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Non Linear Analysis of Two lobe Journal Bearings with Surface roughness

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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 two lobe bearing is analyzed to determine the effect of surface roughness during non linear loading. A non-linear time transient analysis was performed using the fourth order runge kutta method. The effect of eccentric ratio is studied and the variation of attitude angle is discussed. The journal center trajectories were calculated and plotted. Flow factor method is used to evaluate the roughness and the finite difference method is used to predict the pressure distribution over the bearing surface.*

Keywords— ***two lobe, eccentricity ratio, surface roughness.***

I. 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. 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 [27].

II. FUNDAMENTALS OF MULTI LOBE 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. Among the non-circular bearings like elliptical and three lobe bearings are most commonly used.

A. Two lobe bearing

Two lobe bearings are made up of two circular arcs each its own centre of curvature O_1&O_2 displaced a distance d from the geometric centre of the bearing O. In the present work two lobes of 160^0 arc each are separated by two axial 20^0 extensions in the horizontal direction. In the figure 1, geometry and co-ordinate system used for the analysis of two lobe bearing are shown. 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. From simple trigonometry, following relationships can be obtained [23, 28].

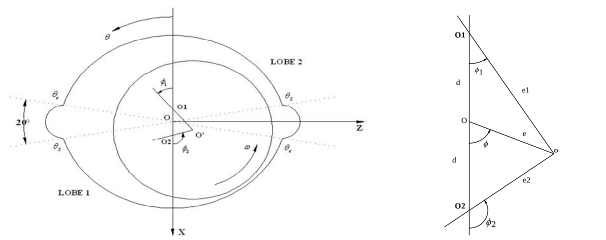

Figure 1: Geometry and trigonometry of two lobe bearing,

For Lobe 1

$$e^2 = e^2 + d^2 - 2ed\cos(\pi - \phi) \qquad (1)$$

dividing both sides by c^2, one gets,

$$\varepsilon_1^2 = \varepsilon^2 + \delta^2 + 2\varepsilon\delta\cos\phi \qquad (2)$$

or

$$\varepsilon_1 = \sqrt{\varepsilon^2 + \delta^2 + 2\varepsilon\delta\cos\phi} \qquad (3)$$

Where δ=d/c is the bearing ellipticity ratio and ε_1=e/c is the eccentricity ratio of lobe 1. Also

$$\tan\phi_1 = \frac{\varepsilon\sin\phi}{\delta + \varepsilon\cos\phi}$$

$$\phi_1 = \tan^{-1}\left(\frac{\varepsilon\sin\phi}{\delta + \varepsilon\cos\phi}\right) \qquad (4)$$

For Lobe 2,

$$e_2^2 = e^2 + d^2 - 2ed\cos\phi \qquad (5)$$

dividing both sides by c_2, one gets,

$$\varepsilon_2^2 = \varepsilon^2 + \delta^2 - 2\varepsilon\delta\cos\phi \qquad (6)$$

or

$$\varepsilon_2 = \sqrt{\varepsilon^2 + \delta^2 - 2\varepsilon\delta\cos\phi} \qquad (7)$$

Where $\varepsilon_2 = e_2/c$ is the eccentricity ratio of lobe 2.
Also

$$\tan(\pi - \phi_2) = \frac{\varepsilon \sin\phi}{\delta - \varepsilon\cos\phi}$$

$$\phi_2 = \pi - \tan^{-1}\left(\frac{\varepsilon \sin\phi}{\delta - \varepsilon\cos\phi}\right) \qquad (8)$$

The fluid film thickness for the bearing is given by
For Lobe 1,

$$h_1 = 1 + \varepsilon_1 \cos\theta \qquad (9)$$

For Lobe 2,

$$h_2 = 1 + \varepsilon_2 \cos\theta \qquad (10)$$

III. ANALYSIS OF TWO LOBE JOURNAL BEARING

In an elliptical bearing the pressure wedge created in the lower left-hand side has a very high convergence and therefore the resultant horizontal force will be high. This force is not sufficiently balanced by the forces acting to the left unless the shaft center moves upward and to the right. 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. The convergence criterion adopted for pressure calculation is $\left|1 - \frac{\sum \bar{p}_{old}}{\sum \bar{p}_{new}}\right| \le 10^{-4}$ with a chosen bearing eccentricity ratio and attitude angle picked at random, say ϕ_{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 two lobe journal bearing is Reynold equation for transient state is given by

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

where t is time, x and z are the Cartesian coordinates, μ is the absolute viscosity of the lubricant, U is th 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. However, it does not provide the vital information on minimum film thickness.

IV. RESULTS

The variation of Sommerfeld number and attitude angle of two lobe bearing for various roughness orientations having 20^0 axial grooves are determined. There is a deviation from the normal results because of the roughness profiles. Graphs are plotted between Sommerfeld number Vs eccentricity ratio and attitude angle Vs eccentricity ratio for the values of L/D=1.0 are shown in figures 4(a) to 4(f)

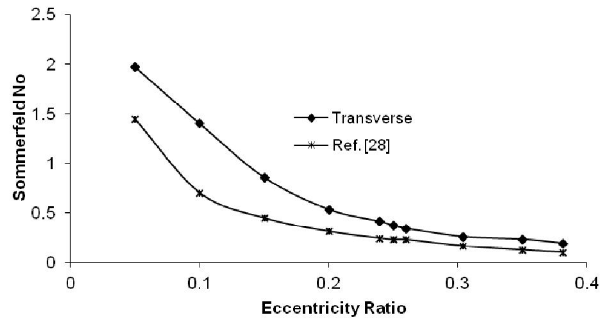

Figure 4 (a): Variation of Sommerfeld number with Eccentricity ratio for transverse roughness

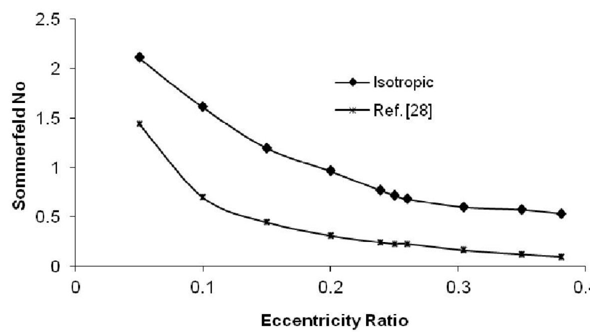

Figure 4 (b): Variation of Sommerfeld number with Eccentricity ratio for isotropic roughness

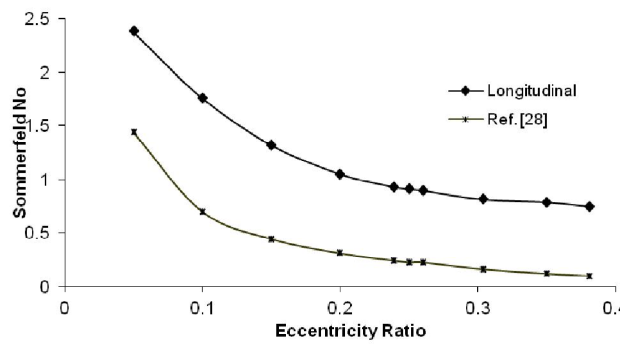

Figure 4(c): Variation of Sommerfeld number with Eccentricity ratio for longitudinal roughness

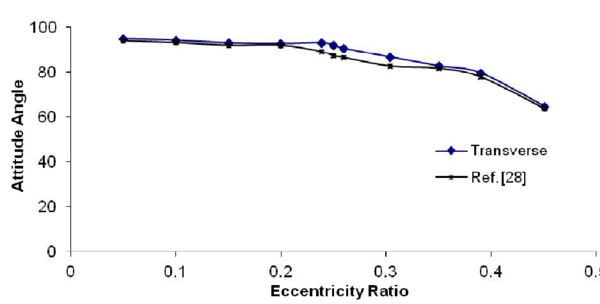

Figure 4 (d): Variation of Attitude angle with Eccentricity ratio for
Transverse roughness

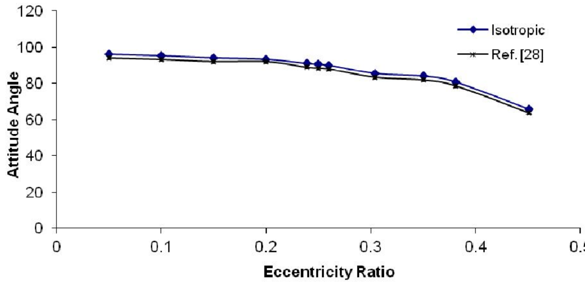

Figure 4 (e): Variation of Attitude angle with Eccentricity ratio for
Isotropic roughness

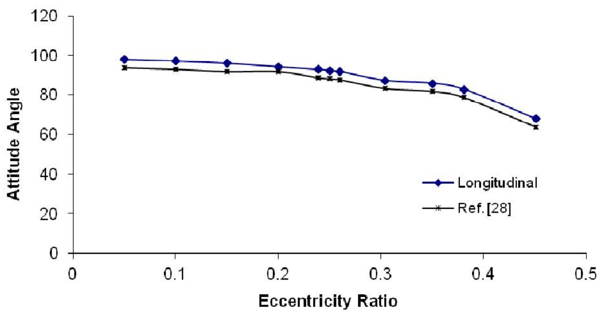

Figure 4 (f): Variation of Attitude angle with Eccentricity ratio for
Longitudinal roughness

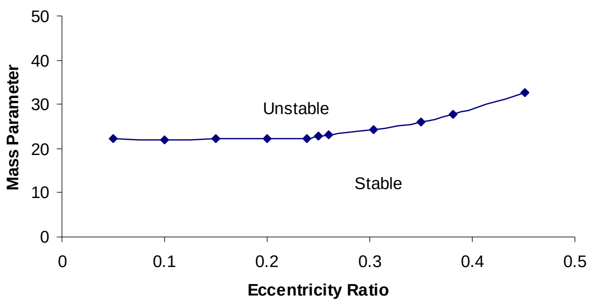

Figure 4(g): Variation of mass parameter with Eccentric ratio for two lobe journal bearing with transverse roughness at L/D ratio =1

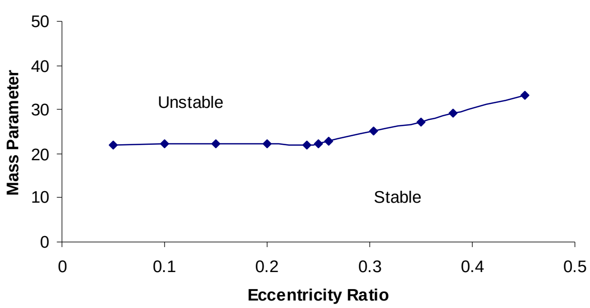

Figure 4(h): Variation of mass parameter with Eccentric ratio for two lobe journal bearing with longitudinal roughness at L/D ratio =1

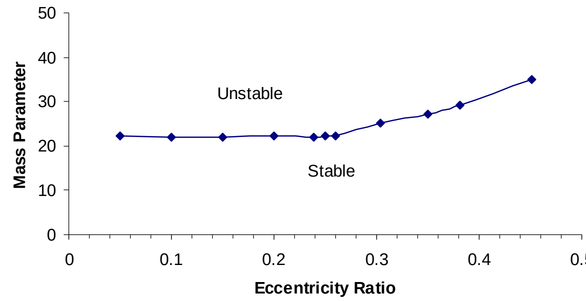

Figure 4(i): Variation of mass parameter with Eccentric ratiov for two lobe journal bearing with transverse roughness at L/D =1

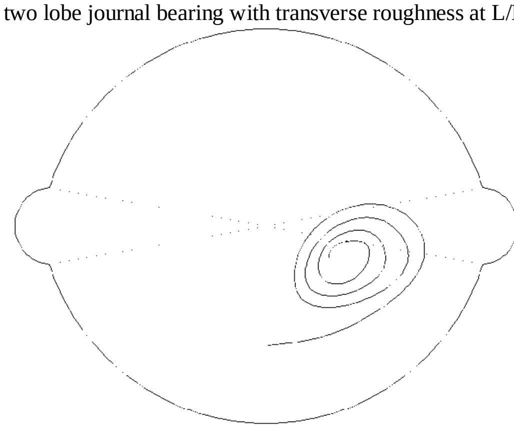

Figure 4(j): Motion trajectory of the shaft centre L/D=1, eccentric ratio=0.4 Transverse surface roughness.

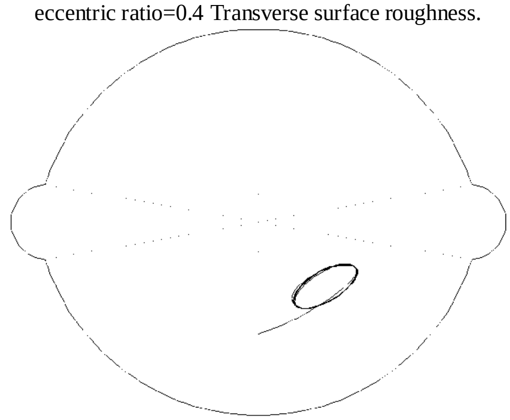

Figure 4(k): Motion trajectory of the shaft centre L/D=1, eccentric ratio=0.4 Isotropic surface roughness.

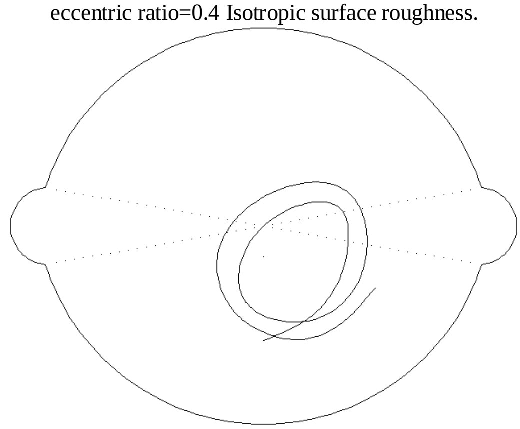

Figure 4(l): Motion trajectory of the shaft centre L/D=1, eccentric ratio=0.4 Longitudinal surface roughness.

A non-linear time transient analysis method is used to simulate the journal center trajectory and thereby to estimate the stability parameter, which is a function of speed. The journal centre trajectories are plotted for various time steps. By observing the plots we can say that the rotor system is stable, critical or at unstable state. Above a certain value of mass parameter there is a transition region in which bearing system changes from stable to unstable which is known as critical mass parameter. Critical mass parameter for a particular eccentricity ratio is found when the trajectory of the journal center ends in a cycle.

V. CONCLUSIONS

The Non linear analysis is done on the two lobe bearing for L/D ratio =1.0 with three types of the surface roughness orientations. Longitudinal surface roughness has greater effect on the Sommerfeld number. Pressure is maximum exactly midway between the lobes. These maximum pressure locations are separated by 160 degrees. Because of this, the rotor will be well balanced while rotating and vibrations due to bearing will be minimum. Sommerfeld number and eccentricity are plotted for three surface roughness orientations. The sommerfeld number decreased with increase in eccentricity ratio. It is the clear evidence that the surface roughness has the effect on the performance of static characteristics of the finite bearings. From the graphs it is observed that the calculated values are deviated from the actual values. Longitudinal surface roughness is more effective when compared with isotopic and traverse roughness.

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