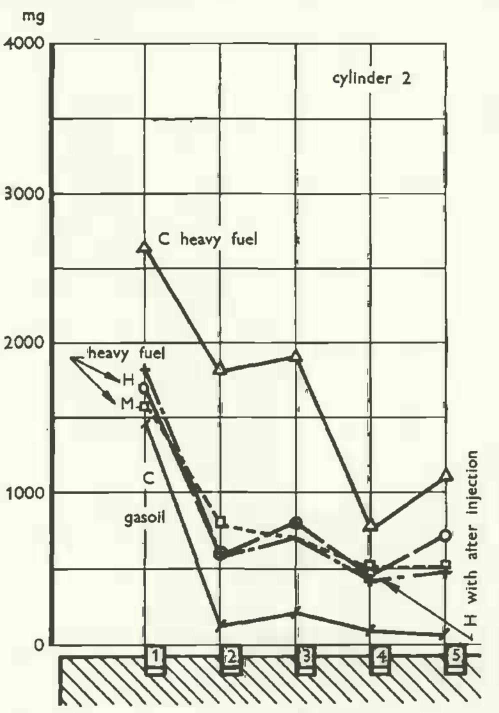


Fig. 22. Piston-ring wear, cyl. 2; 100 bhp, 430 rpm.

The lower wear figures after use of the H pistons are clearly to be seen. The mean values of piston-ring wear for the two cylinders are shown in Fig. 23.

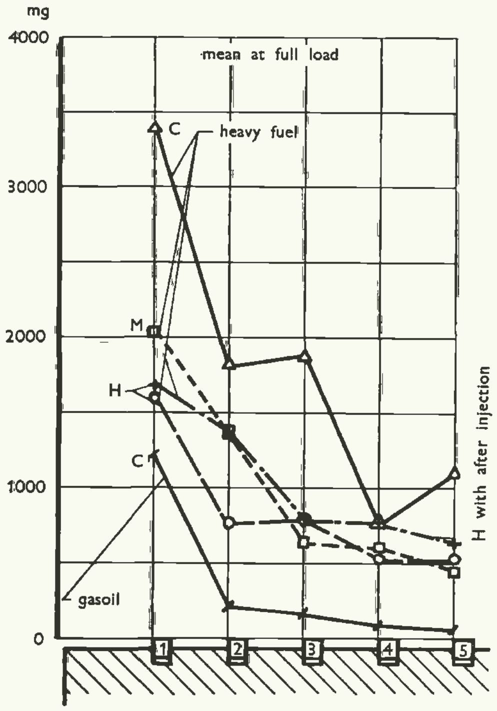


Fig. 23. Mean piston-ring waer, 100 bhp, 430 rpm.

When heavy fuel is used, piston-ring wear is in all cases a maximum for the C piston and a minimum for the H piston. These minimum values, however, are much higher than when gas oil is used.

Delayed injection was not found to have any considerable effect.

Piston-ring wear after 30% power test. The difference in piston-ring wear after the endurance tests with H and C pistons is so slight that the values can be regarded as identical.

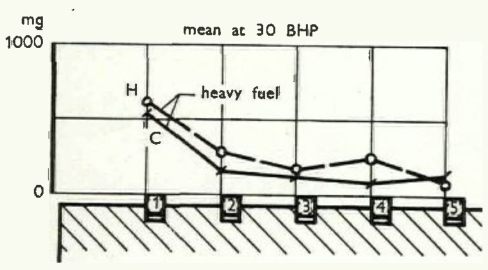


Fig. 24. Mean piston-ring wear, 30 bhp, 430 rpm.

Conclusions and further comments on the wear results

The conclusion that may be drawn from the results is that high piston-head temperatures have a favourable effect as regards fouling and piston-ring wear. Even though these tests were extensive and took a considerable length of time, they still failed to show any clear effect of combustion on fouling and piston-ring wear. A higher piston-head temperature (more satisfactory combustion) was associated with a higher piston-ring temperature. Either the more satisfactory combustion or the higher piston-ring temperature may have brought about the reduction in wear and fouling.

The appearance of the valves shows that no corrosive action is to be expected at high wall temperatures of the combustion space.

Piston-ring wear of the H piston when using gas oil was not measured. In connection with the experiments with the hot pot in the piston, the latter was run for 65 hours on gas oil. If the values for the piston-ring wear during these experiments are converted to a 50 hour run, the values given in the following Figure are obtained.

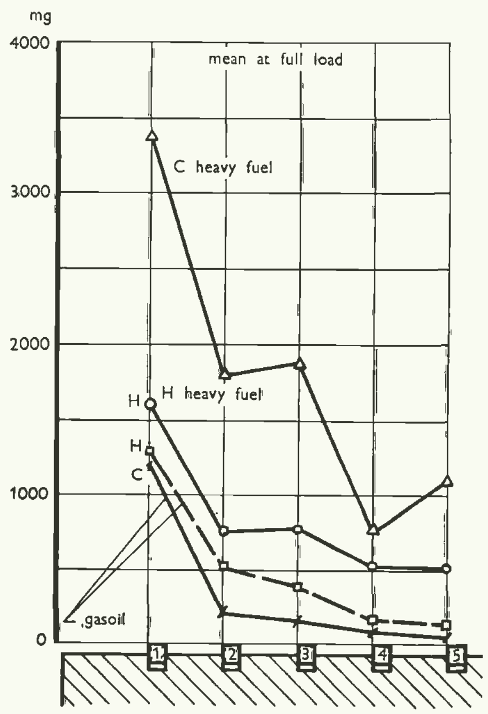


Fig. 25. Mean piston-ring wear after 50 hrs run on heavy fuel and gas oil, H and C pistons, 100 bhp/430 rpm.

These figures are higher than those for the C piston using gas oil, but the engine was started up many times during the above-mentioned 65 hours.

It is possible from the piston-ring wear values to calculate the ratio $\frac{\text{wear with heavy fuel}}{\text{wear with gas oil}}$. This value is low for the top piston ring and increases for the lower rings; the ratio also increases with decreasing piston-ring temperature.

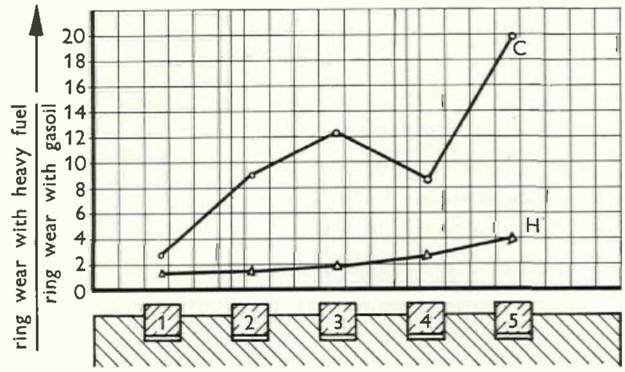


Fig. 26. Ratio of piston-ring wear for heavy fuel and gas oil.

If these ratios are related to the measured temperature of the piston-ring grooves, which for the purpose of this discussion is assimilated to the piston-ring temperature, the ratio is found to decrease rapidly with increase in temperature up to 250 °C .This decrease becomes less above 250 °C.

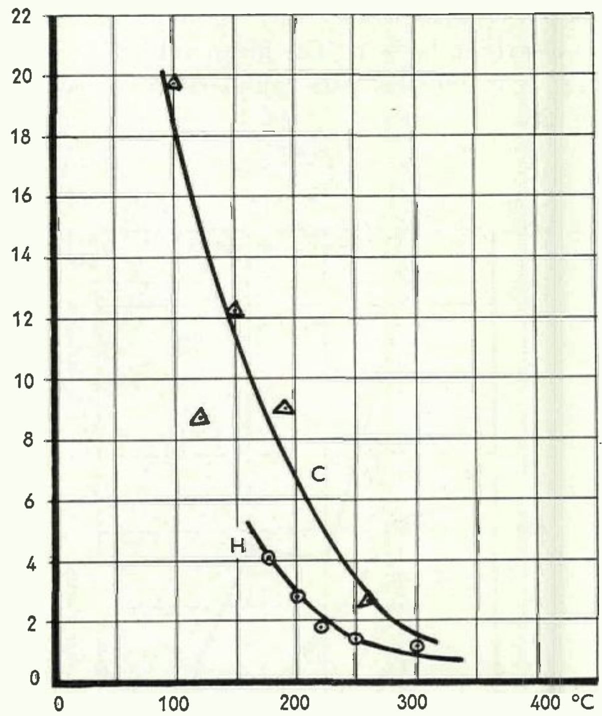


Fig. 27. Ratio of piston-ring wear for heavy fuel and gas oil at different temperatures of the H and C piston-ring grooves.

It can be concluded from this that the piston-ring temperature must be at least 200° to 250 °C if wear is to be kept as low as possible when using heavy fuels. This requirement is contrary to that imposed for the prevention of piston-ring sticking.