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The Effects of Different Oil Supply Pressure on Journal Bearing Performance

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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 oil supply pressure (3 bar, 5 bar, 7 bar) by examining the performance of the selected connecting rod bearing as an example; radial journal bearing users and bearing designers. Width the change of the oil supply pressure at the dynamically loaded connecting rod bearing; how the minimum bearing clearance, the maximum oil film pressure, the friction torque, the shaft orbit 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 width; how it affected the performance of the journal bearing under dynamic load was carefully examined. From the results of the research; increased oil supply pressure; It has been determined that the bearing clearance, the maximum oil film pressure is increased, the friction torque and hydrodynamic power losses increase. The orbit drawn by the shaft orbit in connecting rod bearing is stable. The results of this research showed that the different oil supply pressure 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.

Keywords: Ricardo Hydra Engine, oil supply pressure, connecting rod bearing.

Farklı Yağ Besleme Basıncının Kaymalı Yatak Performansına Etkisi

Öz

Bu çalışmada, tek silindirli 4 zamanlı dizel Ricardo Hydra motorunun biyel kolu yatağının performansı farklı yağ besleme basıncında (3 bar, 5 bar, 7 bar) incelenmiştir. Dinamik olarak yüklenen biyel kolu yatağındaki yağ besleme basıncı değişiminin; minimum yatak boşluğuna, maksimum yağ filmi basıncına, sürtünme torkuna, mil merkezinin çizdiği yörüngeye ve yatak yuvasında meydana gelen hidrodinamik güç kayıplarına nasıl tesir ettiği teorik olarak incelenmiştir. Ricardo ORBIT V1.2 yazılımı kullanılarak, literatürde yaygın olarak kullanılan hızlı ve doğru bir çözüm sunan Sonlu Hacim Yöntemi ile analizler gerçekleştirilmiştir. Farklı yağ besleme basınçlarında; biyel kolu yatağının dinamik yük altındaki performansını nasıl etkilediği dikkatlice incelenmiştir. Araştırma sonuçlarından; artan yağ besleme basıncı; Yatak boşluğunun, maksimum yağ filmi basıncının arttırıldığı, sürtünme momentinin ve hidrodinamik güç kayıplarının arttığı belirlenmiştir. Mil merkezinin çizdiği yörüngesinde hemen hemen sabit olduğu görülmüştür. Bu araştırmanın sonuçlarırnda, farklı yağ besleme basıncılarının yatağın minimum yatak boşluğuna, maksimum yağ filmi basıncına, sürtünme torkuna, biyel kolu yatağındaki mil merkezi yörüngesine ve dinamik yük altındaki hidrodinamik güç kayıplarına tesir ettiğini göstermiştir.

Anahtar Kelimeler: Ricardo Hydra motor, yağ besleme basıncı, biyel kolu yatağı.

1. Introduction

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 at 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 ½ length-to-diameter ratio used. The oil supply pressure 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 (Brito *et 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 oil supply pressure (3 bar, 5 bar, 7 bar) 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,

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

with the following boundary conditions:

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

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

The finite volume method is a powerful technique that overcomes the limitations of other approaches. It a 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,

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

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

Introduction of a nodal switch function such that,

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

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

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

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

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

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

where

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

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

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

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 pespectively.

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

$$[A]\{\alpha\} = \{R\} \tag{13}$$

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 oil supply pressure 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.

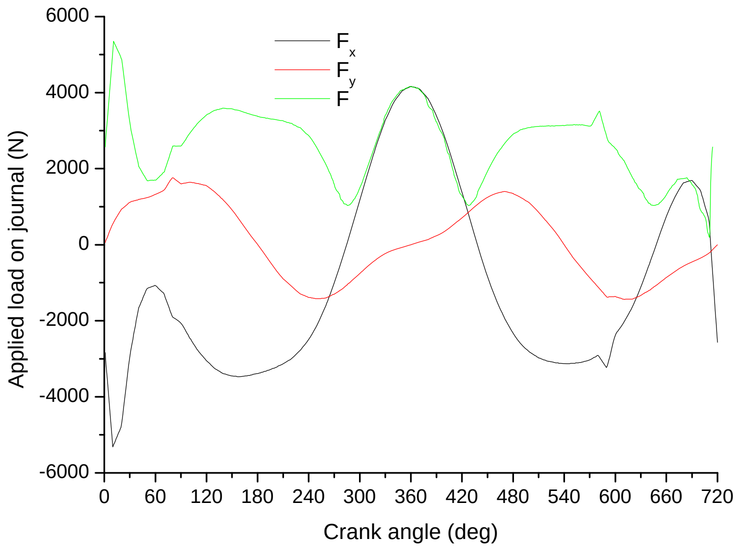

Figure 1. Graph of change of the bearing load (total bearing load, horizontal and vertical components of bearing load) to the crank angle of 4000 rpm

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 oil supply pressure of the journal bearing when the oil supply pressure of the journal bearing is changed in the connecting rod bearing the same dynamic load effect is given in Figure 2. It shows that as the oil supply pressure increases, the maximum and minimum values of the minimum journal bearing clearance in the connecting rod bearing generally increase.

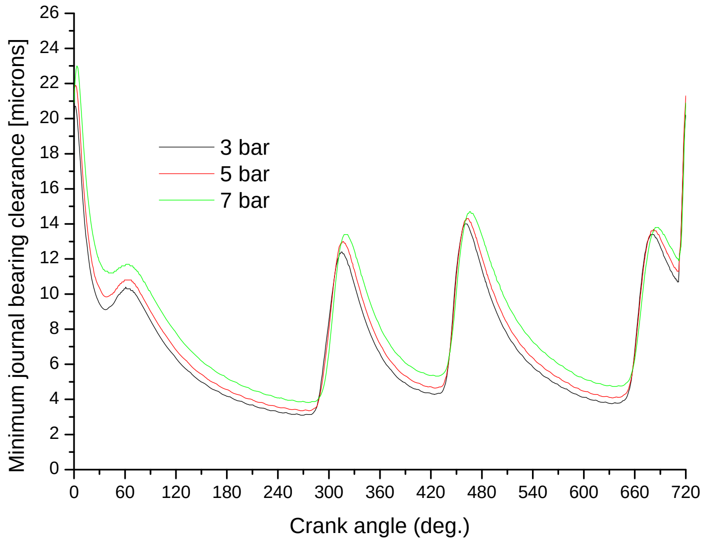

Figure 2. Effects of varying oil supply pressure on minimum journal bearing clearance

When the data in Figure 2 were examined, it was seen that the minimum value of the minimum journal bearing clearance was 3,1 µm at the crank angle of 267°, the highest value was 20,7 µm and it was at the value of 1° crank angle. At the oil supply pressure 5 bar, the lowest value of the minimum journal bearing clearance occurs at crank angle of 266° oil supply pressure a magnitude of 3,36 µm, whereas its highest value is 21,9 µm and occurs at a crank angle of 2°. At oil supply pressure of 7 bar, the minimum value of the minimum bearing clearance is 3,82 µm at the crank angle of 274 degrees, and the maximum value of the minimum bearing gap is at the crank angle of 3,5° and 23 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 oil supply pressure than at lower oil supply pressure. Figure 3 shows variation of peak oil film pressure at different oil supply pressure under constant dynamic load. At constant crank angle, the maximum values of peak oil film pressure in Figure 3 are clearly seen to decrease as the oil supply pressure increases.

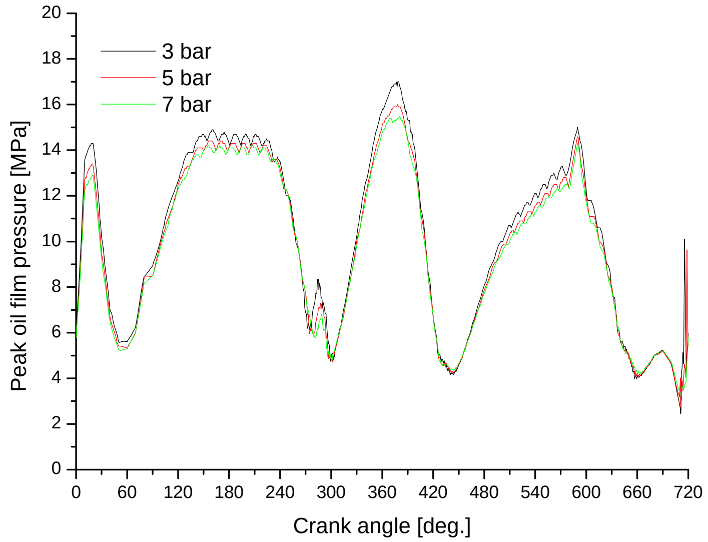

Figure 3. Effects of varying oil supply pressure on peak oil film pressure

From the figure it is seen that the lowest of the peak oil film pressure at oil supply pressure 3 bar occurs at 711° crank angle with a value of 2,44 MPa, whereas the peak oil film pressure is 17 MPa and occurs at 376,5° crank angle. When the oil supply pressure is 5 bar, 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 oil supply pressure 7 bar, the smallest value of peak oil film pressure is 3,22 MPa and occurs at a crank angle of 711,8°, while the maximum value is 15,5 MPa and is found to take place at a crank angle of 380,5°. The increase in oil supply pressure 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 oil supply pressure, 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 oil supply pressure is better than that at oil supply pressure. The graph showing variation of effects of oil supply pressure on friction torque is given in Figure 4.

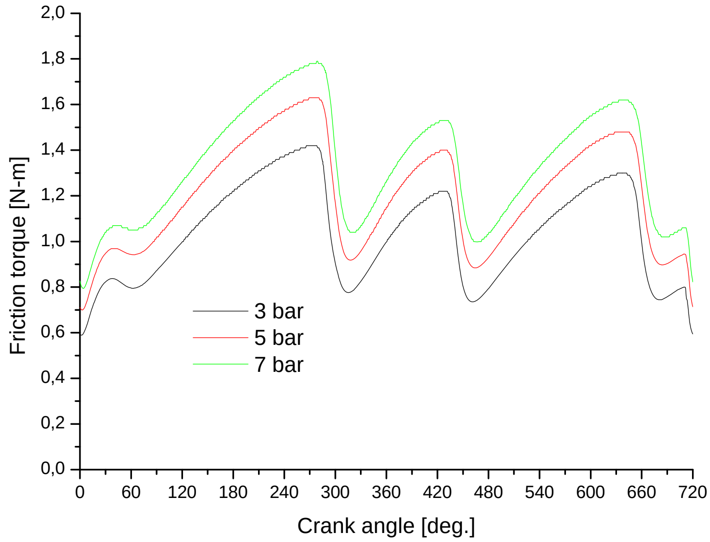

Figure 4. Effects of oil supply pressure on friction torque

From Figure 4, it is clearly seen that at constant crank angle the oil supply pressure increases the friction torque. Thus, at oil supply pressure of 7 bar, the friction torque value of 1,79 Nm. In addition, it was found that at oil supply pressure 3 bar the friction torque exhibits its minimum value of 0.589 Nm at a crank angle of 1° and its maximum value of 1,42 Nm at 266° 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 oil supply pressure variations on the power losses on the journal bearing is given in Figure 5.

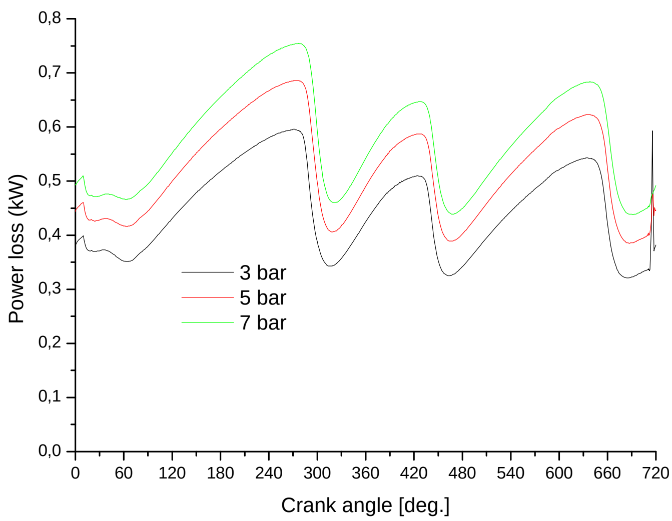

Figure 5. Effects of oil supply pressure variations on power loss

From the figure, it is clearly seen that at the same crank angle, as the oil supply pressure increases the power losses in the radial journal bearings generally increase, too. The increase in oil supply pressure 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 oil supply pressure, In order to lower the power loss resulted from friction, it is necessary to reduce the coefficient of friction. The effects of oil supply pressure variations on the journal orbits are shown in Figure 6. 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 oil supply pressure. 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.

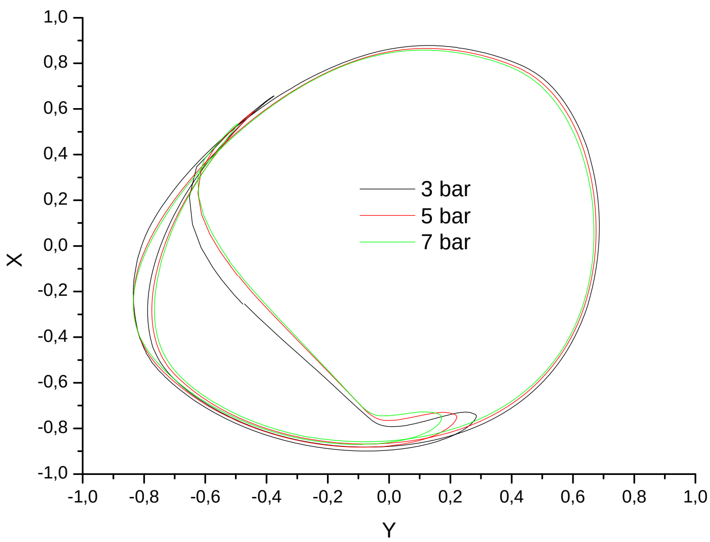

Figure 6. Effects of oil supply pressure variations on journal orbits

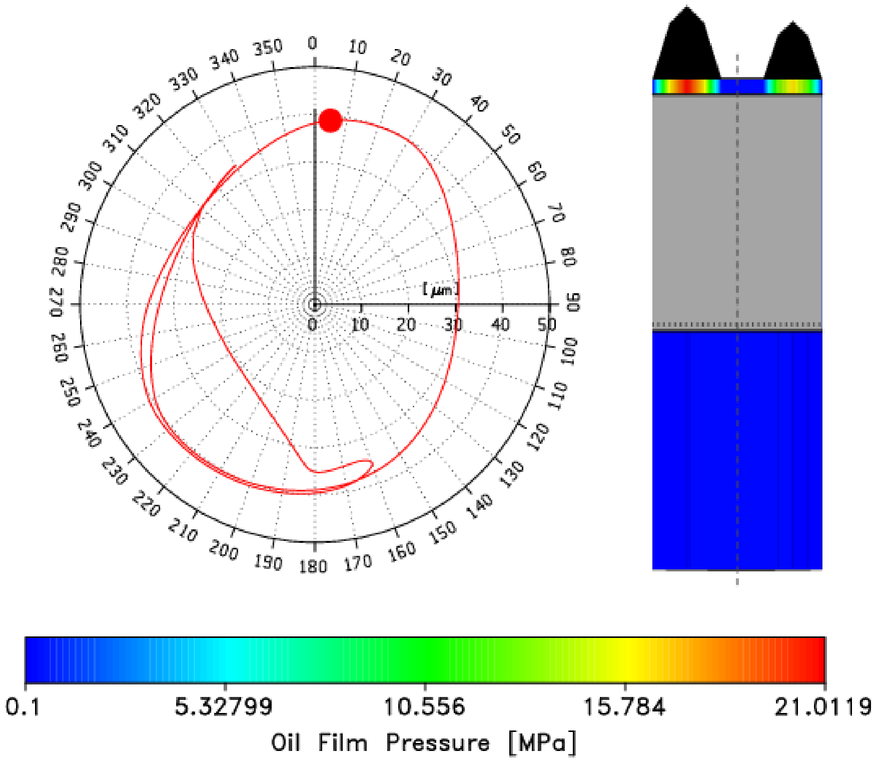

Figure 7. Journal orbit in a dynamically loaded journal bearing running at 4000 rpm (a); pressure distributions on circumferential direction (b); pressure distributions on transverse direction (c)

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 oil supply pressure in connecting rod bearing under dynamic load. The results are summarized below.

As the oil supply pressure 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.

Journal bearing performance at higher oil supply pressure is better than the performance at lower oil supply pressure as far as the peak oil film pressure is concerned.

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

An increase in oil supply pressure 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 5 bar oil supply pressure.

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