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INFLUENCE OF LOAD ON LINER TEMPERATURE AND WEAR IN A HIGH SPEED DIESEL ENGINE

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SUMMARY

Wear tests conducted with high speed diesel engines indicated that engines running on varying loads suffered more wear than engines running on full load most of the time. It was suspected that the increased wear was due to increased corrosion caused by low liner temperatures at part load conditions. To verify this, liner temperatures were measured in a single cylinder high speed diesel engine run at various loads, coolant outlet temperatures and rates of coolant circulation.

It is concluded from these experiments that, for the coolant outlet temperatures currently employed in high speed diesel engines, the liner temperature will be low enough at part loads to cause increased corrosive wear. This can be countered by employing coolant temperatures of the order of 105–110° C. which can be obtained in a closed cycle coolant system by moderate pressurisation of 5–10 psig.

INTRODUCTION

From the wear tests conducted in the Internal Combustion Engineering Department, Indian Institute of Science, with two single cylinder diesel engines—one run at varying load and the other at constant load, it was observed that the wear was more in the engine with a varying load than in the engine with a constant load—Table I (Fig. 1). A subsequent wear test with the former engine running continuously at full load indicated that the wear was reduced. These engines were identical in every mechanical detail. The fuel and lubricating oil used were the same. The cooling systems and the coolant temperatures were the same as also the number of cold starts. The only difference was that while one engine was run at full load for 90% of the running time, the other was run at full load for 50% of the total

TABLE I

Wear Tests—Test Schedule

Both the engines were run for 16 hours a day and 6 days a week, as per the following daily schedule

	Engine No. 1 (No. 20160)	Engine No. 2 (No. 20638)
Warm up ..	½ hour : 6-00 to 6-30 a.m.	½ hour : 6-00 to 6-30 a.m.
Full load ..	7½ hours : 6-30 to 10-30 a.m.; 3-30 to 7-00 p.m.	14½ hours : 6-30 a.m. to 6-30 p.m.; 7-30 to 10-00 p.m.
Half load ..	7 hours: 10-30 a.m. to 2-30 p.m. ; 7-00 to 10-00 p.m.	..
10% overload ..	1 hour : 2-30 to 3-30 p.m.	1 hour : 6-30 to 7-30 p.m.
Total hours per day ..	16 hours	16 hours

RING WEAR

	Engine No. 1						Engine No. 2			
	0-500 hrs.	500-1000 hrs.	1000-1500 hrs.	1500-2000 hrs.	2000-2500 hrs.	2500-3000 hrs.	0-500 hrs.	500-1000 hrs.	1000-1500 hrs.	1500-2000 hrs.
PISTON RING WEAR :										
(*a*) Increase in gap Compression Ring No. 1 ..	0·021″	0·030″	·004″	·007″	·013″	·001″	0·006″	0·006″	·003″	·001″
do. 2 ..	0·018″	0·014″	·004″	·013″	·006″	·002″	0·006″	0·006″	·003″	·009″
do. 3 ..	0·008″	0·009″	·003″	·012″	·004″	·002″	0·007″	0·005″	·003″	·014″
Oil Scraper Ring ..	0·010″	0·009″	·005″	·031″	·014″	·012″	0·008″	0·010″	·003″	·015″
(*b*) Loss in weight : Gm. Compression Ring No. 1 ..	0·428	0·537	·124	·281	·314	·240	0·250	0·251	·100	·180
do. 2 ..	0·442	0·200	·216	·351	·160	·110	0·213	0·128	·108	·177
do. 3 ..	0·209	0·138	·135	·239	·086	·043	0·106	0·063	·079	·133
Oil Scraper Ring ..	0·187	0·105	·079	·430	·090	·133	0·054	0·039	·027	·086

There was no noticeable difference in the wear of other components.

running time, and at part load during most of the remaining period. It was suspected that the increase in wear of this engine might be due to increased corrosive wear caused by low operating temperatures of the liner and other parts during part load runs. It was therefore decided to determine the temperatures of some of these components under all possible operating conditions of the engine.

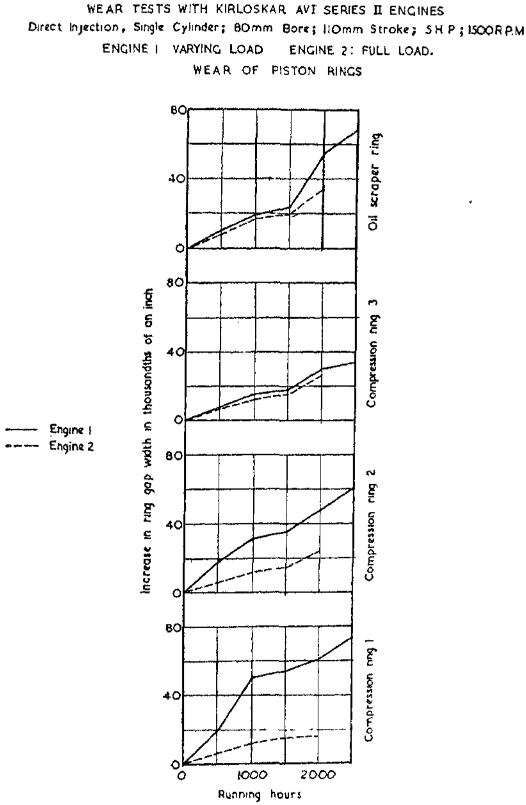

FIG. 1

SCOPE OF THE PRESENT WORK

The present work was concerned with measuring the temperature of the liner at a point corresponding to the T.D.C. position of the first compression ring, this region being the most affected by temperature.

In a constant speed engine the operating variables are the load, the coolant temperature and the rate of coolant circulation. The liner temperature was proposed to be measured when these operating conditions were varied.

TEST SET-UP

A single cylinder high speed diesel engine (5 H.P.—1,500 R.P.M.) similar to those used in the foregoing wear tests was coupled to an electric dynamometer. Standard instrumentation for measuring temperatures, fuel flow, speed, etc., was provided.

The cooling system was so arranged that any one of the three following schemes could be employed as desired:—

(1) Closed cycle cross-flow in cylinder head with natural convective flow in the cylinder barrel;

(2) Open cycle cross-flow in the cylinder head with natural convective flow in the cylinder barrel, cold water-supply being from the mains; and

(3) Open cycle Z flow with cold water from the mains entering at the bottom of the cylinder barrel and hot water going out in the cylinder head (Fig. 2).

The rate of circulation could be found out by means of a flow meter inserted in the closed cycle system and by direct measurement in the other systems.

Thermocouple and its location in the liner.—An iron constantan thermocouple was used in conjunction with a potentiometer. The thermocouple wires were first sheathed in fibre glass and then in a fine stainless steel tube. The thermocouple was located in the liner in the following way. At a point corresponding to the T.D.C. position of the first compression ring and in a direction parallel to crankshaft axis, a 3/32″ diameter hole was drilled through the cylinder block into the liner to a depth of 1/32″ from the inner face of the liner. The hole in the block was enlarged and tapped to $\frac{1}{4}''$ B.S.F. size. A $\frac{1}{4}''$ B.S.F. brass screw of the shape indicated in Fig. 3, with an inner hole to match the metal sheath of the thermocouple was fitted to the cylinder block and locked in position with a locknut. The thermocouple was passed through the inner hole of the brass screw to its position in the liner and locked in position by means of a glandnut at the end of the brass screw (Fig. 3). An ice-bath served as the cold junction.

Figure 4 is the pictorial representation of the test set-up.

TEST PROCEDURE

The engine was warmed up and loaded to a predetermined value. Keeping the load constant, the coolant outlet temperature was varied from 65° C. to the highest possible value in steps of 5° C. For each of these coolant temperatures the following observations were recorded: Engine speed, liner temperature, fuel

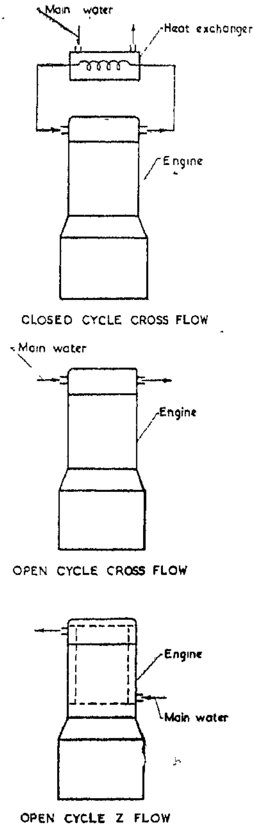

Fig. 2. Engine Cooling System.

consumption, exhaust temperature, heat given to coolant. At this load and at each of these coolant temperatures the rate of coolant circulation was varied, and for each rate of circulation the above observations were repeated.

This procedure was repeated for each of the following loads: zero, ¼, ½, ¾, 1, and 1·1 full load. The above procedure was followed for all the three types of cooling system.

B.S. grade A fuel and S.A.E. 30 heavy duty lubricating oil were used for all tests.

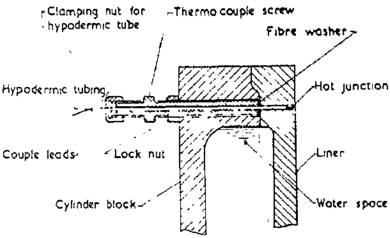

FIG. 3. Thermo-Couple Arrangement for Measuring Liner Temperature.

RESULTS

The results are presented in Figs. 5, 6, 7, 8 and are summarized here:

A. Closed cycle cross-flowsystem:

(1) Liner Temeperature *vs.* Rate of Coolant Circulation. For a given coolant outlet temperature the liner temperature was practically independent of rate of coolant circulation (Fig. 5).

(2) Liner Temperature *vs.* Coolant Temperature. For a given load the liner temperature increased with coolant temperature. This was true at all loads (Fig. 5).

(3) Liner Temperature *vs.* Load. For a given coolant temperature the liner temperature increased with load (Fig. 5).

(4) As the coolant temperature was increased there was a reduction in the heat lost to cooling water and an improvement in the specific fuel consumption and brake thermal efficiency. This was true at all loads (Fig. 6).

(5) The exhaust temperature was unaffected by changes in the coolant temperature (Fig. 6).

B. Open cycle cross-flow and open cycle Z flow cooling systems.

The results in these cases followed the general trend observed with the closed cycle cross-flow system but the actual values of the liner temperature were lower with open cycle cross-flow and lowest with open cycle Z flow. For example, at half

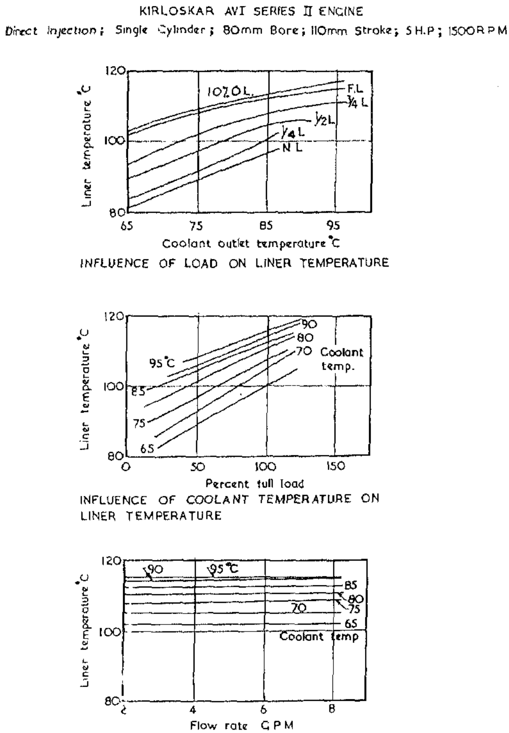

FIG. 5. Influence of Rate of Coolant Circulation on Liner Temperature.

load and 75° C. coolant temperature, the liner temperature was 97° C., 89·3° C. and 88·5° C. for closed cycle cross-flow open cycle cross-flow and open cycle Z flow respectively (Fig. 7).

DISCUSSION AND CONCLUSIONS

Whatever the cooling system employed, for a given coolant temperature, the liner temperature falls off as the load is reduced. For example, with a coolant temperature of 75° C.—a normal value in diesel engine practice—at half load, the temperature attained by liner is 97° C., 89·3° C. and 88·5° C. with the three systems of cooling as compared to 108·0° C., 100·1° C., 97·25° C. at full load.

Such low liner temperatures are conducive to the deposition of sulphur acids and consequent increased corrosive wear.

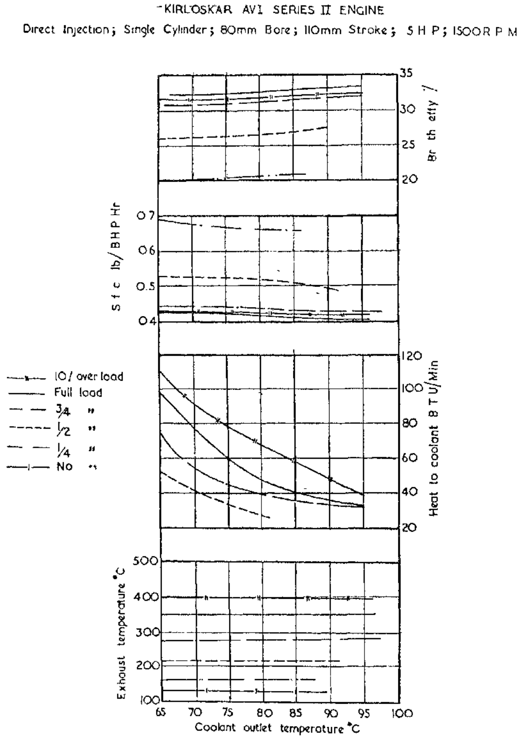

FIG. 6. Influence of Coolant Outlet Temperature on Exhaust Temperature, Heat to Coolant Specific Fuel Consumption and Brake Thermal Efficiency.

In the wear tests mentioned in the first part of this report, the cooling system employed was of the open cycle cross-flow type with a coolant outlet temperature of 75° C. Corresponding to this coolant temperature the liner temperature was 100·1° C. at full load and 89·3° C. at half load. Thus there was a drop of 11° C. at half load. The engine with the varying load was operated at this low liner temperature for a considerable period of the test. This may be the reason for he increased wear noticed in the engine and it can be attributed to corrosive wear. 'he reduction in wear when this engine was subsequently run on full load can be .dduced as corroborating evidence to support the above supposition.

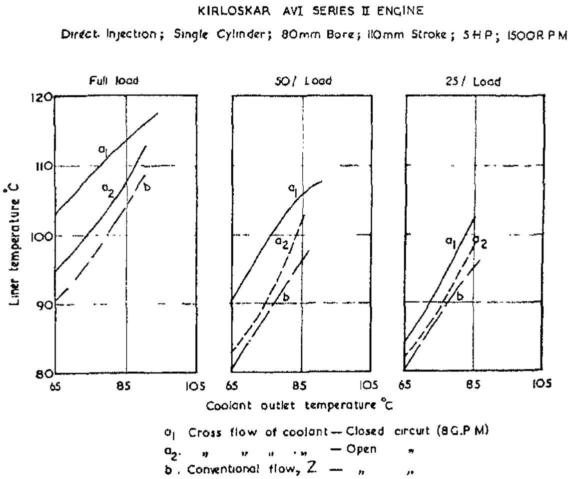

FIG. 7. Influence of Load on Liner Temperature with Different Cooling Systems.

Millington[1] reports that a minimum temperature of 130° C. is desirable at ie top of the liner. Groth[2] also supports this view. Broeze and Wilson[3] also upport the above view regarding the minimum temperature; they also recommend maximum temperature of 150° C. since, in their opinion, temperatures above this nay lead to excessive wear due to failure of lubrication, excessive lacquering, carbon ormation and ring sticking.

From the foregoing, it seems that the lower and higher limits for liner tempera-ure are 130° C. and 150° C. respectively. The present investigation shows that he liner temperature is 15–30° C. higher than the coolant temperature as the load icreases from light load to 10% overload. A coolant temperature of 115° C. vill produce a liner temperature which will be within the limits mentioned above t all load conditions. This high coolant temperature can be obtained with mode-ate pressurising of the cooling system by 5–10 psig. Best results can be expected rom the closed cycle cross-flow cooling system. Improved thermal efficiency s an additional advantage of employing higher coolant temperatures.

FUTURE WORK

Wear tests may be conducted with higher coolant temperature to verify the above conclusions.

ACKNOWLEDGMENTS

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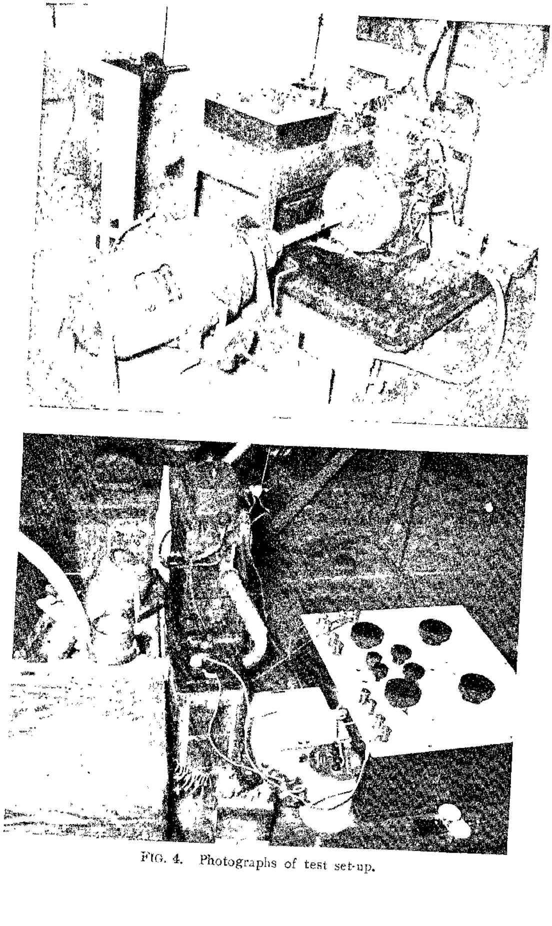

FIG. 4. Photographs of test set-up.