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INFLUENCE OF OIL TYPE ON THE PERFORMANCE CHARACTERISTICS OF A TWO AXIAL GROOVE JOURNAL BEARING

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ABSTRACT

The use of Environmentally Adapted Lubricants (EALs) is a subject of growing interest to industry as legislation increasingly demands the replacement of mineral oil lubricants. Vegetable based fluids are widely seen as providing lubricants from a renewable source as well as meeting demands for improved biodegradability. However, at present, utilisation of such fluids is limited due to their rapid oxidation. EALs produced from other base stocks (i.e. synthetic esters) have been shown to provide performance benefits in hydrodynamic thrust bearings. In the present study, a hydrodynamic journal bearing test rig has been employed to compare the performance of three EALs (a VG32 saturated ester, rapeseed base fluid and a Propylene Glycol Dioleate) relative to three mineral turbine oils (ISOVG32, ISOVG46 and ISOVG68) in the hydrodynamic regime. Results are given in terms of temperature, power loss and minimum film thickness. The impact of oil VI is also discussed.

1 INTRODUCTION

The use of Environmentally Adapted Lubricants for industrial applications is a major area of current interest given increasing environmental awareness and the potential for performance enhancement, as demonstrated in previous studies, for example [1,2].

Both thrust and journal bearings are major mechanical components in numerous applications including hydroelectric turbines. In order to maximise the efficiency of these devices, the choice of a suitable lubricant is essential. A review of the literature reveals a shortage of experimental data regarding the performance of synthetic and vegetable base fluids (e.g. rapeseed) in hydrodynamic lubrication and how this performance compares with that of mineral oils.

A summary of published literature regarding experimental investigations performed with fixed geometry hydrodynamic journal bearings is provided in [3].

In [4], a series of lubricating fluids (mineral, synthetic and VI enhanced oils) were tested in a journal bearing test rig. It was found that bulk fluid properties have the

greatest impact on oil-film temperatures while temperature-viscosity properties were seen to be of significance. When comparing synthetic and mineral fluids with the same initial viscosity grade, despite higher temperatures at the inlet side (due to shear in the “wedge” of reducing oil film thickness), thicker films were seen for the synthetic lubricants due to their higher “operating” viscosity in the region of minimum film thickness. A hypothesis is promoted whereby high VI permits the use of a synthetic lubricant with lower base viscosity grade in comparison with a mineral oil, reducing heat originating at the inlet side of the bearing but maintaining oil film thickness.

Experiments referred to in [5] showed that bearings lubricated with higher VI fluids demonstrated slightly lower bearing surface temperatures than those seen for lubricants with lower VI. This, combined with a 10% lower power loss for the higher VI lubricant, was observed despite little or no difference being seen in minimum oil film thickness.

Temperature-viscosity properties were also examined for a two axial groove bearing in [6]. Viscosity was found to have a pronounced effect on bearing operating temperature. Comparison of temperatures along the bearing centreline with theory show best correlation for a low viscosity fluid. Thermal distortion was also seen to alter bearing geometry. Results from these tests were confirmed by experiments with different oils in a two axial groove journal bearing [7].

Work carried out for the present study is aimed at extending available knowledge in respect to the use of EALs in hydrodynamic lubrication, primarily for hydropower applications. This is achieved through examining the operation and performance of a fluid-film journal bearing with a range of mineral and synthetic fluids. This study also provides an opportunity to examine the influence of viscosity index (VI) on bearing performance given that synthetic and vegetable oils have, in general, considerably higher VI values than mineral oils.

2 MATERIALS AND EQUIPMENT

2.1 Hydrodynamic Journal Bearing Test Rig

Tests referred to in this paper are performed in a test-rig specially developed for the purpose of investigating fluid-film journal bearing performance. A schematic diagram of the equipment is given in Fig. 1.

The test bearing (1) is mounted in a spherically formed, self-aligning unit (2) which is in turn mounted within a larger housing (3). Oil from a 40 litre tank is supplied and drained through an inlet and outlet on the underside of the main housing. Requisite load is applied by means of a vertically mounted hydraulic cylinder (4) located immediately above the bearing housing. This is attached via pins to the support frame and a yoke around the main housing. An accumulator is utilised in order to maintain constant loading, monitored by means of a load cell (5). The shaft is driven by a 3-phase electric

motor (6) coupled to a frequency converter to permit speed variation. Oil flow rate is monitored by a gear wheel type flow-meter.

The machine has a so-called “no seal” configuration. This means that there are no shaft seals installed to close the gap between the bearing housing and shaft. This has the advantage of removing a possible source of friction losses and also means that the bearing is never submerged in a “bath” of oil.

Figure 1. Schematic diagram of test rig showing principal components.

2.2 Test Bearing and Instrumentation

A babbitt-faced two axial groove bearing (split in two halves) is used in this study. A shaft diameter of 179.72 mm gives a bearing radial clearance (cold) of $154 \pm 14 \mu\text{m}$. The bearing is 129 mm wide giving a projected area of 23184 mm^2 . The bearing is a standard product and as such is representative of commercially available equipment.

Figure 2. Location of sensors installed in the bearing. (View shows inside surfaces of upper and lower bearing halves). The figures beneath the diagram indicate the relative angular position on the bearing circumference.

The journal bearing has been equipped with a number of sensors to provide as comprehensive a set of data as possible for performance characteristics. Fig. 2 shows the location of the sensors mounted in the bearing. In the lower bearing half, temperature sensors are mounted at regular intervals (i.e. at 15, 30, 45, 75, 90, 105, 120, 135 and 165 degrees from the horizontal split, inlet side). Two additional sensors are located 5mm in from either side of the bearing at the 135° position. Sensors are also mounted in the oil supply and drainage and water cooling lines to monitor inlet / outlet temperatures. Additional instrumentation is provided by the frequency converter unit. Details of measurement uncertainty for the various sensors are given in Table 1.

Table 1. Measurement uncertainty

Thermocouple (type K)	$\pm 1.0\text{ }^{\circ}\text{C}$	Rotational speed	$\pm 0.4\text{ }\%$
Friction torque	$\pm 0.2\text{ }\%$	Flow rate	$\pm 3.0\text{ }\%$
Bearing load	$\pm 0.05\text{ }\%$	Shaft displacement (verified by calibration)	$\pm 5\text{ }\mu\text{m}$

Sensors for measuring oil-film temperature are carefully mounted so that oil “tapped” from the film comes into direct contact with the sensor.

Shaft displacement (inductive) sensors have a tip diameter of 8mm and standard M10 thread. Sensors are mounted at the 240° and 330° positions respectively relative to the inlet side at the horizontal split (i.e. with a 90° angle between them). The sensor tip is set at a slight radial displacement (no greater than 0.4 mm) below the bearing inner surface so an output voltage reading will still be observed should the bearing come into direct contact with the steel shaft. This also eliminates any risk of damage to the sensor.

Data from the various sensors and other monitoring equipment is logged by means of a PC based modular data acquisition unit. This is identical to that described in [8].

2.3 Lubricants

Table 2. Lubricant characteristics

	Density 15°C (kg/m ³)	Viscosity 40°C (mPas)	Viscosity 100°C (mPas)	Viscosity Index
VG32 mineral	847	27	4.5	113
VG46 mineral	857	37	5.6	114
VG68 mineral	860	56	7.2	106
Synthetic ester	934	29	5.3	152
Rapeseed oil	919	32	7.0	213
PGD	904	19	5.0	209

The mineral oils used in these experiments are all standard, commercially available, fully-formulated turbine oils. These are utilised as a reference set for comparison with the EALs. Lubricant properties are shown in Table 2.

The EALs have considerably higher viscosity index (VI) values than the mineral oils. The value for the synthetic ester is 50% higher than that for the mineral oils while the rapeseed and PGD lubricants have values twice as large as the mineral oils.

3 EXPERIMENTAL PROCEDURE

All experiments are carried out in the hydrodynamic journal bearing test rig described previously. The same method is employed for all test lubricants, allowing for direct comparisons to be made.

In this study, speeds of 1000, 1500 and 2000 rpm together with loadings of 1, 1.5 and 2 MPa are employed. These settings are determined by the operational limits of the equipment and have also been used in order to provide a set of data permitting future comparison with results obtained from tests carried out with other lubricants and with other test apparatus. In all tests carried out during this study, oil is supplied at constant temperature and flow rate i.e. $40^{\circ}\text{C} \pm 0.5^{\circ}\text{C}$ and $2 \text{ l/min} \pm 0.05 \text{ l/min}$.

In the initial “warm-up” stage, the (unloaded) shaft is first brought up to speed in order to form an oil film. Loading is then applied by means of an hydraulic cylinder whereby the bearing housing is hoisted such that a specified pressure, related to the projected shaft area, is achieved between the bearing and the shaft. The machine is then left to run until the desired oil inlet temperature (in this case $40^{\circ}\text{C} \pm 0.5^{\circ}\text{C}$) is attained at the required load-speed setting.

Once this stage is complete, the machine continues to be run but with careful control of the oil inlet temperature, achieved using a water-cooled heat exchanger. With the inlet temperature held constant, the machine is now run until thermal equilibrium is reached.

Once thermal equilibrium is attained, i.e. “steady-state conditions” when variation in oil-film temperatures does not exceed $\pm 0.2^{\circ}\text{C}$ during a period of 20 minutes, the various variables are measured. Further tests can be carried out by increasing load and/or speed, following the procedure as given above. Due to the bulk of the test rig and the consequent thermal properties, it is important to ensure that the housing has heated through completely before taking any final measurements.

4 RESULTS AND DISCUSSION

Figs. 3 and 4 show temperature distribution along the bearing circumferential midline for a shaft rotational speed of 2000 rpm and loadings of 1 and 2 MPa respectively. Fig. 5 shows the corresponding temperature distribution for settings of 1000 rpm and 2 MPa.

As expected, when comparing values for all the oils, the highest temperatures are seen for the VG68 mineral oil. In general, temperature profiles along the bearing centre line are very similar for all the test fluids at all speed and load combinations. For the two cases illustrated at the highest speed of 2000 rpm (Figs. 3 and 4), the temperature plots appear to move through a simple vertical shift (i.e. an identical temperature difference is seen between results for any two fluids at all measurement points). However, at the lower speed setting (Fig. 5) values seen for the VG68 mineral oil at the inlet and outlet positions are very close to those for other fluids while a larger temperature difference is seen in the region of thinnest film thickness.

Figure 3. Midline temperatures at 2000 rpm, 1 MPa

Figure 4. Midline temperatures at 2000 rpm, 2 MPa

Figure 5. Midline temperatures at 1000 rpm, 2 MPa

Examining the temperatures found for the synthetic ester shows values in all three test cases approximately halfway between those for the VG32 and VG46 mineral oils. At the highest speed / load setting, the readings are slightly below the halfway values. This is due purely to the difference in fluid viscosities.

Temperature values for the synthetic rapeseed base fluid along the circumferential midline of the bearing are only slightly lower than those for the VG46 mineral oil (see Fig. 3). However, if the speed is decreased, values for the two oils are seen to be almost exactly the same (see Fig. 5). In this case, temperatures at the bearing inlet for these two oils are almost the same as for the VG68 mineral. In comparison with the VG32 mineral however, peak values for the rapeseed base fluid are approximately 4-5°C warmer in all cases, a result of the higher viscosity of the latter.

Looking at the results for the PGD fluid, these are by far the lowest values seen for any of the test oils, as would be expected from the low viscosity grade. Comparing Figs. 3 and 4, it is apparent that the change in loading does not have as substantial an effect on the PGD when compared to the other test fluids.

Significant temperature fade at the outlet side of the bearing is caused by the presence of the taper. The fact that the final thermocouple is located in the cavitation region and is also mounted at the edge of the downstream oil supply groove means that “back-flow” of the colder oil reaches the thermocouple i.e. hot oil in the cavitation zone mixes with cold oil entering the bearing from the groove.

Figure 6 illustrates the actual viscosity of the fluids according to the temperatures measured at the various points along the bearing centreline at 2000 rpm and 2 MPa. This permits us to compare fluid viscosities under identical operating conditions.

Figure 6. Fluid viscosity at each point along the bearing centreline at 2000 rpm, 2 MPa

From the plot it can be seen that in the region of highest temperature (thinnest film, lowest viscosity) the values found for the VG46 mineral and synthetic ester fluids are essentially identical. However, in the inlet region, represented by measurements at 15°, 30° and 45°, viscosities for the synthetic ester are lower than that for the VG46 mineral oil. This is beneficial in reducing losses within the bearing while maintaining acceptable minimum film thickness. Thicker viscosities for the EALs in comparison with VG32 mineral oil suggest better film-building ability at the measured temperatures.

Flow rate may have an impact on temperatures within the oil film. A reduced flow rate of 1.6 l/min was used for a number of additional tests. This showed a marginal difference in measured temperature values whereby the bearing ran slightly warmer. The higher rate of flow was retained as it was found to be easier to control oil inlet temperature. Conversely, due to the limitations of the oil pump, it was not possible to run the thinner oils at flow rates higher than 2 l/min.

Power losses in the system as a whole (the two support bearings as well as the test bearing) are measured via the frequency converter unit. This provides a means of comparing relative losses due to friction for each of the test fluids. Fig. 7 shows the results obtained during tests with an applied load of 2 MPa at 1000, 1500 and 2000 rpm.

It is clear that losses are affected by the VI of the fluid. For all three EALs, losses increase with speed at a greater rate than for the mineral oils. However, this must be kept within perspective – the difference between corresponding values is not considerable. Indeed, at the lowest speed there is no real difference. Again, this follows the predicted behaviour as dictated by the high VI value. The VG68 mineral oil continues to show highest losses for all test cases.

Figure 7. Power losses in the system at 2 MPa load

An indication of minimum oil film thickness measured in the bearing, i.e. the minimum clearance between the shaft and bearing at given load and speed, is shown in Figs. 8 and 9. In this case, a correction has been made for thermal expansion of the shaft using the recorded temperature data. A temperature value obtained from half the difference between the maximum and minimum values measured in the bearing and a reference value (taken as the oil inlet temperature, 40°C) is used to calculate variation of the radial gap between the shaft and bearing. This is then used to correct readings taken from the two inductive sensors installed in the test rig. This correction is applied with reference to data for bearing and shaft temperatures given in [6,9]. A favourable comparison is found between results obtained here for the VG32 mineral oil and those in [10] for the same type of fluid.

Figure 8. Minimum oil-film thickness at 2000 rpm and loadings 1, 1.5 and 2 MPa

Figure 9. Minimum oil-film thickness at 2 MPa and speeds 1000, 1500 and 2000 rpm.

Results given here cannot be taken as absolute values since certain factors, such as thermal expansion and pressure effects on the facing material, have not been determined exactly. However, these plots provide a means of comparing results obtained for the bearing with the different lubricating fluids at the given operational settings.

The correction for thermal expansion does not incorporate any allowance for bearing / housing expansion as this is not deemed necessary. Due to the sheer bulk of the bearing housing (width 450 mm, height 745 mm, mass 400 kg), combined with the fact that the incoming oil circulates around the housing before entering the bearing, thereby providing effective cooling, it is not thought that the housing warms up to any notable extent. A constant temperature of $< 40\text{ }^{\circ}\text{C}$ is maintained in the housing material. On the other hand, shaft temperature can reach higher values, hence its inclusion in the temperature correction. This has the effect that the radial clearance effectively reduces slightly as temperature increases. Two views of the bearing housing are given in Fig. 10 illustrating the non-symmetrical form and large dimensions.

Figure 10 a) and b). Bearing housing. The visible shaft section has diameter 268 mm.

From Figs. 8 and 9 it can be seen, as expected, that the thickest oil (VG68 mineral) gives the thickest oil film in each load / speed case. For all the test fluids, an increase in speed or decrease in load results in increased minimum film thickness. It is also clear that VI plays an important role as load and speed are increased.

Comparing the different oils, measurements for the synthetic ester are in line with those seen for the VG46 mineral with slightly thicker values for the ester at higher load / speed. The rapeseed fluid also demonstrates similar readings at lower load / speed settings. The higher VI of the fluid is demonstrated by the thicker film seen at high load and speed. This is also seen for the PGD in comparison with VG32 mineral.

A viscosity-temperature diagram for the test fluids is shown in Fig. 11. From this it can be seen that the higher VI of the EALs gives similar viscosity at elevated temperatures compared to corresponding mineral oils. The value for the synthetic ester eventually converges with that for VG46 mineral at approx. 115 °C. However, in the temperature range of specific interest (40 – 100 °C), it can be seen that viscosity of the synthetic ester lies between those of the VG32 and VG46 mineral oils.

Figure 11. Plot of Dynamic Viscosity v. Temperature for the four test fluids

At the same time, referring back to Figs. 3, 4 and 5, maximum temperatures for the synthetic ester and rapeseed fluids are seen to be marginally lower than those for the VG46 mineral oil. These results are due to a combination of the higher VI and heat capacities of the EALs. Heat capacity data available for the mineral oil shows a value of approximately $1.74 \times 10^6 \text{ JK}^{-1}\text{m}^{-3}$ while a value of $1.84 \times 10^6 \text{ JK}^{-1}\text{m}^{-3}$ is found for rapeseed oil [11]. The corresponding value for the synthetic ester is $1.85 \times 10^6 \text{ JK}^{-1}\text{m}^{-3}$.

The results obtained allow us to conclude that high VI fluids can adequately match the performance of mineral oils with higher viscosity grade. In this respect the three EALs tested perform well in comparison with the mineral oils.

In commenting on the rapeseed base fluid, it should be noted that utilisation of such vegetable base fluids is restricted, for example to lower temperature applications, due to factors relating to their chemistry such as oxidation and hydrolysis. In tests to measure oxidative stability, e.g. Baader test DIN 51554 part 3, rapeseed oils display considerably shorter life-times when compared to those found for synthetic esters [12].

5 CONCLUSIONS

It is clear from the results obtained that the EALs function satisfactorily in comparison with the mineral oils tested in this particular bearing arrangement. The synthetic ester fluid exhibits characteristics very close to those found for VG46 mineral oil. At the higher loads and speeds in particular it demonstrates lower temperatures, a more favourable viscosity distribution along the film and comparable minimum oil film thickness values to the mineral oil.

The PGD fluid has been shown to perform well at the higher loads and speeds in terms of minimum film thickness and power losses when compared with VG32 mineral oil. This is equally true for the rapeseed base fluid when compared with VG68 mineral oil.

Viscosity Index is seen to play an important role, together with heat capacity, in producing a thicker oil film. This suggests that it may be possible to replace lubricants in certain applications with thinner, high VI fluids.

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