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Analysis of Load Carrying Capacity in Squeeze Film of Long Spiral Hydrodynamic Journal Bearing Operating with Ferro Fluid.

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ABSTRACT: In the present paper, focuses on a theoretical study of squeeze film behavior in an infinitely long spiral journal bearing operating with ferro-fluid. Jenkins Model with uniform strength of magnetic field is applied and based on modified Reynolds equation expressions for pressure and load carrying capacity is obtained. Computed results are compared with the circular bearing.

Keywords: Non-circular Bearing, Squeeze film, Ferro Fluid, Jenkins model, uniform magnetic field.

I. INTRODUCTION

The study of a squeeze film between plates is a traditional one and it refers to the approach of one plate towards the other plate. The brake disc and diaphragm clutch are the examples for the industrial application, Fluids with strong magnetic properties have drawn considerable attention during the last decade. The term magnetic liquid (or ferrofluid) does not refer to an intrinsic ferromagnetic liquid but refers to a stable colloidal suspension of small particles of ferromagnetic materials in a carrier fluid. such suspensions behave like any liquid and act as a ferromagnetic material. For the purpose of ensuring colloidal stability, a surfactant, such as oleic acid, is usually introduced into the suspension to create, around each single particle, a coating layer to prevent the agglomeration of the particles through magnetic and molecular attraction by keeping the distance between them sufficiently large. When a magnetic field is applied to the Ferro fluid, each particle experiences a force that depends on the magnetization of the magnetic material of the particles and on the strength of the applied field.

Ferro fluids were prepared during the last decade and studied first by Neuringer and Rosenweig [1]. Tarapov[2] studied magnetic fluid lubrication of cylindrical bearing. Walker and Buckmaster[3] considered a thrust bearing, while Tipei[4] derived a pressure differential equation for magnetic fluid lubrication and applied it to study short bearings. They found that a ferrofluid lubricant increased the pressure as well as load capacity of the bearings, improved bearing stability and stiffness, and reduced wear, noise and maintenance costs. Chi et al[5] theoretically and experimentally studied a three pad Journal bearing lubricated with a ferrofluid. All the above investigations used the Neuringer and Rosenweig [1] for the lubricant flow. Sinha [6] analysed ferrofluid lubrication of cylindrical rollers with cavitation and Shukla and kumar[7] derived an equation for ferrofluid lubrication, both using the Shilomis model [8] for lubricant flow. They found that Brownian motion of the liquid together with rotation of the magnetic moments within the particles produced rotational viscosity which supported more load. Ram and Verma [9] studied a porous inclined slider bearing lubricated with a ferrofluid flowing as per the Jenkins model [10]. Using a simplification of Maugin[11] that $\overline{M} = \overline{\mu}.\overline{H}$. The above model is more realistic than the Neuringer and Rosenweig [1] model because of its considerqation of the bearing material constant, which measures the angular momentum of the fluid per unit mass and per unit field strength.

The studies of the hydrodynamic behaviors of journal bearings are based on the Reynolds equation, which was derived from Navier- Strokes and Continuity equations using many assumptions [12,13]. Few of the non circular bearings such as two lobe bearings, multi lobe bearings, lobes with pressure dams etc, are analyzed and performance characteristics are evaluated by N.P.Mehta at al [13&15]. Bouyer, J, Fillon, M, Pierre – Danos. I,[16], discussed the behavior of the hydrodynamic journal bearing with two lobes, induces high friction during transient periods, direct contact between the journal and bearing. The static performance characteristics of the elliptical bearing with different preload factor were discussed by Jayachandra Reddy and et al,[17] and the performance characteristics of the elliptical bearing under turbulent flow regime were evaluated by Hashimoto. H and et al,[18] results show the significance of preload factor in non circular bearings.

1.1: Non Circular bearings :-Hydrodynamic journal bearings have been widely used to support high-speed rotating machinery, such as turbines, compressors and pumps. In such high speed machinery, there is every possibility for the onset of whirl or instability; therefore, to overcome the instability noncircular bearings are widely used. Therefore, it is an important engineering problem to improve the operating characteristics of non-circular bearings in the high speed operating conditions for enhancing the quality of rotating machinery. One of the non-circular hydrodynamic journal bearings i.e. spiral bearing, is shown in the figure-1.

In the present analysis, our aim is to study and compare ferro-fluid squeeze film behavior in a long spiral bearing using the flow models of Jenkins with uniform magnetic field.

II. ANALYSIS AND SOLUTION

The physical Configuration of a short convergent spiral hydrodynamic journal bearing is shown in the Figure - 1.

The parametric equation of the convergent spiral is given by

$$R^* = R_1 + K\theta_R \;, \quad \text{where, K (Spiral constant)} = \frac{R_2 - R_1}{\theta_R}, \qquad \text{--------(1)}$$

In the above equation (1), if K is Positive i.e., $R_2 > R_1$ represents divergent spiral otherwise i.e., $R_1 > R_2$, represents convergent spiral. In the present paper convergent spiral journal bearing is considered and the preload factor (m) is expressed by

$$m = \frac{C_{\max} - C_{\min}}{C_{\max}} \qquad \text{------(2)}$$

$$h = C_{\max} + \left(C_{\max} - C_{\min}\right)\left(\frac{\theta}{360}\right) + e\cos\theta \qquad \text{--------(3)}$$

The film thickness form in non dimensional form, for convergent spiral is given as

$$h_C^* = 1 - m(\theta/2\pi) + \varepsilon\cos(\theta) \qquad (4)$$

The range of preload factor is $0 \le m \le 1$.

m = 0 refers to the circular bearing and the radial clearance "C" is used instead of minimum clearance $C_{\min}$. i.e. the radial clearance $C = C_{\max} = C_{\min}$.

m = 1 refers to metal to metal contact between the bearing and journal.

II.1: Boundary Conditions:

The boundary conditions for pressure distribution are (i) $P = 0$ at $\theta = 0$ and (ii) $P = 0$ at $\theta = 2\pi$. The half-Sommerfeld boundary condition leads to a more realistic prediction. However, the flow continuity at the end of pressure curve is not satisfied. The pressure becomes zero at $\theta = \pi$ and stays zero till $\theta = 2\pi$.

The Reynolds boundary conditions are (i) $P = 0$ at $\theta = 0$ (ii) $\frac{\partial p}{\partial \theta} = 0$ at $\theta = \theta_2$ and (iii) $P = 0$ at $\theta = \theta_2$. Using these assumptions many solutions were achieved for circular profiles and non circular profiles.

II.2: Relationship between preload factor (m) and eccentricity (□)

To evaluate the relationship between the eccentricity ratio and preload factor, considering the film thickness (h) is zero at θ = π, i.e. Half-Sommerfeld condition, substituting in the above equation (4),

$$\varepsilon \le 1 - m/2 \qquad \text{--------(5)}$$

i.e., the limit of eccentricity can't exceed by 0.5 for a maximum preload factor of 1.

III. Jenkins model with Uniform H

When the lubricant flows as per Jenkins model and H is uniform, the equation giving the film pressure 'P' [9-11]

$$\frac{d}{dx}\left\{\frac{h^3}{1-\frac{\rho\alpha^2\bar{\mu}H}{2\eta}}\frac{dp}{dx}\right\} = 12\,\eta\,\bar{h} \quad \text{---------} (6)$$

Where 'ρ' is the fluid density, α^2 is the material constant, $\bar{\mu}$ is the magnetic susceptibility, η is the fluid viscosity and

$$\bar{h} = \frac{dh}{dt} \quad \text{---------} (7)$$

Using equation (1) and the dimensionless quantities.

$$\bar{h} = \frac{h}{c},\ \beta = \frac{\rho\alpha^2\bar{\mu}H}{2\eta},\ \bar{p} = \frac{c^2 p}{\eta R^2 \bar{\in}},$$

where $\bar{\in} = \frac{d\in}{dt}$ -------- (8)

Equation (2) transforms to

$$\frac{d}{d\theta}\left\{\bar{h}^3\frac{d\bar{p}}{d\theta}\right\} = 12\,(1-\beta)\cos\theta \quad \text{------}(9)$$

Solving equation (5) under boundary conditions

$$\frac{d\bar{p}}{d\theta} = 0, \text{ when } \theta = \pi,\ \bar{p}(0) = 0 \quad \text{--------}(10)$$

Which ensure that $\bar{p}$ is maximum when $\theta = \pi$, the dimensionless pressure is

$$\bar{p} = \frac{6(1-\beta)}{\epsilon}\int_0^{\pi}\frac{1}{\bar{h}^2}\,d\theta \quad \text{----------}(11)$$

The load capacity W can be represented as

$$W = \left|LR\int_0^{2\pi} P\cos\theta\,d\theta\right| \quad \text{-----} (12)$$

III. RESULTS AND DISCUSSION

The following are the design parameters used in the present work.

(i). Eccentricity ratio (ε) = 0.1 -0.8;

(ii). Preload factor (m) = 0, 0.1 - 0.4 (m=0, represents Circular Bearing model) and

(iii). Material Constant (β) = 0.1 – 0.5.

From figures-2 to 4 represent the dimension less Load verses at various parameters like eccentricity ratio(ε) , preload factor (m) and Material Constant (β) of spiral bearings operating with ferro- fluids. Figur-2 represents the squeeze load vs eccentricity ratio at different material constant for both circular bearing(m=0.0) and spiral bearing(m=0.3). It is observed that eccentricity ratio increases the squeeze load also increases for both the circular and spiral bearings. Figure-3 denotes the squeeze load vs different material constant at fixed eccentricity ratio(e=0.3) for both circular and spiral bearings. It is observed that as the material constant increases the load carrying capacity decreases for both the bearings. This is due to the increase in the strength of the magnetic field. Figur-4 gives the squeeze load vs preload factor(m) at constant eccentricity ratio and different material constant. It is observed that as the preload factor increases the load carrying capacity increases. This is due to the narrow gap between bush and bearing.

IV. CONCLUSIONS

Ferro-fluid squeeze film behavior in long journal spiral bearing is studied using Jenkins model of uniform strength. Load carrying capacity of the spiral bearing is more than the circular bearing. Bothe the Eccentricity ratio and preload factors increase the load capacity. Bearing material constant decreases the load carrying capacity.

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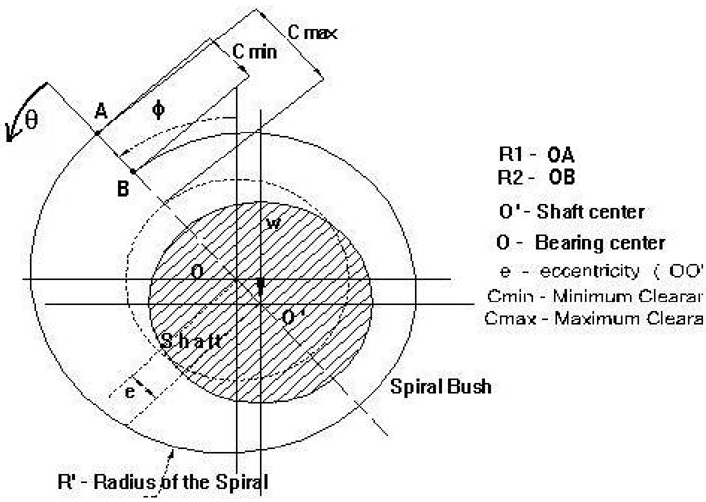

R = Shaft Radius B = Length of the Bearin
O_B = Bearing Center O_J = Journal Center
e = Eccentricity h = Fluid Film Thickness.

Figure - 1: Physical configuration of Convergent Spiral Squeeze film Bearing..

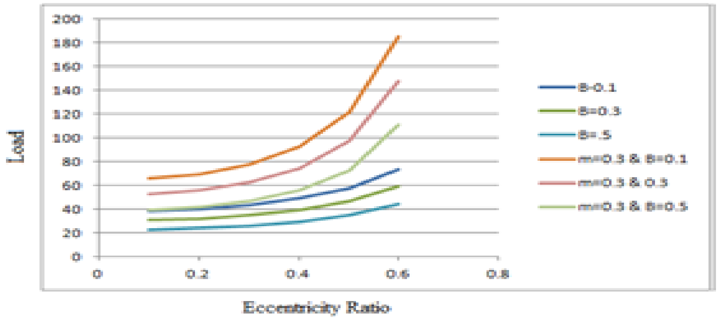

Figure-2: Squeeze load vs Eccentricity ratio at different Material Constant and Preload factor.

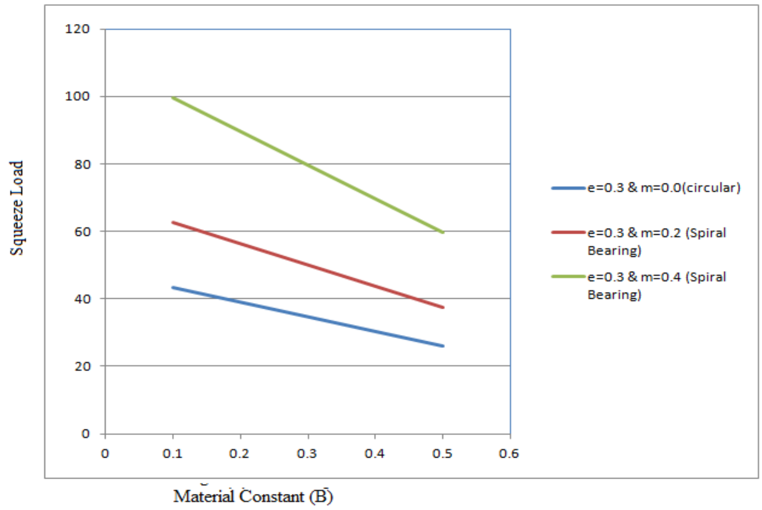

Figure-3: Squeeze Load vs Material Constant at constant Eccentricity ratio for both Circular and Spiral Bearings.

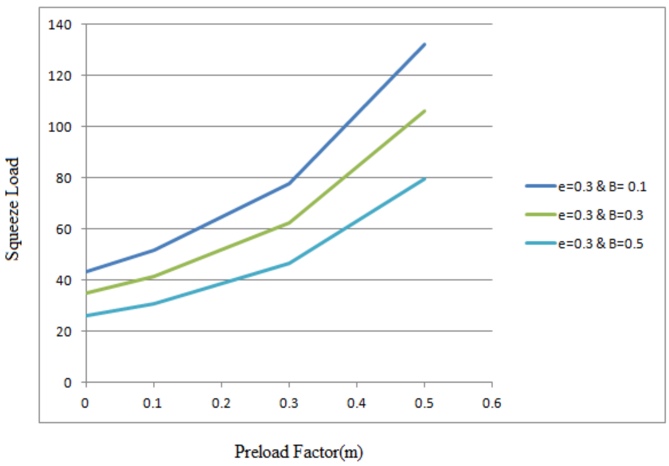

Figure-4: Squeeze Load vs Preload Factor(m) at constant Eccentricity Ratio for different Material Constant.