

The Effects of Different Shaft Speeds on Journal Bearing Performance

Hakan Adatepe

Department of Energy Systems Engineering,
Faculty of Engineering, Giresun University,
28200, Giresun, Turkey,

Abstract:- In this study, the performance of the connecting rod bearing of the one cylinder 4-stroke diesel Ricardo Hydra Engine was investigated at different shaft speeds (2000 rpm, 4000 rpm, 6000 rpm) by examining the performance of the selected connecting rod bearing as an example; radial journal bearing users and bearing designers. The change of shaft speed at the dynamically loaded connecting rod bearing; how the minimum bearing clearance, the maximum oil film pressure, the friction torque, the shaft center in connecting rod bearing and the hydrodynamic power losses that occur in the journal bearing have been theoretically investigated. Using the Ricardo ORBIT V1.2 software analysis were performed using the Finite Volume Method which provides a fast and accurate solution commonly used in the literature. At the different connecting rod bearing shaft speed; how it affected the performance of the journal bearing under dynamic load was carefully examined. From the results of the research; increased shaft speed; It has been determined that the bearing clearance, the maximum oil film pressure is increased, the friction torque and hydrodynamic power losses increase. The shaft orbit in connecting rod bearing is stable. The results of this research showed that the different shaft speeds plays an important role in minimum bearing clearance, the maximum oil film pressure, the friction torque, the shaft orbit in connecting rod bearing and the hydrodynamic power losses under the dynamic load.

Key words: Ricardo Hydra engine, shaft speed, connecting rod bearing

1. INTRODUCTION

It is quite important for the radial journal bearings users and designers to predict the bearings performances before hand as these bearing play a very important role of housing shafts in a diverse field such as in petrol and diesel internal combustion engines, jet engines, compressors, gear and piston pumps, rolling mills, mechanical presses etc. The performance of dynamically loaded bearings is also important issue for the engine industry. Recently, two approaches may be observed in the course of the development of bearing design and analysis. On the one hand, there is a need in industry for a quick method or algorithm for engineers who desire to have a reliable and rapid design tool. On the other hand, the more involved applications drive researchers to explore lubrication phenomena in a comprehensive depth, using very complex mathematical methods, with a hope to solve a more realistic bearing problem. Lubrication theory for the dynamically loaded journal bearing is mathematically complex and, over the last few decades, several analytical approaches have been proposed. The multigrid techniques based on the Elrod algorithm and the finite element methods of analysis are among the most popular. The finite element methods are probably the most accurate and versatile, but tend to be very time consuming and require high level of knowledge, not accessible to the common designer and, so, remaining confined to research and development. Therefore, based on simplifying premises, engineers and designers prefer to use simpler and still accurate methods, such as the mobility method and the impedance method. In general, these two approximate techniques, which belong to the category of rapid methods, are employed to perform analysis of simple journal bearings. Rotational speed directly affects the performance of journal bearings operating under dynamic loads. For this reason, prediction of how will the speed affect the bearings minimum clearance, peak oil film pressure, friction torque, oil flow rate, journal orbit and power losses on the bearing is substantially important for researchers dealing with designing of journal bearings. In this study, the performance of a connecting rod bearing of a 4 cycle 6 VEB-X MK III Ruston and Hornsby diesel engine under different engine speeds was studied. The load example used here was selected based on a similar load example widely used and well known in literature (Adatepe H.). A journal bearing is common in many types of machinery. The main function is to support radial load and facilitate motion as well as transfer of power. A journal bearing consists of two main components where the shaft called journal rotates freely in its bushing known also as bearing. Ahmad M.A., Kasolang S. and al. investigated the effects of the oil groove position on the friction force and moment in hydrodynamic journal bearings. Axial groove is a common supply method of distributing lubricant within a journal bearing. Lubricant is generally fed at a specific supply pressure to ensure that the journal and the bearing surface are separated. Shearing action between lubricant and bearing parts creates frictions which contribute to power loss in journal bearing. In this study, an experimental work conducted to determine the effect of oil groove location on torque and frictional force in hydrodynamic journal bearing. A journal diameter of 100mm with a 1/2 length-to-diameter ratio used. The shaft speed set at The supply pressure was set to 0.2, 0.5, and 0.7 MPa at seven different groove locations, namely, -45°, -30°, -15°, 0°, +15°, +30°, and +45°. Measurements of torque and frictional force obtained for speed values of 300, 500 and 800 rpm at different radial loads. They observed that the change in oil groove location has affected the fluid frictional force and friction coefficient to some extent (Ahmad et al. 2014). Brito F.P., Miranda A.S. and Fillon M. investigated the effects of grooves in single and twin axial groove journal bearings under varying load direction. Multi axial groove hydrodynamic journal bearings often preferred over single

grooved ones to improve lubricant distribution. However, when assessing the influence of loading angle for a twin axial groove bearing, the authors detected the occurrence of phenomena such as strong negative flow rate in one of the grooves for a broad range of loading angles which was deleterious to bearing performance. Conversely, single groove bearings display acute starvation for specific loading ranges. The authors used a previously proposed thermohydrodynamic approach to compare the role of single and twin groove bearings under variable loading direction. Results show that a groove deactivation strategy (using check valves) might be a best-of-both-worlds strategy, optimizing groove flow rate distribution, reducing temperature and eccentricity levels (Britoet al.2012).In the present study, the performance of the connecting rod bearing of the one clinder 4-stroke diesel Ricardo Hydra Engine was investigated at different shaft speeds (2000 rpm, 4000 rpm, 6000 rpm) by examining the performance of the selected connecting rod bearing as an example; radial journal bearing users and bearing designers.

2. MATERIAL AND METHODS

Using the Ricardo ORBIT V1.2 software analysis were performed using the Finite Volume Method which provides a fast and accurate solution commonly used in the literature (Ricardo, 2000). The starting point of any fluid-film lubrication analysis is solving the general hydrodynamic Reynolds Equation as expressed in Equation (1) subject to the edge pressure boundary conditions given in Equations (2) and (3) Thus,

$$\frac{\partial}{\partial z} \left(\frac{h^3}{12\mu} \frac{\partial P}{\partial z} \right) + \frac{1}{R^2} \frac{\partial}{\partial \theta} \left(\frac{h^3}{12\mu} \frac{\partial P}{\partial \theta} \right) = \frac{1}{R} \frac{\partial h}{\partial \theta} + \frac{\omega}{2} \frac{\partial h}{\partial z} + \frac{\partial h}{\partial t} \quad (1)$$

with the following boundary conditions:

$$z = 0, \quad P(z, \theta) = P_{i,j} \quad (2)$$

$$z = L, \quad P(z, \theta) = P_{n,j} \quad (3)$$

The finite volume method is a powerful technique that overcomes the limitations of other approaches. It conserves mass and rigorously determines the point of rupture and reformation of the oil film within the bearing clearance zone via application of the JFO (Jakobsson-Floberg-Olsson) boundary conditions. A mass conserving algorithm originally proposed by Elrod is implemented to solve Reynolds Equation (Woods, 1989). The key point of this approach are:

Consideration of compressibility effects of the lubricant via usage of the bulk modulus β , where

$$\beta = \rho \frac{\partial P}{\partial \rho} \quad (4)$$

Introduction of the nodal mass fraction variable α which has dual interpretations,

$$\alpha_{i,j} = \frac{\rho}{\rho_c} = \frac{\text{Oil density}}{\text{Density at cavitation}} \quad (\text{flooded cell; } \alpha_{i,j} \geq 1) \quad (5)$$

$$\alpha_{i,j} = \text{void fraction} \quad (\text{cavitated cell; } \alpha_{i,j} < 1) \quad (6)$$

Introduction of a nodal switch function such that,

$$g_{i,j} = 1 \quad (\text{flooded cell; } \alpha_{i,j} \geq 1) \quad (7)$$

$$g_{i,j} = 0 \quad (\text{cavitated cell; } \alpha_{i,j} < 1)$$

Using Equation (5)-(7) to express nodal oil film pressures as

$$P_{i,j} = P_c + \beta g_{i,j} (\alpha_{i,j} - 1) \quad (8)$$

Writing the governing Reynolds Equation in terms of the variable α as

$$\frac{\partial m_\theta}{\partial \theta} + \frac{\partial m_z}{\partial z} = \frac{\partial |\rho_c \alpha_{i,j} h_{i,j}|}{\partial t} \quad (9)$$

where

$$(m_\theta)_c = \rho_c \frac{V_\theta}{2} \left(\alpha_{i,j-1} h_{i,j-1} [1 - g_{i,j-1}] + g_{i,j-1} h_{i,j-1} + \frac{g_{i,j-1} g_{i,j}}{2} [h_{i,j} - h_{i,j-1}] \right) \quad (10)$$

$$(m_z)_p = \left(\frac{h^3}{12\mu} \right)_{avg} \beta \rho_c \left(\frac{g_{i-1,j} [\alpha_{i-1,j-1}] + g_{i,j} [\alpha_{i,j-1}]}{\Delta z} \right) \quad (11)$$

$$(m_z)_c = \rho_c \frac{V_z}{2} \left(\alpha_{i-1,j} h_{i-1,j} [1 - g_{i-1,j}] + g_{i-1,j} h_{i-1,j} + \frac{g_{i-1,j} g_{i,j}}{2} [h_{i,j} - h_{i-1,j}] \right) \quad (12)$$

And the terms with subscripts p and c represent the Poisseuille and Couette flow components of the mass flux terms m_θ and m_z in the θ and z directions respectively.

Application of Equation (9) to computational cell that define the bearing clearance zone generates a system of linear algebraic equations of the form

$$[A]\{\alpha\} = \{R\} \quad (13)$$

where $[A]$ is the coefficient matrix associated with the column of unknown values of $\{\alpha\}$, and $\{R\}$ is a column of known terms associated with Equation (9). The ADI (Alternating Direction Implicit) technique is employed to solve Equation (9) and this is a computationally of the coefficient matrix $[A]$.

The finite volume technique is the preferred method amongst the three solvers since it a) is mass conserving b) tracks cavitating regions rigorously and c) is computationally fast. In particular, the finite volume method is the solver of choice when it comes to accurately accounting for oil holes and calculation of rise in oil temperature (Gulwadi, 2003).

3. RESULTS AND DISCUSSION

Bearing dynamic load diagram effect of full circumferential grooves. Results from three shaft speed values on the connecting rod bearing are presented and discussed in this section. The graph of the change of the bearing load (total bearing load, horizontal and vertical components of the bearing load) to the crank angle of the 4-stroke diesel Ricardo Hydra Engine connecting rod bearing is shown in Figure 1.

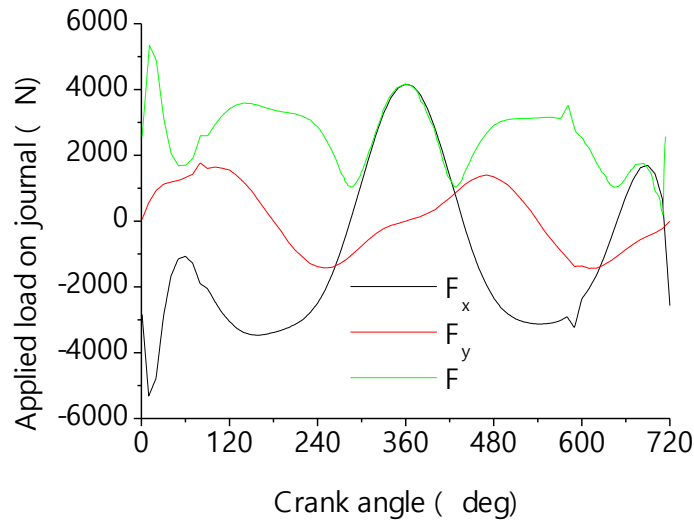


Figure 1. Graph of change of the bearing load (total bearing load, horizontal and vertical components of bearing load) to the crank angle of 4000 rpm

When the analysis of Figure 1 was analyzed, it was seen that the maximum bearing load was 5349,3 N and 10° crank angle, while the minimum bearing load was 710° at the crank angle and 193 N. The graph of the change in the shaft speed of the journal bearing when the shaft speed of the journal bearing is changed in the connecting rod bearing the same dynamic load effect is given in Figure 2.

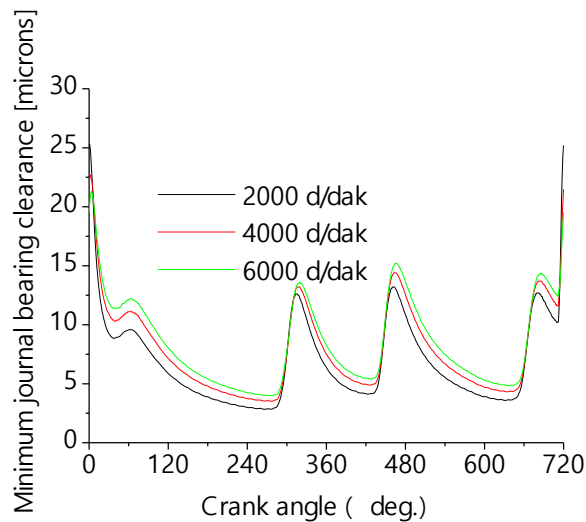


Figure 2. Effects of varying shaft speed on minimum journal bearing clearance

It shows that as the shaft speed increases, the maximum values of the minimum journal bearing clearance in the connecting rod bearing generally decrease, but the minimum values of the minimum journal bearing clearance increase. When the data in Figure 2 were examined, it was seen that the minimum value of the minimum journal bearing clearance was 2,84 μm at the crank angle of 265°, the highest value was 25,3 μm and it was at the value of 0,75° crank angle. At the shaft speed 4000 rpm, the lowest value of the minimum journal bearing clearance occurs at crank angle of 275° shaft speed a magnitude of 3,52 μm , whereas its highest value is 22,7 μm and occurs at a crank angle of 2,12°. At a shaft speed of 6000 rpm, the minimum value of the minimum bearing clearance is 3,99 μm at the crank angle of 277 degrees, and the maximum value of the minimum bearing gap is at the crank angle of 3,87° and 21,3 μm . The increase in minimum journal bearing clearance at the same time means the increase in oil film thickness keeping the minimum oil film thickness at higher values is good for bearing performance. This

helps reducing the risk of interaction between shaft and bearing and hence enabling safe operation of the bearing generally speaking, in terms of minimum journal bearing clearance bearing performance is enhanced at higher shaft speed than at lower shaft speed. Figure 3 shows variation of peak oil film pressure at different shaft speeds under constant dynamic load.

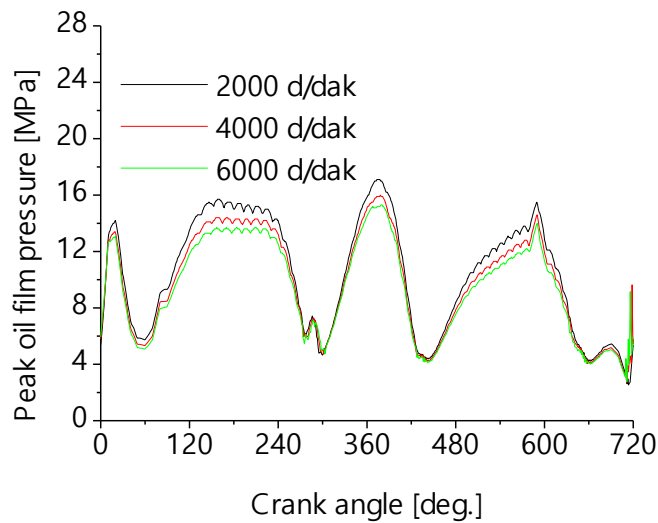


Figure 3. Effects of varying shaft speed on peak oil film pressure

At constant crank angle, the maximum values of peak oil film pressure in Figure 3 are clearly seen to decrease as the shaft speed increases. From the figure it is seen that the lowest of the peak oil film pressure at shaft speed 2000 rpm occurs at 713,7° crank angle with a value of 2,53 MPa, whereas the peak oil film pressure is 17,1 MPa and occurs at 373° crank angle. When the shaft speed is 4000 rpm, the lowest value of the peak oil film pressure is 2,72 MPa and takes place at a crank angle of 711,3°. The greatest of the maximum oil film pressure in this case is exhibited at a crank angle of 378° with a value of 16 MPa. At shaft speed 6000 rpm, the smallest value of peak oil film pressure is 2,76 MPa and occurs at a crank angle of 711,25°, while the maximum value is 15,3 MPa and is found to take place at a crank angle of 378,8°. The increase in shaft speed led to corresponding increase in the minimum oil film thickness and the increased oil film layer affected peak oil film pressure negatively. Provided that the load is the same, it is important that at shaft speed, oil film is kept at low level for better journal bearing performance. Based on the maximum value of peak oil film pressure, it is also said here, that the journal bearing performance at shaft speed is better than that at shaft speed. The graph showing variation of effects of shaft speed on friction torque is given in Figure 4.

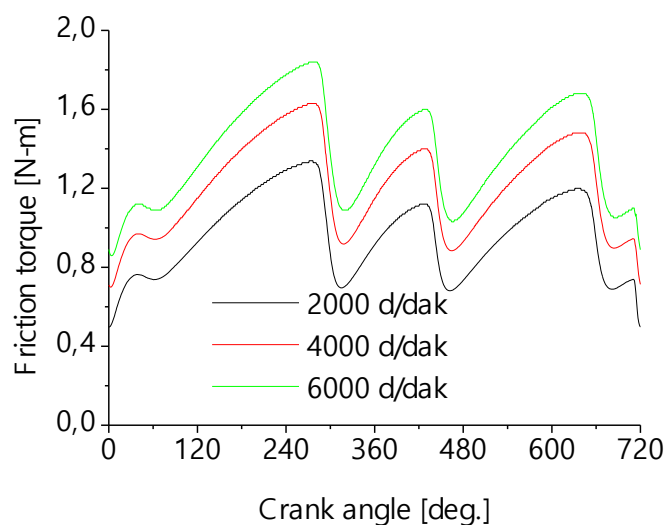


Figure 4. Effects of shaft speed on friction torque

From Figure 4, it is clearly seen that at constant crank angle the shaft speed increases the friction torque. Thus, at shaft speed of 6000 rpm, the friction torque value of 1,84 Nm. In addition, it was found that at shaft speed 2000 rpm the friction torque exhibits its minimum value of 0,497 Nm at a crank angle of 0° and its maximum value of 1,84 Nm at 270,8° crank angle. An increase in friction torque or in other words, high friction torque negatively affects the bearing performance. The graph showing the effects of engine shaft speed variations on the power losses on the journal bearing is given in Figure 5. From the figure, it is clearly seen that at the same crank angle, as the shaft speed increases the power losses in the radial journal bearings generally increase, too. The increase in shaft speed has led to an increasing oil film thickness and this led to power losses. generally, under the same load conditions, power losses in journal bearings tend to increase at different running shaft speed, In order to lower the power loss resulted from friction, it is necessary to reduce the coefficient of friction.

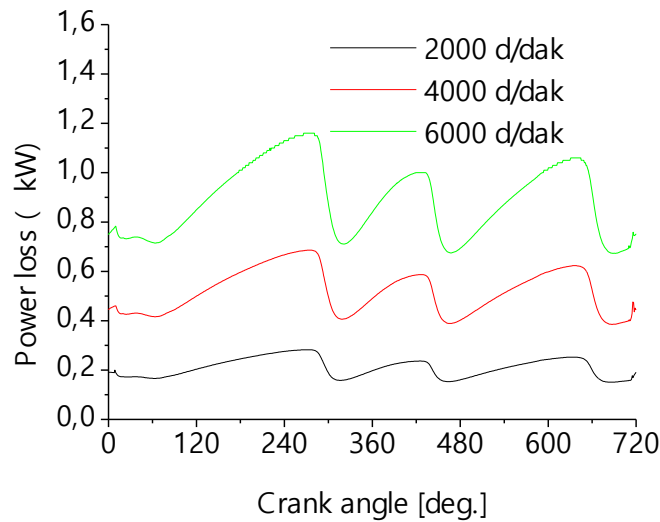


Figure 5. Effects of shaft speed variations on power loss

The effects of shaft speed variations on the journal orbits are shown in Figure 6.

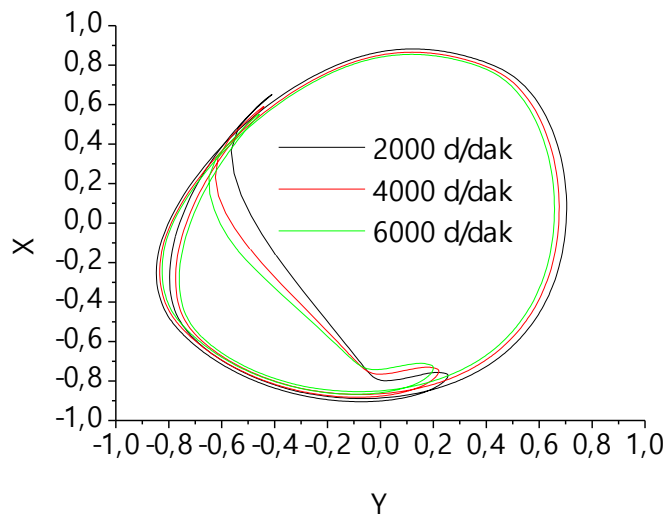


Figure 6. Effects of shaft speed variations on journal orbits

In is seen from Figure 6 that the journal orbits look very similar to each other and that the most suitable journal orbit is obtained at 5 bar shaft speed. The circumferential and transversal pressure distributions of the journal orbit under dynamic load and rotating at 4000 rpm are shown in Figure 7. Analysis Figure 7, it is found that the peak oil film pressure is 21,01 MPa and occurs at a crank angle of 378.

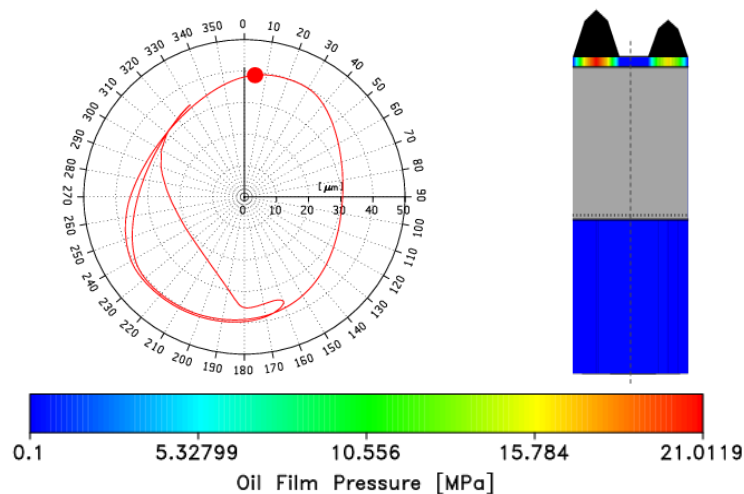


Figure 7. Journal orbit in a dynamically loaded journal bearing running at 4000 rpm (a); pressure distributions on circumferential direction (b); pressure distributions on transverse direction (c)

4. CONCLUSIONS

In this study, it has been investigated how the minimum journal bearing clearance value, the maximum oil film pressure value, friction moment value, power losses and shaft orbit are affected by the increase in shaft speed in connecting rod bearing under dynamic load. The results are summarized below.

As the shaft speed increases, it is seen that the maximum values of the minimum journal bearing clearance in the connecting rod bearing are generally increased, but the minimum values of the minimum journal bearing clearance decrease.

As the shaft speed increases, it is seen that the peak oil film pressure in the connecting rod bearing are generally increased, but the minimum values of the peak oil film pressure decrease.

It has been determined that with the increase in the shaft speed in the connecting rod bearing, the friction moment values occurring in the journal bearing have also increased.

An increase in shaft speed on a radial journal bearing running under a dynamic load generally tends to increase the power losses on the journal bearing.

The journal orbits look very similar to each other and that the most suitable journal orbit is obtained at 4000 rpm shaft speed.

5. REFERENCES

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