Non Linear Analysis on Multi Lobe Journal Bearings

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Abstract: Multi lobe journal bearings are used in machines which operate at high speeds and high loads. In this paper the multi lobe bearing are analyzed to determine the effect of surface roughness during non linear loading. A non-linear time transient analysis is performed using the fourth order Runge Kutta method. The finite difference method is used to predict the pressure distribution over the bearing surface. The effect of eccentric ratio is studied and the variation of attitude angle is discussed. The journal center trajectories were calculated and plotted.

Keywords: multi lobe, journal bearings, eccentricity, non linear, Sommerfeld number, attitude angle, finite difference method.

1. INTRODUCTION

In hydrodynamic fluid film bearings there is a relative motion between two mechanical surfaces with a particular wedge shape. The fluid is dragged into the film and hydrodynamic pressures are generated and able to support an externally applied load. These bearings are also known as self-acting bearings. The importance of roughness in predicting bearing performance has gained considerable attention in Tribology [2], [5]. Hydrodynamic bearings operating at high speed are often encountered with problems of instability, known as whirl and whip. Instability may ruin not only the bearings but the machine itself. Multi lobe journal bearings maintain the stability of the bearings at higher speeds and loading conditions.

2. MULTI LOBE JOURNAL BEARINGS

Multi-lobe bearings are essentially bearings with more than one bearing pad that enable a combination of number of pads, rotation of bearing, clearance, preload, and offset. This produces a stabilizing effect on the shaft and can increase load capacity. Satisfactory dynamic characteristics are an essential requirement of a good bearing design and bearings of non-circular cross-section hold good promise for applications where bearing stiffness and stability are major considerations. In general, non-circular bearing geometry enhances shaft stability under proper conditions; this will also reduces power losses and increase oil flow, thus reducing bearing temperature [17].

Two lobe bearings are made up of two circular arcs each its own centre of curvature O1,O2 displaced a distance d from the geometric centre of the bearing O. The two lobes have 160° are each are separated by two axial 20° extensions in the horizontal direction. The three lobe bearing, consists of each lobe displaced by an equal distance, called ellipticity, from the centre of the bearing. The maximum span of a lobe is 120°. The lobes however are separated by axial grooves for admitting oil inlet and in this work 20° grooves are adopted. The net span of each lobe is 100°. The four lobe bearings, consists of four lobes. The maximum span of a lobe is 90°. The lobes however are separated by axial grooves for admitting oil and in this work 10° grooves are considered, so net span of each lobe is 80°.
For any given shaft position, lobe eccentricity ratios and attitude angles can be related with bearing eccentricity ratio, attitude angle and ellipticity ratio by simple trigonometry relations obtained.

3. ANALYSIS OF MULTI LOBE JOURNAL BEARINGS

3.1 Finite difference scheme: Steady state analysis

For steady loading and incompressible lubricants the generalized Reynolds equation becomes

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

This is a second-order non-homogenous, partial differential equation, which is difficult to solve analytically. As it stands it contains. To solve the equation numerically an attempt must first be made to reduce the number of variables to a few compact dimensionless groups [11], [14].

By using the substitutions

\[ n \text{ (power-law index)} = 1, \]
\[ \theta = \frac{x}{R}, \quad h = \frac{h}{c}, \quad \gamma = \frac{y}{\frac{2}{3}}, \quad \text{and} \quad \tilde{p} = \frac{p}{6\nu U R}. \]  

We have the dimensionless Reynolds equation

\[
\frac{\partial}{\partial \theta} \left( \tilde{h} \phi_{xx} \frac{\partial \tilde{p}}{\partial \theta} \right) + \left( \frac{D}{L} \right)^2 \frac{\partial}{\partial \gamma} \left( \tilde{h} \phi_{yy} \frac{\partial \tilde{p}}{\partial \gamma} \right) = 6 \frac{\partial \tilde{h}}{\partial \theta}
\]  

Since \( \tilde{h}=1+\varepsilon \cos \theta \) which is a function of \( (\theta, \varepsilon) \)

The boundary conditions are:

1. \( \tilde{p}(\theta, 0) = 0 \) for \( \theta = \theta_1 \) and \( \theta = \theta_2 \).

2. The pressure at the end of the bearings is assumed to be zero (atmospheric), \( \tilde{p}(\theta, \pm 1) = 0 \).

3. The pressure distribution is symmetrical about the mid-plane of bearing.

4. \( \frac{\partial \tilde{p}}{\partial \gamma}(\theta, 0) = 0 \) at \( \theta = \theta_2 \).

A developed view for half of the bearing is drawn in. the area is divided into a number of mesh sizes \( (\Delta \theta \times \Delta \gamma) \) and using central difference quotients, the equation can be written in the form as

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

Where

\( p_{i,j} \) is the pressure at any point \((i, j)\),

\( h_i \) is the film thickness at any point \((i, j)\),

\( p_{i+1,j}, p_{i-1,j}, p_{i,j+1}, \) and \( p_{i,j-1} \) are the pressures at the four adjacent points.
On simplifications the above equation becomes

\[
\begin{align*}
P(i,j) &= \left[ \phi_i \left[ P(i+1,j) + P(i-1,j) \right] + \left[ \frac{D^2}{\Delta y^2} \phi_j \left[ P(i,j+1) + P(i,j-1) \right] + \left( \frac{3\varepsilon}{2h} \right) \left[ P(i+1,j) - P(i-1,j) \Delta \phi \sin \theta \right] + \left( \frac{2\varepsilon \sin \theta}{h} \right) \Delta \theta \right] \\
&= \left( \frac{2 \phi_i + \phi_j}{D^2} \left[ \frac{\Delta y^2}{\Delta y^2} \right] \right) \\
&= \left( \frac{2 \phi_i + \phi_j}{D^2} \left[ \frac{\Delta y^2}{\Delta y^2} \right] \right)
\end{align*}
\]

It is seen that the pressure at any mesh point \((i, j)\) is expressed in terms of pressure at four adjacent points. To start the iteration process the pressure at all the mesh points are assumed and those at the boundaries are set. Then the above equation is solved for all mesh points. The equation will not be satisfied. The error at the point \((i, j)\) is

\[
(\text{Error})_{i,j} = \text{R.H.S of Eqn (3.10)} - P_{i,j}
\]

The new pressure can be computed using a successive over relaxation scheme (SOR) as

\[
(P_{i,j})_{\text{new}} = (P_{i,j})_{\text{old}} + (\text{Error}_{i,j}) \cdot \text{orf}
\]

Where ‘orf’ is the over relaxation factor.

The use of relaxation factor is the above equation accelerates the convergence of the numerical process. It is how ever very difficult to estimate the optimum value, ‘orf’ generally varies between 1.2 to 1.5.

The process will be repeated till the specified accuracy is obtained by convergence criterion as
\[
\left| \frac{\left( \sum p_{i,j} \right)_{k-1} - \left( \sum p_{i,j} \right)_k}{\left( \sum p_{i,j} \right)_k} \right| \leq \text{a very small quantity}, \quad (8)
\]

Where \( k \) is the number of iterations. The allowable error is to be kept very small of the order of small fractions of 1%. For obtaining steady state forces, generally Reynolds equation is solved with the help of finite difference scheme as described earlier, satisfying boundary conditions. The latest values of the \( \overrightarrow{P} \) are used iteratively to solve the set of equations until pressure at all mesh points converge simultaneously. Since the bearing is assumed to be symmetrical about its central plane \( (\overline{y} = 0) \).

4. RESULTS AND DISCUSSION

The steady state characteristics Sommerfeld number and attitude angle of two lobe bearing for various roughness orientations have been determined. Using Finite Difference Method, the Pressure has been determined for the multi lobe journal bearings with different surface roughness. Maximum bearing pressure was obtained for multi lobe bearings with Longitudinal surface at eccentricity= 0.4, shown in fig 3 – fig 5. Steady state characteristics are deviated from the normal results because of due to the roughness profiles. Graphs are plotted between Sommerfeld number Vs eccentricity ratio and attitude angle Vs eccentricity ratio for multi lobe bearings with longitudinal surface roughness at L/D=1.0 shown in fig. 6 and 7.

Fig 3: Pressure distribution in Two Lobe Journal bearing

Fig 4: Pressure distribution in Three Lobe Journal bearing
Fig 5: Pressure distribution in Four Lobe Journal bearing

Fig 6: Variation of Sommerfeld number with eccentricity ratio for with longitudinal surface roughness

Fig 7: Variation of Attitude angle with eccentricity ratio for with longitudinal surface roughness
The journal centre trajectories are plotted for various time steps [13]. Critical mass parameter can be obtained for above a certain value of mass parameter there is a transition region in which bearing system changes from stable to unstable.

![Diagram](image1)

**Fig 8:** Motion trajectory of the shaft centre under unidirectional constant load $\frac{L}{D} = 1$, $\varepsilon = 0.4$, for (a) Transverse, (b) isotropic and (c) longitudinal surface roughness for two lobe bearing.

![Diagram](image2)

**Fig 9:** Motion trajectory of the shaft centre under unidirectional constant load $\frac{L}{D} = 1$, $\varepsilon = 0.4$, for (a) Transverse, (b) isotropic and (c) longitudinal surface roughness for three lobe bearing.

![Diagram](image3)

**Fig 10:** Motion trajectory of the shaft centre under unidirectional constant load $\frac{L}{D} = 1$, $\varepsilon = 0.4$, for (a) Transverse, (b) isotropic and (c) longitudinal surface roughness for four lobe bearing.
5 Conclusion

The static characteristics of the finite bearings for L/D ratio = 1 was determined for three types of the surface roughness orientations. Longitudinal surface roughness had greater effect on the Sommerfeld number. The load carrying capacity is increased by 42.89% compared with the bearing having zero surface roughness. Non linear time transient analysis has been used to obtain the stability and unstability of the bearings, at different surface roughness values. The journal trajectory has been plotted for the multi lobe bearings. Critical mass parameter for a particular eccentricity ratio is found when the trajectory of the journal center ends in a cycle. It is the clear evidence that the surface roughness has the effect on the performance of characteristics of the finite bearings.

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