Analysis of multi lobe journal bearings with surface roughness using finite difference method

K PhaniRaja Kumar 1+, SUdaya Bhaskar 2 and MManzoor Hussain 3

1 Project Manager, IBG Manufacturing, Tech Mahindra America, USA
2 Assistant Professor, Department of Mechanical Engg., Al Habeeb College of Engg & Tech, Telangana, India.
3 Professor, Department of Mechanical Engineering, JNT University Hyderabad, Telangana, India.
Corresponding Author E-mail: phani.katuru@gmail.com

Abstract. Multi lobe journal bearings are used for high operating speeds and high loads in machines. In this paper symmetrical multi lobe journal bearings are analyzed to find out the effect of surface roughness during non-linear loading. Using the fourth order Runge-Kutta method, time transient analysis was performed to calculate and plot the journal centre trajectories. Flow factor method is used to evaluate the roughness and the finite difference method (FDM) is used to predict the pressure distribution over the bearing surface. The transient analysis is done on the multi lobe journal bearings for three different surface roughness orientations. Longitudinal surface roughness is more effective when compared with isotopic and traverse surface roughness.

1. Introduction
Hydrodynamic journal bearings are used for carrying high loads in different machines. In these fluid film bearings there is a relative motion between the bearing and journal surfaces, when the journal rotates. The fluid is dragged into the wedge shape gap due to which hydrodynamic pressures are generated and able to support an externally applied load. The importance of roughness in predicting bearing performance has gained considerable attention in Tribology. Due to the operation of hydrodynamic bearings at high speed, the problems of instability is encountered. Instability of the journal bearings will ruin the bearings and machine itself. Multi lobe journal bearings maintain the stability of the bearings at high speeds and different loading conditions.

2. Fundamentals of multi lobe bearings
The Multi-lobe bearings have lobes depending on the number of lobes they are classified as two lobe, three lobe and four lobes. In two lobe they accommodate clearance, preload, offset and provision for lubricants to take away the heat during running conditions. Due to this load carrying capacity is increased and in turn produces a stabilizing effect on the shaft. The multi lobe bearings are more stable in major considerations for dynamic characteristics. Non-circular bearing geometry enhances shaft stability under proper working conditions; they reduce power losses and increase oil flow rate, due to which bearing temperature is reduced. Non-circular journal bearings are used in high speed machinery.
3. Analysis of multi lobe journal bearing
The Finite difference method is used for solving the modified Reynolds equation. For steady loading and incompressible lubricants the general Reynolds equation is in the form of

\[
\frac{\partial}{\partial x} \left( h^{n+2} \phi_{xx} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left( h^{n+2} \phi_{yy} \frac{\partial p}{\partial y} \right) = 6 \nu U^n \frac{\partial}{\partial x} \left( h + \sigma \phi_{xx} \right) \tag{1}
\]

The above equation (1) is a second-order, non-homogenous, partial differential equation. Analytically it is difficult to solve the equation. Numerical approach is done to solve the equation, initially some of the variables are made dimensionless to reduce the number of variables. The projected area of the bearing surface is divided into mesh sizes of \((\Delta \theta \times \Delta y)\) as shown in figure 1. Using central difference quotients, the equation is written as

\[
\phi_{xx}^{P} \left[ \frac{p_{l+1,j} - 2p_{l,j} + p_{l-1,j}}{\Delta \theta^2} \right] + \left( \frac{D}{L} \right)^2 \phi_{yy}^{P} \left[ \frac{p_{i,j+1} - 2p_{i,j} + p_{i,j-1}}{\Delta y^2} \right] - \frac{3}{2} \varepsilon \left[ \frac{p_{l+1,j} - p_{l-1,j}}{h_i (\Delta \theta)} \sin \theta_i \right] = -\varepsilon \frac{\sin \theta_i}{h_i^3} \tag{2}
\]

Where \(p_{l,j}\) is the pressure at any point \((l,j)\), 
\(h_i\) is the film thickness at any point \((i,j)\),
\(p_{l+1,j}, p_{l-1,j}, p_{l,j+1}\) and \(p_{l,j-1}\) are the pressures at the four adjacent points of \(p_{l,j}\).

The three types of surface roughness used in this analysis are Isotropic surface roughness \((\gamma=1)\), Transverse surface roughness \((\gamma=0.3)\) and Longitudinal surface roughness \((\gamma=3)\). Generalized Reynolds equation of steady state is solved in the finite difference method with over relaxation factor to obtain the non-dimensional pressure distribution in each lobe.

**Figure 1:** Finite difference scheme - Mesh
The convergence criterion adopted for pressure calculation is

$$\left| -\sum_{i} \frac{p_{i,old}}{\sum_{i} p_{i,new}} \right| \leq 10^{-4}$$

with a chosen bearing eccentricity ratio and attitude angle picked at random, say $\phi_{int}$ there will be set of lobe eccentricities and attitude angles for which the solution of Reynolds equation will provide the magnitude of forces generated in the pressure wedge both the upper and the lower wedges must be zero. The basic differential equation, which governs the pressure distribution in the lubricant fluid inside the gap of a multi lobe journal bearing is Reynolds equation for transient state is given by

$$\frac{\partial}{\partial x} \left( h^3 \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial z} \left( h^3 \frac{\partial p}{\partial z} \right) = 6\mu U \frac{\partial h}{\partial x} + 12\mu \frac{\partial h}{\partial t} \tag{3}$$

where $t$ is time, $x$ and $z$ are the Cartesian coordinates, $\mu$ is the absolute viscosity of the lubricant, $U$ is the peripherical velocity of the rotor and $h$ is the film thickness, according to the bearing geometry the hydrodynamic pressure is calculated by using the above expressions for transient state. Now the Reynolds equation is solved numerically for pressure by finite difference method by satisfying the Reynolds boundary conditions. The problem of presenting eccentricity as a function of the Sommerfeld number presents some difficulty. The bearing eccentricity has advantage of simplicity. It is a physical dimension easily visualized in that it tells how far the center of the shaft is away from the center of the bearing.

### 4. Results

The variation of Non Dimensional Load and Non dimensional frictional force of multi lobe bearing for various surface roughnesses’s are determined and plotted. There is a deviation in results due to the surface roughness profiles.

**4.1 Variation of Non Dimensional pressure in multi lobe journal bearings with Longitudinal, Transverse and Isotropic surface roughness at eccentricity ratio = 0.6 are shown in figures 2 to 4.**

![Figure 2: Variation of Non dimensional pressure in two lobe journal bearing for different surface roughness at eccentricity ratio = 0.6](image)
Figure 3: Variation of Non dimensional pressure in three lobe journal bearing for different surface roughness at eccentricity ratio = 0.6

Figure 4: Variation of Non dimensional pressure in four lobe journal bearing for different surface roughness at eccentricity ratio = 0.6
4.2 Non Dimensional Load at different eccentricity for different surface roughness

**Figure 5:** Variation of Non Dimensional Load at different eccentricity ratios for two lobe journal bearing

**Figure 6:** Variation of Non Dimensional Load at different eccentricity ratios for three lobe journal bearing
Figures 5 to 7 represent the effect of load carrying capacity at different eccentricity ratios for two lobe, three lobe and four lobe journal bearings. It is observed from the figures that three lobe journal bearing and four lobe journal bearing are having maximum load carrying capacity when compared to two lobe journal bearing. The longitudinal surface roughness has more effect on maximum load carrying capacity when compared to transverse and isotropic surface roughness.

4.3 Non Dimensional Friction force at different eccentricity for different surface roughness

Figure 7: Variation of Non Dimensional Load at different eccentricity ratios for Four lobe journal bearing

Figure 8: Variation of Non Dimensional Friction force at different eccentricity ratios for two lobe journal bearing
Figure 9: Variation of Non Dimensional Friction force at different eccentricity ratios for three lobe journal bearing

Figure 10: Variation of Non Dimensional Friction force at different eccentricity ratios for four lobe journal bearing

Figures 8 to 10 represents the effect of frictional force at different eccentricity ratios for two lobe, three lobe and four lobe journal bearings. It is observed from the figures that two lobe journal bearing is having maximum frictional force when compared to three lobe and four lobe journal bearing. The longitudinal and transverse surface roughness has more effect on friction force when compared to isotropic surface roughness.
From figure 11, it has been observed that the three lobe journal bearings are more stable when compared to two lobe and four lobe journal bearings. The path trajectories are plotted at eccentricity ratio 0.6, with longitudinal surface roughness.

5. Conclusions
The analysis is done on the multi lobe journal bearings, with longitudinal, transverse and isotropic surface roughness orientations. The Longitudinal surface roughness has more effect on Multi lobe journal bearings. Pressure is maximum exactly midway between the lobes. Due to this, the rotor will be well balanced while rotating and vibrations due to bearing will be minimum. From the results it indicates that the surface roughness plays a major role on the performance of static characteristics of the multi lobe journal bearings. From the graphs it is observed that the three lobe and four lobe journal bearing performance is almost close when compared with two lobe journal bearings. Longitudinal surface roughness is more effective when compared with isotopic and traverse roughness.

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