



A STUDY OF ENGINE WEAR AS INFLUENCED
BY LUBRICANT CONDITIONS

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ABSTRACT

In this work a comprehensive review of the basic lubrication mechanisms that would exist in engine lubrication will be discussed. The subsequent effect of lubricants (new or used) on engine wear is also, herein, summarized and followed by a description of test procedure on a specially constructed test machine to analyse wear under dry or lubricated contacts. Results have also been presented to compare the effect of change in oil properties and the presence of contaminants after prolonged time of use on the wear rate.

A detailed theoretical study is also given to assess the effect of engine wear on the hydrodynamic performance. The dimensional changes in crank bearings due to wear have been considered in applying hydrodynamic lubrication theories and their effect on the load carrying capacity, frictional resistance, oil flow rate and side leakage and maximum hydrodynamic pressure could be fully computed and analysed.

KEYWORDS

Engine Wear, Friction, Lubricant, Hydrodynamic, Oil Flow, Leakage.

INTRODUCTION

Lubricating oils may perform in different mechanisms according to operating conditions and type of component to be lubricated. In general, the different lubrication regimes which exist in engines are as follows:
Boundary lubrication, hydrodynamic lubrication and elastohydrodynamic lubrication [1].

Under boundary lubrication regime, friction and wear between mating surfaces in relative motion are determined by surface properties as well as by lubricant properties other than viscosity, namely, polarity and chemical behaviour (reactivity with metals) [2]. Boundary lubrication can occur in engine parts when pressures get high and velocities are relatively low or when a thick fluid film cannot be formed. This can be seen during starting and stopping (bearings, rings, cylinders), and during normal running at the ring-cylinder

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interface, at top and bottom dead centers, between highly loaded parts such as valve stems and rocker arms, crank-shaft timing gears and chains, in oil pump control gears, and in ignition distributors. However, when the temperature of the surface increases, the boundary film begins to melt and then the thermal agitation energy becomes greater than the adhesion energy due to the polarity and to molecules which become very mobile and disoriented. The film is then desorbed and cannot prevent metal-to-metal contact, so that the coefficient of friction and wear rise sharply [1].

Hydrodynamic lubrication is a system in which the shape and relative motion of sliding surfaces form an oil film having a hydrodynamic pressure sufficient enough to sustain applied loads without rupture [2].

In this case, the oil viscosity is the predominant property which affects the hydrodynamic performance. This lubrication regime is present whenever a wedge shaped film configuration can exist with high relative speed between surfaces forming the wedge, e.g. cylinder liner and rings, crankshaft bearings, conrod bearings and all sliding bearings present. Theoretically, if the engine is kept running without stop and start and assuming a continuous sufficient oil supply, the metallic surfaces will be fully separated by the oil film and frictional resistance becomes very low with almost zero mechanical wear [3]. However, operating under practical conditions, the following limitations influence hydrodynamic performance:

- When the lubricant film reaches a minimum value about the same size as the surface roughness, interaction between the peaks of asperities may be readily high.
- Abrupt loads and speeds variation affects hydrodynamic action and behaviour becomes unstable.
- Higher temperatures would eventually affect the oil viscosity by decreasing it with a consequential effect of deterioration in hydrodynamic oil film.

Hydrodynamic lubrication is especially appreciated in connecting rods and crankshaft bearings.

Elastohydrodynamic lubrication is a field of application of hydrodynamic lubrication between surfaces in line or point contacts. The contact pressures influence the elastic deformation of surfaces which in turn, governs the generated hydrodynamic pressures. Elastohydrodynamic lubrication is thus a type of lubrication in which friction and oil film thickness between two bodies in relative motion are determined by combining the elastic properties of these bodies and the viscous properties of the lubricant. These viscous properties include variation of viscosity with pressure, temperature and velocity gradient or rate of shear [4]. Elastohydrodynamic lubrication is thus a characteristic feature of lubricated line or point contacts e.g. cam shaft, rolling bearings and gears.

Higher oil viscosities ensure better hydrodynamic performance and consequently lower adhesive wear. However, cold-starting wear is very much influenced by oil viscosity. A rise in viscosity helps to maintain the film on the cylinder walls during shut-down, but makes it more difficult to reconstruct the film during starting due to the decrease in pumpability of the heavier oil. For steady running wear under adhesive conditions, the problem of pumpability is less important but a rise in oil viscosity at the cylinder film temperatures is always helpful [1].



The influence of viscosity index improvers on wear depends greatly on their polarity, if we discard problems of volatility and shear stability. At equal viscosity; polar products such as dispersant polymethacrylates, create four times less wear than pure mineral oils, but non-polar additives, such as polyisobutane fail to provide base oils with any particular antiscuffing properties [5].

The influence of detergents, dispersants and anti-oxidant additives in reducing wear in the cylinder/piston/ring group has been examined under corrosive conditions. However, with the exception of basicity reserve additives, standard antiwear additives are useful for operating conditions favourable to adhesive wear (heavy load and high temperature) in the same way as for corrosive conditions [5].

As regards the influence of lubricating oils on corrosive wear, oil additives can reduce corrosive wear in petrol engines either by neutralizing the acid products responsible (alkaline additives) or by isolating metal surfaces from contact with the corrosive agent (antiwear additives that improve resistance or polar additives).

The nature of lubricants emphasises the belief that lubricants cannot do much to prevent abrasive wear and that such wear is more or less independent of viscosity; the only relation thought to exist between oil quality and abrasion had to do with the effect of detergency on filterability and on the erosiveness of the oil carrying tiny particles in suspension. During abrasion by hard particles in the oil, the thickness of the oil film must be larger than the size of the particles to minimize their effect. Recent researches show that there is a possibility that chemical composition of the lubricant influences adhesive wear and therefore some antiwear additives can reduce abrasiveness of the wear and dust particles in the oil [1,5].

EXPERIMENTAL WORK

In the present work, wear tests were conducted on a simulated crossed cylinder test rig. By rubbing cast iron piston rings against a hard steel rotating shaft under dry and lubricated situations, comparative results have been recorded and herein displayed. Tests carried out under lubricated contacts were conducted using new oils and used oils. Different applied loads at various running speeds were adopted.

Based on the idea of crossed cylinders test rigs [4], a test rig has been designed and constructed in Machine Design Laboratory, Cairo University. Full details of construction are given in Figure 1, while photographic view of the machine is given in Figure 2.

The machine, as shown in Figure 1 is composed of three main groups: driving system, testing and loading system and measuring system.

- Driving system: The main shaft (1) is supported on two-self aligning ball bearings (2) and is attached at its end to a V-belt pulley (3). The shaft is driven via the V-belt drive (4) by means of an $\frac{1}{4}$ HP electric motor (5). Both the motor drive and the shaft bearings assembly are mounted on a steel frame (6) fabricated by welding of structural steel angles.
- Testing and loading system: The specimen to be tested (either bearing bush or piston rings) (7) is clamped between steel plates (8) and fixed to leaf springs (9). The leaf springs (9) are fixed to the loading arm (10) such that the load can be normally applied at the contact zone and in the mean time the friction resistance can be measured. The loading (10) is mounted on a back shaft (11) by means of ball bearings (12). A balancing weight (13) is used to ensure zero loading before starting loads. The load

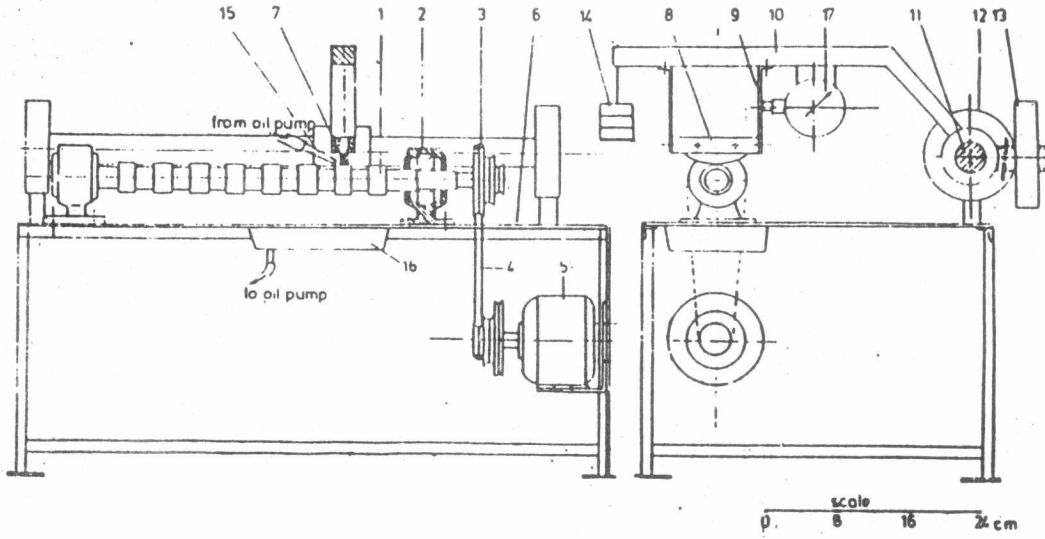


Figure (1) Schematic layout of Test Rig.

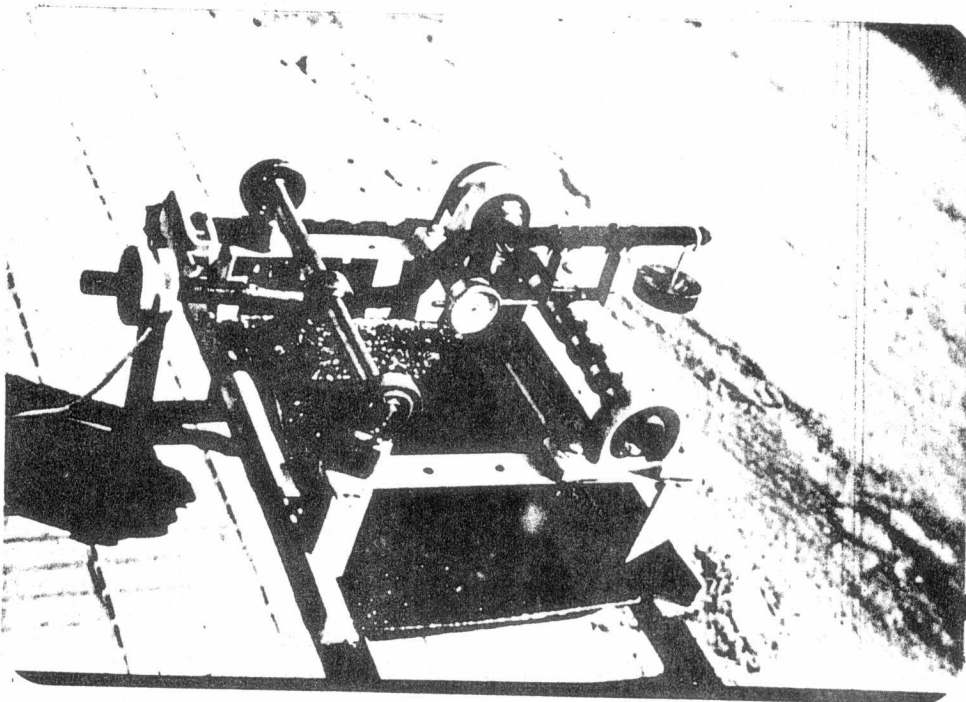


Figure (2) Photographic View of Test Rig.

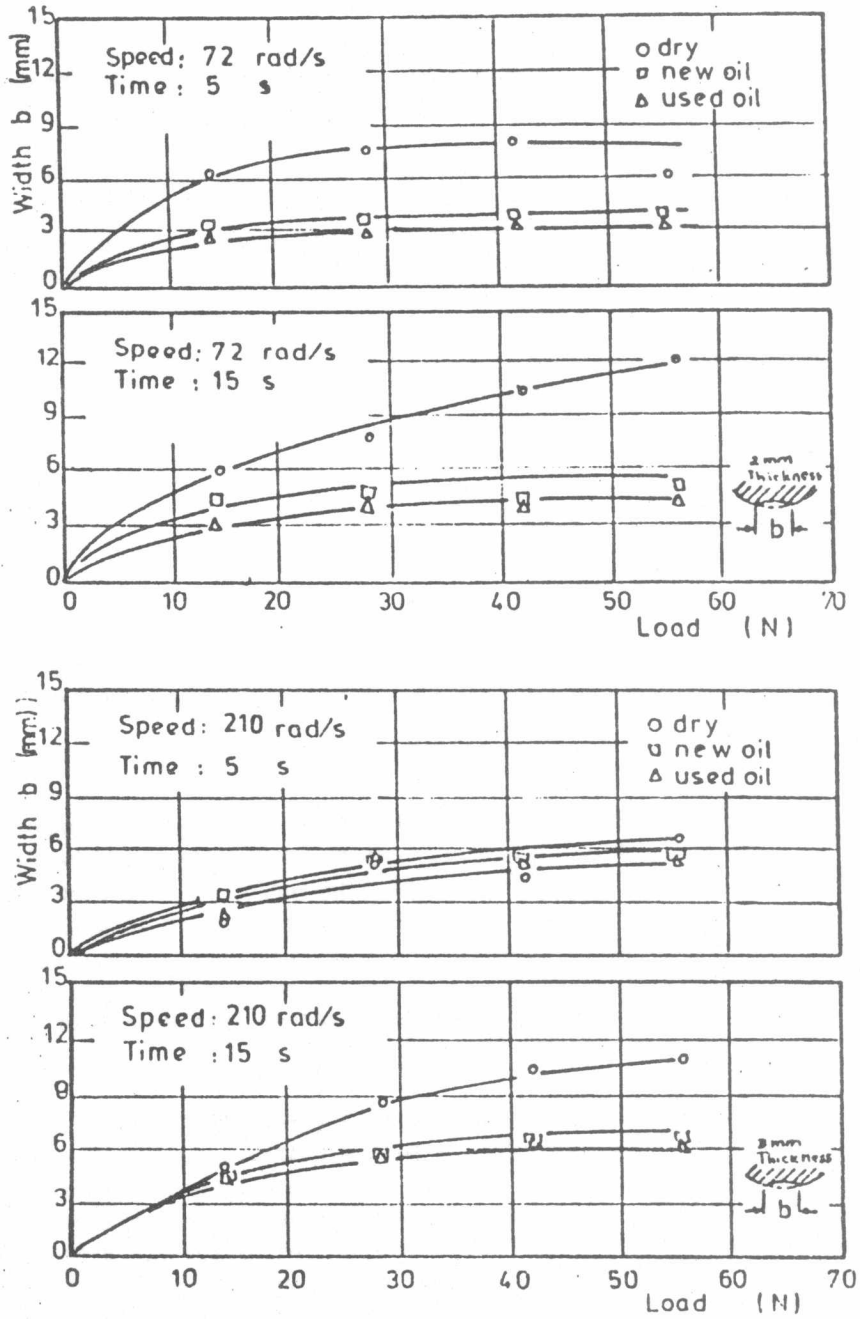


Figure (3) Results of Wear Tests.



is applied by dead weights (14) hanged at the free end of the shaft. By this arrangement, the load can be magnified by the average ratio present.

Oil is introduced into the contact zone by means of the nozzle (15) and sufficient flow of the circulating oil is maintained using gear pump. Drained oil is collected in the reservoir (16) and is recirculated throughout the test period.

- Measuring system: Shaft speed is measured directly using tachometer, while the applied load is computed from the leverage mechanism and dead weights present. The frictional resistance is measured by means of determining the leaf springs stiffness and reading the dial gauge (17) scale. Wear measurements have been simplified to the measure of possible scratch width with load and speed for specific test duration.

The present tests were aimed towards the investigation of the wear in piston rings. So cast iron piston rings of 3.4 x 2 mm cross section and hardness 470 BH were tested. The main shaft has a hardness of 50 Rockwell C.

MISR SUPER 7500 oil was used throughout the test.

EXPERIMENTAL RESULTS:

Sets of results are presented for each running speed in Figure 3. Each set of results display comparative wear results under dry contact and lubricated contacts in new and used oils. The sets of results are given for test duration ranging from 5 and 15 seconds.

Tests were run under applied loads up to 70N at a speed of 210 rad/s.

Referring to attained results, figure 3, it can be seen that the general expected feature of wear variation with load, speed, duration and contact conditions could be fairly concluded from curves. Any increase in the applied load and/or test duration resulted a corresponding increase in wear rate as recognized by the increase in scratch width and depth increase. The use of lubricated contact resulted lower wear, approximately less than half the wear attainable in case of dry contacts. Careful examination of attained results shows that the use of new oil as a lubricant rendered less wear than using used oils.

This type of test is a very simple and helpful means in comparing different contact wear behaviour. Under dry contacts, the wear is mainly attributed to both adhesion and abrasion mechanisms. According to Archard formula [6], the wear under adhesion conditions is proportional to the applied load and inversely proportional to the hardness. The same concept also applies under abrasion wear. So, in the present situation with both adhesion and abrasion are expected to function simultaneously, it would be expected that the wear rate increases with the increase in the applied load as shown in the given results.

With the increase in either the test duration or the running speed, it would be expected that the increase in the travel distance affects the wear by increasing its worn volume.

The role of a lubricant in any sliding contact is three folds: to limit metallic interaction, to form a lubricant oil film and hence decrease real area of contact and to cool the contacting zone. Hence, under any lubricated contact situation the wear rate should be reduced. However, the percentage reduction of wear rate can be taken as a measure of the lubricant quality: lower wear is surely a character of a sliding pair lubricated by good quality oil and vice versa.



Although the contact situation in the present test (line contact) is not representative of cylinder piston ring interaction, the results can be considered of some importance to cams lubrication. However, accepting that these tests are of comparative nature, the attained results would throw some light on the effect of oil quality on wear rate.

The use of used oils, as given in Figure 3, indicated that under all test conditions, the wear increases whenever used oils are tested compared to new oils. This conclusion can be withdrawn from attained results and can be attributed to both the deterioration of oil lubricating properties (viscosity) and the presence of abrasive wear debris.

Used oils are always characterized by lower viscosities coupled with the presence of wear and dust particles. With lower viscosities, the lubricant may not be able to generate sufficient hydrodynamic pressures to sustain the applied loads and hence metal contact may be evident. This type of lubrication failure is quite liable in line contact situations where high applied pressures are present e.g. cam shaft. In this context, it is worth mentioning that the lubricant properties affect not only crank bearings and piston/cylinder contacts but also cams and followers.

The presence of contaminated wear debris and dust particles carried with the oil are surely of bad influence on the surfaces wear. With the oil flowing under pressure in the crank bearings, the kinetic energy of contaminated oil may erode bearings and in the long run affect the crank surface. Moreover, the wear debris and dust particles may exhibit high hardness values and work as abrasives which abrade piston/cylinder surfaces, crank and bearings surfaces and cam/follower surfaces. Hence careful examination of the oil, from the wear point should be assigned to recommend its use.

So it is expected that the present test, can be taken as a demarkation line to identify the limits of continuing using oils.

EFFECT OF WEAR ON THE HYDRODYNAMIC PERFORMANCE

Although oils properties have to be carefully controlled to maintain satisfactory lubrication behaviour [2,5], the dimensional changes in cylinder, piston, and bearings due to either excessive wear or overheating would eventually contribute largely in dictating the final behaviour characteristics.

Improper lubricant selection or the use of lubricant for prolonged time is one factor behind the bearings overheating. On the other hand, the wear of bearings would lead to greater radial clearances which, in turn, would affect the load carrying capacity of the bearing, the oil flow rate and the frictional resistance.

In this part, it has been decided to handle the effect of wear on the bearings hydrodynamic performance. Wear means a change in dimensions. This can be understood as a change in bearings radial clearance and possibly geometrical shape. So, in the present analysis, the effect of changes in radial clearance on the bearing performance characteristics, namely, load capacity, flow rate, friction loss and minimum film thickness, will be fully analysed and discussed. Comparative results are also presented using different types of oils in new or used conditions.


ANALYSIS:

The effect of dimensional changes in lubricated contacts can be accounted for by recalling the basic Reynold's Equation for hydrodynamic lubrication. The attained numerical results for finite bearings will be taken as basis for the analysis to come [7,8].

In the study of lubrication regimes, it is a common practice to group some of the basic parameters in nondimensional normalized forms. Although these normalized parameters have some significance in analysing bearings lubrication, actual (dimensioned) values, as well as non-dimensional parameters variation with change in bearing clearance will be herein used.

The nondimensional terms \bar{W} , \bar{Q} , μ are actually functions of bearing geometrical configurations, the eccentricity, the bearing length to diameter ratio L/D and the radial clearance ratio C/R which is actually a variable dependent on rate of wear.

The parameters L , D , N , and ϵ are input data.

APPLICATION:

The present analysis is applied to the bearings of a 4-cylinder petrol engine with specifications as given in table (1)

Table (1): 4-cylinder European automobile Engine [1]

No. of cylinders	: 4 (1160 cm ³)
power	: 555 BHP
speed at full power	: 6000 rpm
speed at idling	: 1500 rpm
nominal bearing diameter	: 48 (mm)
radial clearance	: .035 to .070 (max. 0.100) mm
length to diameter ratio	: 0.61

for exact dimensions of any bearing, the presented results will hold qualitatively true, while the scale of parameters variation should be corrected for exact quantitative results.

The oils used in the analysis have been taken from the local market

The performance characteristics, in dimensional form, of the engine bearings are plotted in Figures (4 and 5) as computed from the previous data. Figures give the variation of the actual load carrying capacity W with the minimum film thickness ratio h_{min} and eccentricity ϵ with the running speed and clearance ration C/R as parameters. The performance of the Misr Super 7500, oil has been considered in the analysis and herein presented.

In Figure (4a), the Misr Super 7500 oil has shown to be slightly affected by use through the first half of its recommended life. Excessive wear, as would be presented by large radial clearance, would lead to drastic drop in load capacity under full power or idling speed using either new or used oil.

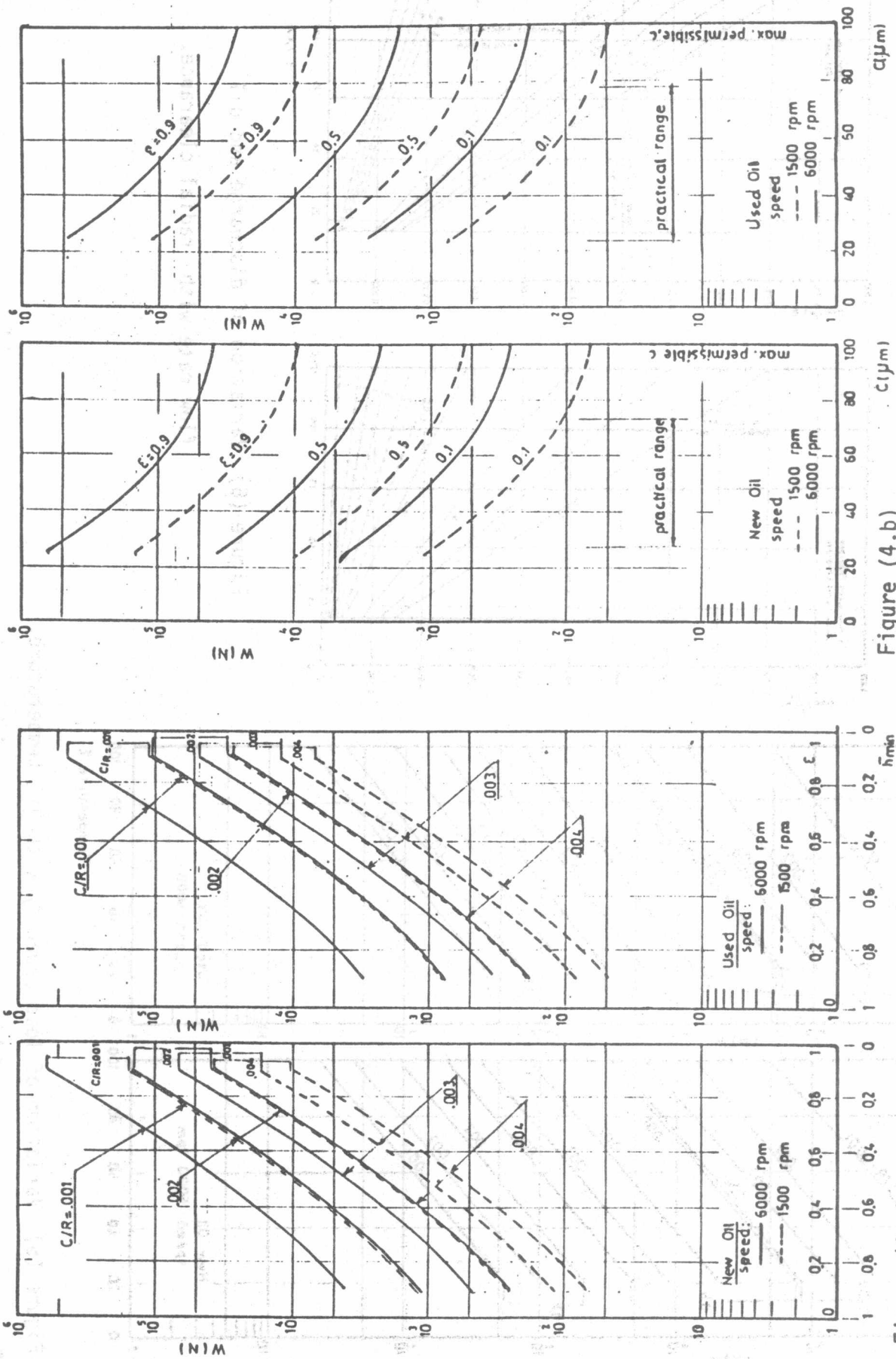


Figure (4.b)

Figure (4.a)

Figure (4) Variation of load capacity with bearing parameters : a-minimum film thickness and eccentricity, b-radial clearance.

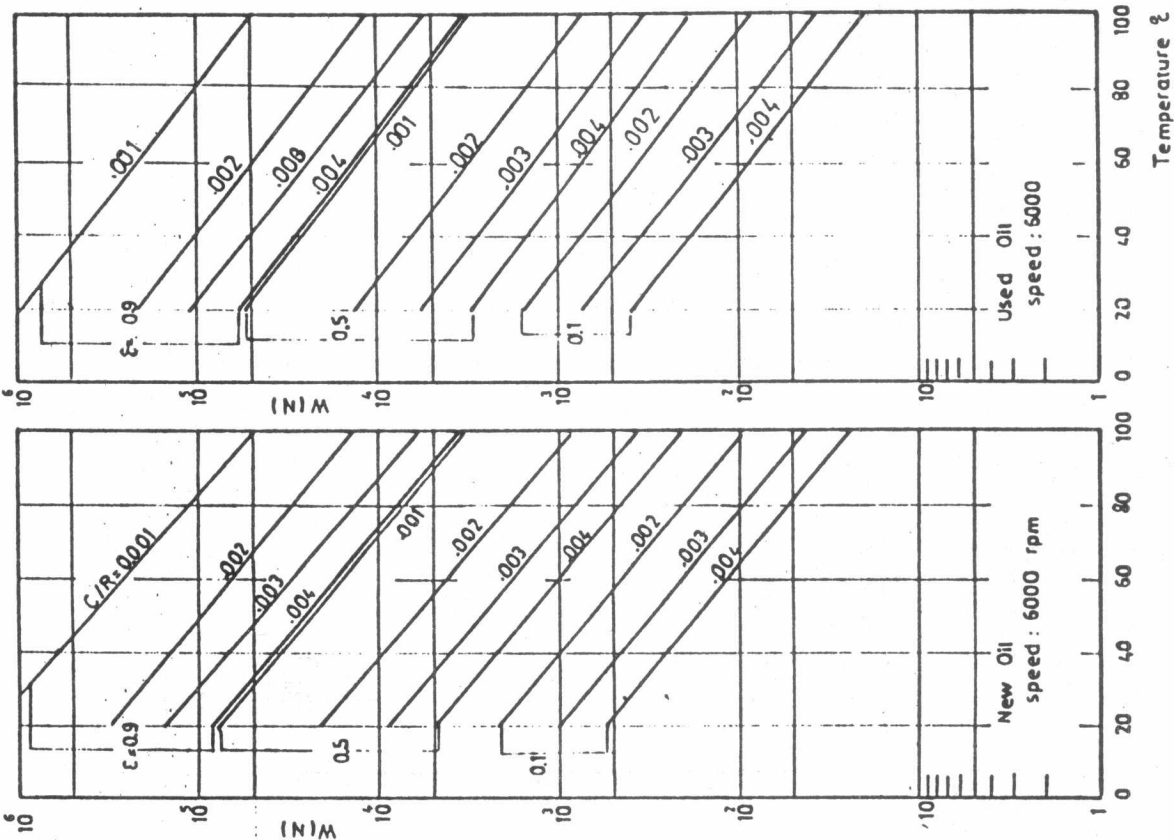


Figure (5) Variation of Load Capacity with Oil temperature.

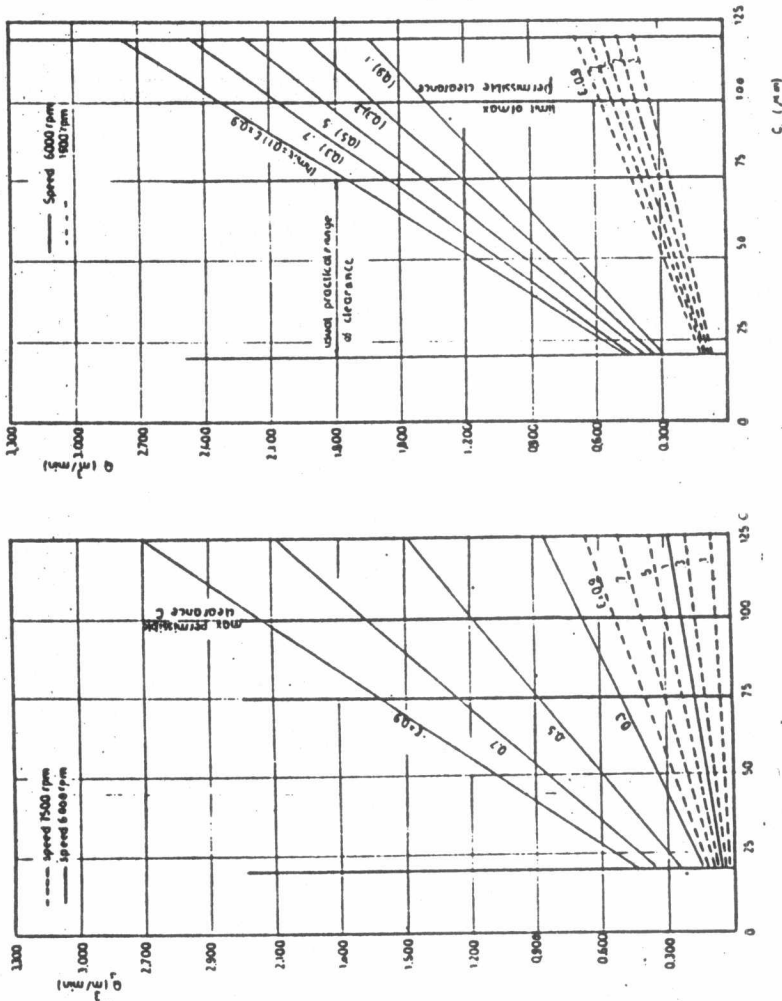


Figure (6) Variation of discharge and oil flow rate with radial clearance.

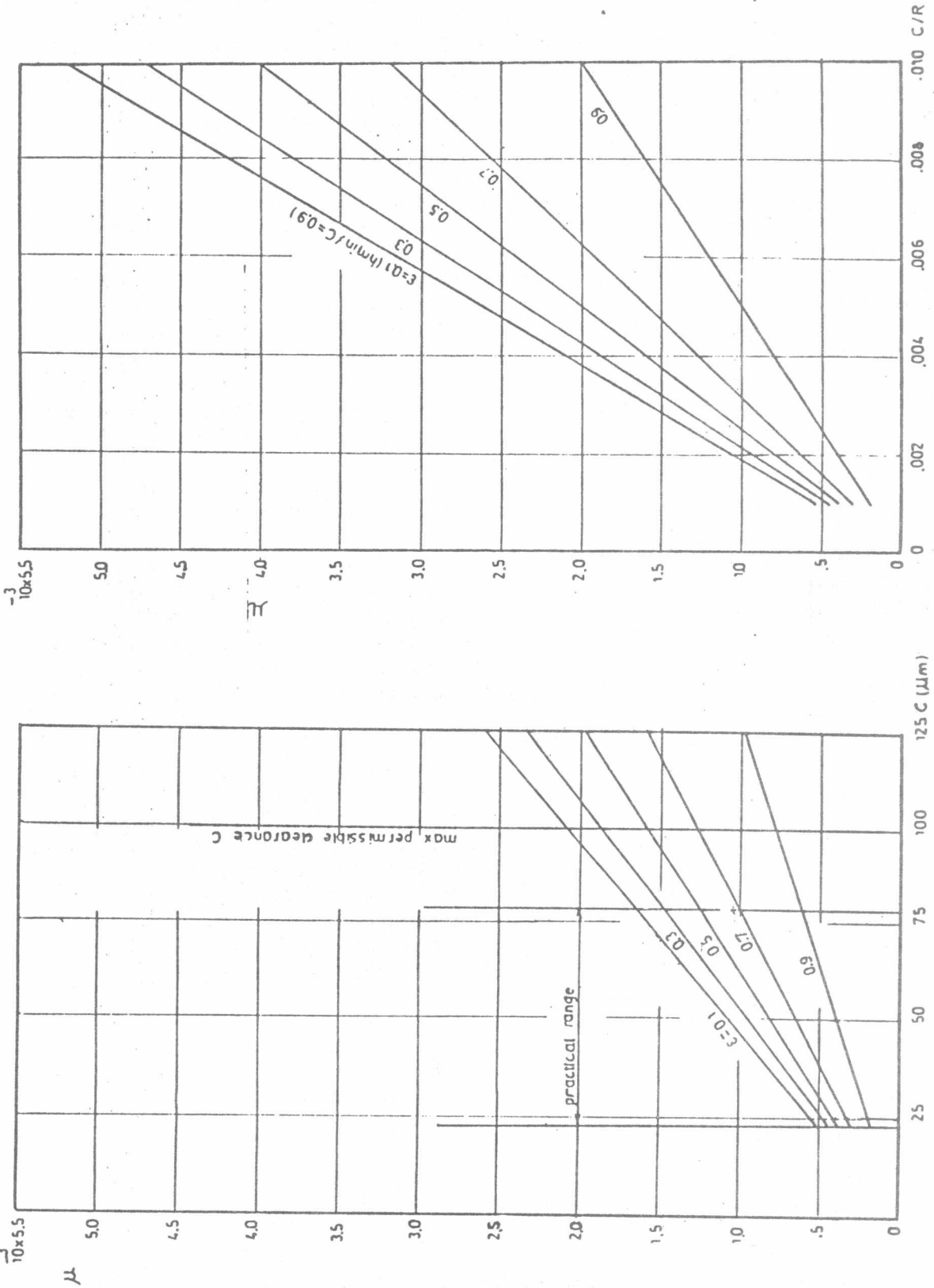


Figure (7) Variation of coefficient of friction with radial clearance.



Of interest to mention that to maintain high load capacity comparable with the actual engine loads, the engine wear should be controlled and limited so that the bearing clearances would not go too high, say beyond a value of $C/R = 0.004$. At relatively high clearances, the load capacity is greatly reduced and may not be sufficient to withstand engine loads. In this context, a limit on engine wear should be put forward to guarantee its rated life and proper performance. As long as one of the basic wear contributions is the lubricant properties and condition, the wear rate limit can be indentified in accordance with the lubricant wear characteristics as influenced by the oil properties and contaminates during continuous use.

Figure (4b) gives the values of the load capacity as a direct function of the radial clearance. In general recommendations, the oil film thickness should not go below 0.0025 to 0.0042 mm for mains bearings and 0.002 to 0.004 mm for big ends bearings. However, for the assigned engine bearings (Table 1) the range of radial clearance is taken practically to be 0.025 to 0.070 mm and it is further recommended that the clearance should not exceed 0.100 mm. These units have been assigned such that the engine runs safe without a probability of oil film failure. Add to this, the choice of the values of clearance are eventually based on the concept of maximum load carrying capacity and minimum power loss with minimum side leakage. The present graphs, Figures (4a) and (4b) confirm that excessive increase in radial clearance due to wear is not recommended as it would deteriorate the hydrodynamic performance.

Due to the expected large change in oils viscosity with temperature increase, Figures (5a and 5b) give the variation of the load capacity with oil temperature rise.

Figures (6 and 7) reveal that the radial clearance between bearing surfaces has a direct impact on oil flow rate, side leakage and frictional resistance. The increase in radial clearance due to possible engine wear would require higher rates of oil delivered to bearings to ensure sufficient oil film generation and to take account of the side leakage. The coefficient of friction displays an increasing behaviour with the increase in radial clearance. This situation affects the bearing performance by increasing the power loss which renders higher temperature rise.

In conclusion, it can be seen that restrict limitation on engine wear rate should be put to assure longer engine life. The proper assignment of lubricant properties and life would, not only, safe guard the engine against excessive wear but would also guarantee high performance. The interaction between lubricant behaviour and engine wear can be summarized as follows: excessive engine wear due to improper oil properties would lead to greater bearings clearance. This increase in bearings clearance leads to reduced load capacity, higher friction and greater oil flow rates. These consequences affect the bearing by reducing the minimum oil film thickness, increasing the oil temperature rise with a consequential reduction in oil viscosity, higher rates of oil side leakage. An ultimate situation, is a bearing working inefficiently with a probability of failure.

NOMENCLATURE

- C : radial bearing clearance (a function of bearing wear rate)
 D : bearing nominal diameter ($=2 R$; R:radius)
 e : eccentricity
 F : friction force
 L : bearing length
 hmin : minimum film thickness relative to bearing clearance where, $(Tr_{min} = hmin/c) = 1-$)



- N** : journal rotational speed
Q : flow rate
Q : side leakage
W : bearing load capacity (applied load)
 μ : coefficient of friction
 η : dynamic viscosity

Non-dimensional terms

- C/R** : radial clearance ratio
L/D : length to diameter ratio
 \bar{Q} : flow rate factor ($= Q/R^2NL$)
 \bar{W} : load parameter ($= (W/LD)/\eta N$)
 ϵ : eccentricity ratio ($= e/c$)

REFERENCES

1. Braithwaite, E.R., "Lubrication and Lubricants". Elsevier Pub. Co., Amsterdam, 1967
2. Schilling, A., "Automobile Engine Lubrication", Scientific Publications (G.B.) Ltd., 1972.
3. Mokhtar, M.O.A., Howarth, R.B. and Davies, P.B., "Wear Characterisitcs of Plain Hydrodynamic Journal Bearings During Repeated Starting and Stopping", ASLE Trans., Vol. 20, 1977, p. 191
4. Dowson, D. and Higginson, G.R., "Elasto-hydrodynamic Lubrication", Pergamon (Oxford), 1966
5. Lubrizol Technical Presentation, "Lubricating Oil and Additives"; The Lubrizol Corporation, Report C-7528, April 1983.
6. Archard, J.F., "Elastic Deformation and Laws of Friction", Proc. Roy. Soc., London, Vol. 243A, 1957, p. 190
7. Pinkus, O. and Sternlicht, B., "Theory of Hydrodynamic Lubrication", McGrawHill, New York, 1961
8. Raimondi, A.A. and Boyd, J., "A Solution for the Infinite Bearings and Its Application to Analysis and Design".
 - Part I : Trans. ASME, Vol. 80, 1958, p. 159
 - Part II : Trans. ASME, Vol. 80, 1958, p. 175
 - Part III : Trans. ASME, Vol. 80, 1958, p. 194

