PERFORMANCE OF A GEAR PUMP BEARING TAKING INTO ACCOUNT ELASTIC DEFLECTION OF BOTH HOUSING AND SHAFT

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SUMMARY
Plain bearings play a significant role in the operation of hydraulic fluid pumps, in particular in those with high discharge pressures. Experience shows that the selection of inappropriate bearing designs can lead to substantial wear and even pre-mature damage of the bearings. There has been a long-standing requirement for a better understanding of the operational behaviour of plain bearings in such applications. Some results from recent attempts to address the problem are reported in this paper. Because of the high hydraulic pressures exerted to the pump case and gear shafts, it is anticipated that the bearing surface will be subject to severe distortion. A predictive tool for elastohydrodynamic lubrication (EHL) of dynamically loaded bearings was thus employed in the study. The EHL results confirmed the existence of significant elastic deflection of the bearing and housing, and in particular the gear shaft during a loading cycle. This elastic effect contributes to the deterioration of operating conditions of the gear shaft bearings.

Keywords: Elastohydrodynamic Lubrication, Gear pump bearing, tribology, dynamic misalignment

1 INTRODUCTION
Operations of hydraulic fluid gear pumps encounter many tribological phenomena, which involve the lubrication and wear of the gear teeth, journal bearings, side plates and inner profile of the casing. All these effects may influence the efficiency and durability of the gear pump operation [1]. However, very few efforts were reported in respect to the understanding of these tribological matters. So in close collaboration with Parker Hydraulics (Warwick), a study was set up to examine the factors that effect the ultimate load carry capacity of the bearings.

Plain journal bearings are used in hydraulic bearing pumps to enable accurate rotational motions of the gear shafts. They are employed also for friction control and for the maintenance of volumetric efficiency of the pumps. In the case of gear pumps with high discharge pressures, journal bearings are subject to very high specific loads, which lead to arduous operational conditions to the bearings. Their performance becomes a critical issue in affecting reliability of gear pumps. For this reason, there has been a long-standing demand for a better understanding of the working mechanism and behaviour of journal bearings in the type of gear pumps.

Recently, authors carried out an investigation of the hydrodynamic lubrication of the journal bearing used in a gear pump application [2]. It included a sensitivity study of the bearing lubrication in respect to different operational parameters. Although a gear shaft bearing is subject to dynamic load, the journal centre was predicted to move very little during a loading cycle, which may lead to substantial thermal loading in the local region. Predictions of very thin oil film thickness indicate the harsh lubrication regime in the region. Further work in the field is required.

Given the high value of applied load and flexible structures of the pump bearing, casing and gear shaft, it is expected that the bearing and journal will experience significant distortion during operation. It is therefore appropriate to carry out an elastohydrodynamic lubrication (EHL) analysis of the bearing.

EHL analysis of dynamically loaded bearings was first successfully introduced by Oh and Goenka [3] for the analysis of engine bearings. Significant progress was achieved in the area, with many much-improved predictive tools being developed. EHL analysis is now widely used in engine bearing analysis [4-6] and has made its contribution towards a better understanding of the working mechanism of engine bearing systems [7,8].

In this study, an EHL predictive tool was used for the simulation of the lubrication of the gear shaft bearings of an Ultra external gear pump made by Parker Hydraulics. The pump is used in engineering machinery, with a fluid discharge pressure in excess of 20 MPa. This discharge pressure is very high, which causes the journal bearings to be heavily loaded. Therefore in analysing the system, it was essential to take into account of elasticity of the bearing, pump casing and the gear shaft.

Experience indicates that the life of the gear shaft bearings is very sensitive to the bearing materials used and the design of the bearing. This study represents part of our efforts to achieve better understanding of the bearing behaviour in a gear pump application.

2 FUNDAMENTALS OF ANALYSIS
Hydrodynamic lubrication of a plain bearing is governed by the Reynolds Equation, (Eq. 1).

\[
\frac{\partial}{\partial x} \left( \frac{h^3}{\eta} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left( \frac{h^3}{\eta} \frac{\partial p}{\partial y} \right) = 6U \frac{\partial h}{\partial x} + 12 \frac{\partial h}{\partial t}
\]

(1)

where \((x, y)\) are the bearing co-ordinates in the bearing axial and circumferential directions. Variable \((p)\) is the
hydrodynamic pressure in the lubricant at the point (x, y), (η) the lubricant viscosity, (u) the entraining velocity of the bearing and journal motions. The oil film thickness between the two bounding surfaces is represented by (h).

Under heavily loaded conditions, the hydrodynamic pressures generated within the bearing lubricant can be high enough to induce deflection of the bearing surface, which in turn will affect the hydrodynamic action of the bearing. When the interaction between the elastic deflection of bounding surfaces and the hydrodynamic pressure is significant, the bearing operates in a regime of elastohydrodynamic lubrication.

When subjected to high pressure, the viscosity of a lubricant becomes thicker, which is termed the piezo-viscous effect. The thickening of the lubricant viscosity can be approximated by an exponential relationship, as given in the Barus expression in Equation 2.

$$\eta = \eta_0 e^{\alpha p}$$  \hspace{1cm} (2)

where (α) is the pressure viscosity index and is a rheological property of the lubricant.

The inclusion of elastic and rheological effects in the bearing analysis results in much more realistic predictions of bearing performance. It also allows for the study of the bearing as a part of the system. In the case of a gear pump bearing, the system consists of the pump casing, bearing, gear shaft and the lubricant.

The numerical solution of the EHL analysis was carried out using the SABRE-EHL software, which is a computer program developed by Glacier Vandervell Bearings. It employs a finite different approach for the approximation of the Reynolds equation. The elastic compliance coefficients of the bearing, housing and journal structures were extracted from a finite element (FE) analysis. These were then coupled to the Reynolds equation in the solution procedure.

The governing mathematical equations constitute a nonlinear system. The Newton Raphson technique was employed to solve the problem numerically. In this study, an incompressible Newtonian fluid model was used to represent the lubricant, including the Barus relationship to approximate the piezo-viscous effect. The effective oil viscosity was calculated using the operating temperature of the bearing.

3 THE STUDY

The bearing analysed in this study is used in an Ultra gear pump made by Parker Hydraulics. The pump is used in engineering machinery for power transmission and hydraulic control. It employs gears with 12 teeth, and thus exerts cyclic forces on the support journal bearings at every 30° of journal rotation. In the course of development, it was shown that the life of the bearing is very sensitive to the bearing material and designs. However, the use of Glacier DU polymer bush (bearing) has made it a very successful product. This work was directed towards obtaining a better understanding of the advantages and limits of the DU polymer bearing.

For the purpose of the study, the pump was heavily over-rated to highlight the potential limits of operation.

For the purposes of the EHL analysis of the bearing, a quarter of the pump casing together with the fitted bearing, (as shown in Figure 1b), was used in an FE analysis to extract the reduced bearing surface stiffness. A half of the gear shaft was also created to calculate the shaft stiffness in the region of the surface facing the journal bearing. To simplify the modelling process, the gear profile was represented by a cylinder of effective diameter, as shown in Figure 1a. The stiffness matrix was again extracted for inclusion in the EHL analysis.

Figure 1: Structural Models for Stiffness Extraction

1a). FE model of a half of gear shaft with simplified gear; 1b). Structure of a quarter of the casing with fitted bearing.

Having obtained the structural stiffness of bearing and journal, it is then possible to investigate their influence upon the bearing performance.

The generation of the bearing loads are governed by the hydraulic pressure, and the gear pump design configurations. A program for the simulation of the bearing forces was developed by one of the authors. Its details are to be reported in a separate paper. From loading simulations, it was shown that the bearing supporting the driven gear shaft was subject to heavier loads and experiences harsher lubrication conditions[2]. It was thus selected for this study. Since the journal is not symmetrically supported, it is essential to consider the full bearing in the analysis.

Figure 2: Load diagram of the driven gear shaft bearing as generated with a discharge pressure of 30 MPa.
The results presented in this paper are for an operating condition of 2400 rpm, with a fluid discharge pressure of 30 MPa. The bearing load diagram is given in Figure 2. The operating temperature of the bearing lubricant was assumed to be 100 ºC, which gave a viscosity of 0.0097 Pa.s.

The dimensions of the bush are given as follows,
- Bearing Diameter: 25.5 mm
- Bearing Length: 28.0 mm
- Diametric Clearance: 0.04 mm

4 RESULTS AND DISCUSSION

Compared with a conventional rigid analysis, the EHL analysis predicted a much-widened thin oil film region, together with substantial edge loading. The increased circumferential extent of the thin film region correlated well with the size of the observed running-in mark on the bearing surface. However, when the elastic deflection of only the bearing and casing structure was considered, the minimum oil film thickness was predicted to occur at the far end of the bearing, instead of at the gear side as indicated by the running-in mark on bearings which has been in service.

This was considered to be due to neglecting the journal deflection. So in the analyses reported here this effect has been included.

Figure 3 shows the film history map from the EHL analysis. It records the thinnest oil film thickness experienced by any point on the bearing surface during a load cycle. The effect of not including journal elasticity is clearly shown.

4.1 Structural Effects

The inclusion of EHL effects leads to non-symmetric oil film profiles in the axial direction. Ignoring journal deflection gives rise to the prediction of the thin film region at the journal free end side of the bearing, as shown in Figure 3a. However, this axial location conflicts with practical observation. This can be attributed by the non-uniform stiffness profile of the gear pump casing. As shown in Figure 1b, the gear side of the casing structure is less stiff due to the under-cut in the inlet port area. More load would then be supported and transmitted by the journal free end of the casing, which leads to the generation of thinner oil film, and thus higher oil film pressure, in the region.

However, journal flexibility has a more significant effect on the bearing operation. The presence of journal deflection causes a shift of the thin oil film region to the gear side of the bearing, as illustrated in Figure 3b. This skewing of the oil film profile can be interpreted as the presence of a dynamic misalignment, which is caused by the journal bending. In Figure 4, a succession of cross section oil film shapes in the thin oil film region is illustrated. The inclination of the oil film profiles across the bearing is clearly shown in the plot.

Effects of the journal misalignment lead to tapered opening of the oil film profile, as shown by the right portions of the curves. They represent the trend lines of the oil film thickness due to the misalignment. However, elastic deformation of the bearing surface acts to reduce the effects. As shown by the portions of the curves to the left side of the bearing, the oil film shapes are deformed to form an almost levelled region. At the very edge of the bearing at the gear side, the oil film collapses due to the zero oil film pressure at the location.

Since bearing running-in modification and wear are mainly controlled by the oil film thickness, results suggest that the gear side of the bearing is prone to running-in modification. This agrees well with the polished running-in mark on the bearing after service. In Figure 5, a sketch of the running-in polish intensity on the unwrapped bearing surface is given. In the diagram, the severity of the surface polish is shown by the darkness of the colour.
An examination of the bearing surface shows that the region of surface polish is located between about 90 and 200 degrees. Measurement of the bearing angle is made in the direction of the gear rotation, from the discharge port side 90 degrees before the line of gear shaft centres.

4.2 Minimum Oil Film Thickness

Figure 6 shows the variation of the minimum oil film thickness along the centre of the bearing over one shaft revolution. It shows that the minimum oil film thickness along the centre line of the bearing is about 2 micrometres and is located roughly at about 110 degree bearing angle, with a brief excursion to about 180 degree as the minimum oil film thickness approaches its thinnest value. This is in contrast to the nearly stationary minimum oil film location as predicted in the rigid analysis [2].

Figure 6: Variation of the minimum oil film thickness and its location

In the analysis, the absolute minimum oil film thickness was predicted to occur at the edge to the gear side of the bearing. The prediction was so low; it indicated the potential oil film breakdown, as shown in Figure 4. Nevertheless, this was confined to a very narrow strip of the surface at the bearing edge.

4.3 Pressure Generation

An isometric view and a contour map of the oil film pressure distribution at the journal rotation angle of 22 degree are shown in Figure 7. The journal angle correlates to the time when the peak pressure reaches the maximum value. Effects of edge loading due to the dynamic misalignment are very significant. The high pressure region is located only to the gear side of the bearing. As shown in the foregoing subsection, the journal bending gives rise to the generation of a dynamic misalignment. The local EHL effect can however change the film profile to a levelled shape in the thin film region, (see Figure 4). The skewed pressure is produced to enable this change in the film profile. The journal misalignment in the bearing also leads to the generation of a distorted cavitation region as shown by the white region in plot.

Figure 7: Isometric view and contour map of the hydrodynamic pressure as the peak pressure reaches the maximum value

The predicted location of the high pressure and thin oil film thickness region correlates well with the problem area observed on the serviced bearings. The spikes in the pressure prediction are thought to be due to numerical instability, since the bearing is so significantly edge loaded.

4.4 Peak Oil Film Pressure

Figure 8 shows the variation of the peak oil film pressure and its corresponding location during a shaft revolution. It is interesting to note that the location of the peak oil film pressure moves little during the loading cycle, but the value of the peak pressure alters abruptly. The peak pressure reaches a significantly high value of over 300 MPa.

Figure 8: Variation of the peak oil film pressure and its location

From the predicted minimum oil film thickness and the peak oil film pressure, it is indicated that the bearing experiences a harsh lubrication condition during operation. The superior marginal lubrication properties of DU bearing material make significant contribution to the successful operation of the bearing system. The use of a less suitable bearing material could lead to problems of wear and fatigue in the bearing.
Further studies are in progress to establish the influence of specific bearing material properties on the bearing performance in such an application. Results are to be reported in the due course.

5 CONCLUSION
A full EHL analysis of a gear pump bearing has been carried out. The work has shown significant influence of the structural elasticity upon the bearing performance. In particular, the gear shaft bending leads to the occurrence of dynamic misalignment of the journal in the bearing clearance space. As a consequence, the bearing operational environment is adversely affected.

The misalignment gives rise to the occurrence of significant edge loading at the gear side of the bearing. Very thin oil films occur in this region.

The minimum oil film thickness was predicted to be so thin that breakdown of the oil film might be inevitable.

The peak oil film pressure can reach over 300 PMa, which is a very high value for the bearing lining material.

This work has shed some light on the difficulties that need to be overcome with gear pump bearings to provide scope for future uprating. Results have demonstrated good agreement with the practical experiences.

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7 REFERENCES