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Study of Variation of Groove Angle on Performance Characteristics of Two-Axial Groove Journal Bearing

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Abstract The present work deals with the study of variation of groove angle on the performance characteristics of a two- axial groove journal bearing. Two grooves are sited axially at ±90º to the load line. Different groove angles have been considered in the present work. The effect of varying groove angle on the performance characteristics is studied. The side flow, bush temperature, mean journal temperature and attitude angle having large variation as groove angle changes. Whereas pressure, minimum thickness of fluid film and eccentricity ratio has very small variation. Smaller groove angle is recommended for performance characteristics of two-axial groove journal bearing.

Keywords: *groove angle, journal bearings, Reynolds equation, two-axial-groove, thermohydrodynamic analysis, viscosity-temperature equation*

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1. Introduction

A journal bearing may fail due to the minimum film thickness between journal and bush. This causes bush and journal contact due this temperature of bush or journal and fluid rises resulting in drastic reduction of fluid viscosity and fluid film thickness. Due to rise in temperature of fluid, viscosity of lubricant decreases. To avoid this, friction between bush and journal must minimum. An accurate thermohydrodynamic analysis is required to get the thermal response of the fluid and bush. Large amount of work have been done on thermal analysis on plain journal bearing. Therefore, a need has been felt to carry out further investigations on the thermal analysis of two-axial groove journal bearing. The effect of varying groove angle on the performance characteristics of journal bearing has been studied.

A large number of investigations on thermal effects in journal bearing have been reported in literature. Allan T. [1] utilized the variation calculus approach and shows how the appropriate matrix equations were derived. Finite element method used for solving the Reynolds equation. Dowson D. and Ashton J. N. [2] considered the thermal effect and computed a solution of Reynolds equation for plain journal bearing configuration. Operating characteristics were evaluated from the computed solutions and results were presented graphically. Ferron J. et al. [3] performed theoretical and experimental thermohydrodynamic analysis of finite length journal bearing. They solved three dimensional energy, heat conduction equations and Reynolds equation simultaneously. They computed mixing temperature by performing a simple energy balance of re-circulating and supply oil at the inlet.

Soni S. C. et al. [4] studied the performance characteristics of a finite two-lobe hydrodynamic journal bearing in terms of the load-carrying capacity, the stiffness and damping coefficients and the non-dimensional critical journal mass at various eccentricity ratios for Reynolds' numbers up to 12000. The computed results based on the lineariszed theory and those obtained using the non-linear theory were compared.

Heshmat H. and Pinkus O. [5] studied mechanism of fluid flow and temperature in groove and recommended empirical relations. This implies that no mixing of fluid occurs inside the grooves. Gethin D.T. and M.K.I. El Deihi [6] presented an analysis based on the finite-element method for the incompressible hydrodynamic lubrication of a cylindrical-bore bearing subjected to different loading directions. The model accounts fully for the extent of the lubricant film in both load-carrying and ruptured parts of the bearing. They considered number of loading directions. Results when computed and show that load-carrying ability, hydrodynamic flow and attitude angle all depend on loading direction for twin-axial groove cylindrical-bore bearing.

Khonsari M. M. and Beaman J. J. [7] considered re-circulating fluid and supply oil in this analysis. They presented thermo hydrodynamic effects in journal bearing operating with axial groove under steady-state loading. Sinhasan R. and Chandrawat H. N. [8] presented a study of the thermoelastohydrodynamic effects in journal bearings. The finite element method was used to solve the governing equations. The conventional three-dimensional energy equation was modified to account for the flow of lubricant in the cavitation region. The journal temperature was established by satisfying the condition of thermal equilibrium of the journal. The performance characteristics and stability margins of a two-axial groove journal bearing were studied.

Gethin D. T. and El-Delhi M. K. I. [9] presented a comparison between predicted and measured performance trends. The analysis confirms the sensitivity of global bearing performance with respect to loading direction and the comparison with experiment suggests that the numerical model was adequate unless thermal operating conditions were extreme.

Nagaraju Y. et al. [10] presented the thermal effects on the static and dynamic performance characteristics of an elliptical journal bearing. The solutions of the lubricant flow and thermal equations were obtained using the finite element method and a direct iteration scheme. The Reynolds boundary condition at the trailing edge of the fluid film in each lobe of the elliptical bearing and the equilibrium locus of the bearing centre position for vertical load was established by an iteration scheme. The static characteristics in terms of the load capacity, attitude angle, end leakage and friction parameter and the dynamic characteristics in terms of critical mass, threshold speed and damped frequency of whirl were studied.

Banwait S. S. and Chandrawat H. N. [11] proposed a non-uniform inlet temperature profiles for accurate simulation. They considered heat transfer from the outer edge of the bush to fluid in the supply groove and concluded that supply groove position plays an important role in the performance of the journal bearing. Costa L. et al. [12] presented extensive experimental results of the thermohydrodynamic behavior of a single axial groove journal bearing. They studied the influence of fluid supply condition in groove. Bearing performance parameters are affected by fluid supply conditions. The effect of supply pressure on minimum fluid film thickness is dependent on location of groove. They concluded that an axial groove located in positive angle from load line in the direction of shaft rotation will reduce maximum temperature and pressure.

Kosasih P. B. and. Tieu A. K [13] considered the flow field inside the supply region of different configuration and thermal mixing around the mixing zone above the supply region for different supply conditions. Flows in the thermal mixing zone of a journal bearing were investigated using the computational fluid dynamics. The complexity and inertial effect of the flow inside the supply region of different configurations were considered.

Jeddi L. et al. [14] outlined a new numerical analysis which was based on the coupling of the continuity. This model allows to determine the effects of the feeding pressure and the runner velocity on the thermohydrodynamic behavior of the lubricant in the groove of hydrodynamic journal bearing and to emphasize the dominant phenomena in the feeding process. Brito F.P. et al. [15] carried out an experimental study of the influence of oil supply temperature and supply pressure on the performance of a 100 mm plain journal bearing with two-axial grooves. The grooves were located at $\pm 90^0$ to the load line. They show that the bearing performance is strongly dependent on the supply conditions. Temperature of supplied fluid has a strong effect on the minimum film thickness. Rise in supply pressure gives rise in oil flow rate. But it has little effect on the maximum temperature and power-loss, except in the case of the lightly-loaded bearing.

Banwait S. S. [16] presented a comparative critical analysis of static performance characteristics along with the stability parameters and temperature profiles of a misaligned non-circular two and three lobe journal bearings operating under thermohydrodynamic lubrication condition. The analysis developed embraces bi-planar journal misalignment of any degree, up to bearing-journal contact.Brito F. P. et al. [17] carried out an experimental study of the influence of oil supply temperature and supply pressure on the performance of a plain journal bearing with two-axial grooves located at ±90º to the load line. Current theoretical models still do not give good estimates of the temperature evolution at the unloaded lobe of the bearing. It was anticipated that a better understanding of the combined effect of the two grooves on the temperature field will improve theoretical modeling, thereby allowing better predictions to be made about the thermal behavior of bearings.

Singh U. et al. [18] theoretically performed a steady-state thermohydrodynamic analysis of an axial groove journal bearing in which oil was supplied at constant pressure. Roy L. [19] obtained steady state thermohydrodynamic analysis and its comparison at five different feeding locations of an axially grooved oil journal bearing. Reynolds equation was solved simultaneously along with the energy equation and heat conduction equation in bush and shaft.

Maneshian B. and Gandjalikhan Nassab S. A. [20] determined thermohydrodynamic characteristics of journal bearing with turbulent flow using computational fluid dynamic technique. Arab Solghar A. [21] describes a thorough investigation on the influence of loading direction on twin groove journal bearings including an assessment of the flow rate distribution through each one of the grooves and its influence on its thermal behavior.

Roy Lintu and. Kakoty S. K [22] presents various arrangements of grooving location of two-groove oil journal bearing for optimum performance. They considered 10^0, 20^0 and 30^0 groove angles to obtain an optimum configuration of the two grooves positions around the circumference of the hydrodynamic journal bearing for maximum oil flow, minimum friction loss, maximum load bearing capacity, and maximum critical speed vis-`a-vis mass parameter, a function of speed. Brito F. P. et al. [23] discussed an extensive parametric study to assess the influence of lubricant feeding conditions. They carried out, feeding pressure, temperature, groove length, groove width ratio and number of grooves on bearing performance.

Kadam Kanifnath et al. [24] predicted the temperature and pressure distributions in the fluid film of a journal bearing using a non-dimensional viscosity-temperature equation. Kadam K. R. et al. [25] studied the influence of non dimensional modified viscosity-temperature equation on thermohydrodynamic analysis of plain journal bearing.

The pressure and temperature distribution in the journal bearing which was almost equal to the temperature obtained by experimental results of Brito F. P. et al. [23]. The results have been validated by comparison with experimental results of Brito F. P. [23] which show good agreement.

The aim of this work is to study the performance characteristics of two-axial groove journal bearing for different groove angles. Thermohydrodynamic performance characteristics of two-axial groove journal bearing using different groove angle have not been reported so far. This paper presents the analysis of static performance characteristics and temperature profiles of a two- axial groove journal bearings at seven different groove angles.

2. Analysis

The mathematical model developed for thermohydrodynamic analysis is described and validated by the authors in their earlier work [24,25]. In the present work, three dimensional energy equation, heat conduction and Reynolds equation were considered for analysis of thermohydrodynamic analysis of a two-axial groove journal bearing. The model was based on the simultaneous numerical solution of the generalized Reynolds and three dimensional energy equations within the fluid-film and the heat transfer within the bush body. The equations used in the analysis along with the simplifying assumptions and boundary conditions are listed in the Appendix I.

2.1. Viscosity-Temperature Equation

The viscosity of fluid film is extremely sensitive to the operating temperature of the journal bearing. With increasing temperature, the viscosity of fluid decreases. From the engineering viewpoint, it is important to know the viscosity value at the operating temperature since it determines the fluid film thickness separating the two surfaces of the journal bearing.

Viscosity and temperature were allowed to vary along the thickness of the journal bearing. The viscosity-temperature relation given by Ferron J. et al. [3] were suitably modified and used in the present model. The viscosity of the lubricant was assumed to vary across the fluid film and around the circumference. The variation of viscosity with temperature in the non-dimensional two degree equation as described by Ferron J. et al. [3] is follows:

$$\bar{\mu} = \frac{\mu}{\mu_0} = k_0 - k_1 \overline{T_f} + k_2 \overline{T_f}^2 \qquad (1)$$

The said two degree viscosity-temperature equation has been modified and developed into three degree polynomial viscosity-temperature equation. This modified viscosity-temperature equation as illustrated below has been used in the present analysis:

$$\bar{\mu} = \frac{\mu}{\mu_0} = k_0 - k_1 \overline{T_f} + k_2 \overline{T_f}^2 - k_3 \overline{T_f}^3 \qquad (2)$$

In equation (1) Ferron J. et al. [3] used the viscosity coefficients as, k_0 = 3.287, k_1 = 3.064, k_2 = 0.777 while the authors considered the following modified viscosity coefficients, k_0 = 3.1286, k_1 = 2.4817, k_2 = 1.1605 and k_3=0.31226 in the present work. The polynomial equation was found to provide improved results. The results obtained from viscosity-temperature equation developed by authors' model gives good results when compared with the experimental results of Ferron J. et al. [3]. The temperature distribution in an aligned plain journal bearing shows very slight variation between temperature and pressure in journal bearing obtained by authors and those obtained by Ferron J. et al. [3].

2.2. Variation of Groove Angle

The effect of variation in groove angle on the performance characteristics of two-axial journal bearing has been studied. Two-axial groove journal bearing geometry considered groove angles ranging from 10^0 to 40^0 in the interval of 10^0 as shown in Figure 1.

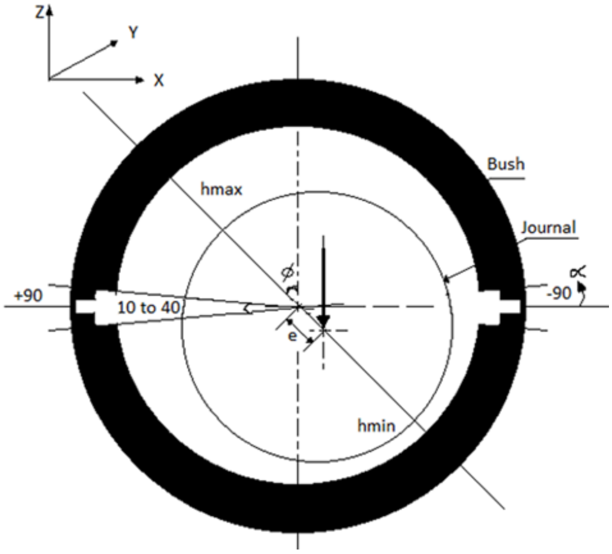

Figure 1. Two-axial groove journal bearing geometry

In almost all bearings, a hole and groove are cut into the bush. It is not possible for the bearing to rotate at any significant speed without some flow through the groove. For design purposes it is necessary to calculate the flow of lubricant through the groove. A groove is provided for distribution of lubricant over a bearing length.

The non dimensional heat input to journal $\bar{q}$ considered is given by,

$$\bar{q} = -\frac{k_f}{\bar{c}\,k_j}\left[\int_0^{2\pi} \frac{1}{\bar{h}}\left(\frac{\partial \overline{T_f}}{\partial \bar{z}}\right) d\alpha\right] - \frac{\bar{c}\,h_{fj}\,R}{k_j}\left(\overline{T_j} - \overline{T_m}\right)\alpha_G \qquad (3)$$

Where α_G is the groove angle in radians. In the above equation, the second term represents the heat transfer between the journal and the lubricant in the oil groove.

3. Solution Procedure

For solution of thermohydrodynamic lubrication of two-axial groove journal bearing a nested iteration scheme was developed. The solution of Reynolds equation requires the viscosity of fluid, which is obtained from generalized energy equation. Reynolds equation was solved using finite element method for determining the pressure in the fluid-film by iterative technique. The negative pressure nodes were set to zero and altitude angle was modified till convergence was achieved. A pressure and temperature field for the initial eccentricity ratio was recognized. Values of the fluid film velocity components were calculated in circumferential, axial and radial directions. Coefficient of contraction of fluid-film was determined. Coefficient of contraction was assumed as unity in positive pressure region. The three dimensional energy equation was solved using finite difference method for obtaining temperature in the fluid film and bush housing.

For a trial eccentricity ratio, the temperature fields for the lubricant, bush housing and the journal were established for the converged thermohydrodynamic pressure. The three dimensional energy equation and Fourier heat conductions were simultaneously solved. The journal temperature was updated after obtaining the converged temperature fields for the fluid and the bush and housing. The energy and Fourier conduction equations were again simultaneously solved with the modified journal temperature. All the above steps were repeated until convergence was achieved. A solution scheme shown in Figure 2 also explains briefly the procedure adopted for solution.

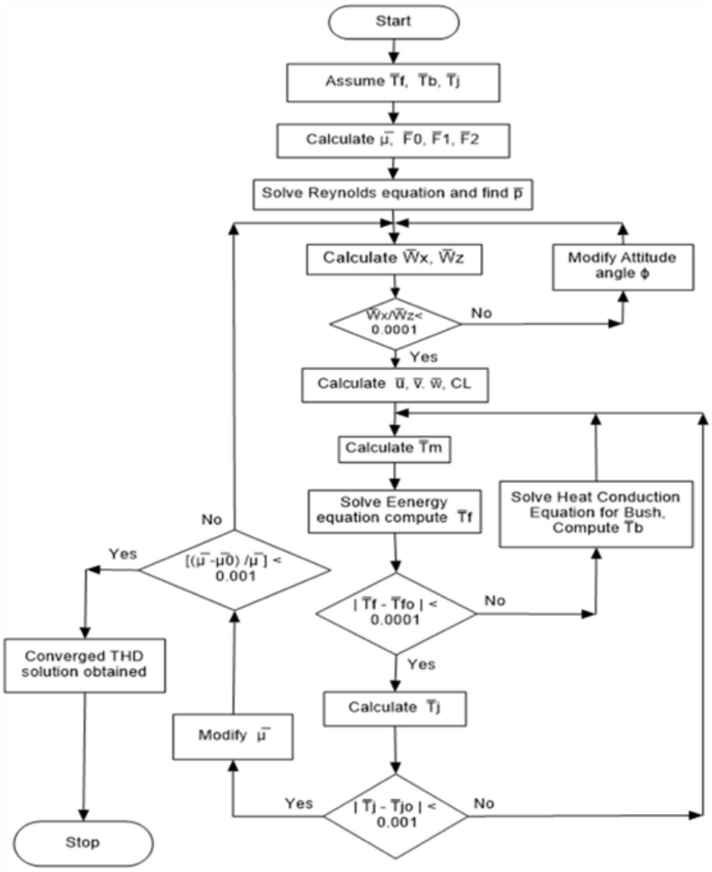

Figure 2. Solution Scheme

Table 1. Bearing dimensions, operating conditions and lubricant properties

Number of Nodes in one element	Node	4	
Outer radius bush	R_2	0.1	m
Radius of journal	R	0.05	m
Length of bush	L	0.08	m
Length to diameter ratio	L/D	0.8	
Groove angle	α_G	10, 20, 30, and 40	^{0}
Attitude angle	Φ	52	^{0}
Radial clearance	$\bar{c}$	0.0029	
Thermal conductivity of fluid	k_f	0.13	W/m °C
Thermal conductivity of bush housing	k_b	50	W/m °C
Thermal conductivity of journal	k_j	50	W/m °C
Convective heat transfer coefficient of bush	h_{ab}	50	W/m^2 °C
Convective heat transfer coefficient of journal	h_{aj}	50	W/m^2 °C
Convective heat transfer coefficient of bush housing from solid to fluid	h_{fb}	1500	W/m^2 °C
Specific heat of lubricant	C_p	2000	J/kg °C
Density of lubricant	P	860	kg/m^3
Viscosity of lubricant at 40°C	μ	0.0277	N-s/m^2
Journal Speed	N	1000, 2000, 3000. 4000 And 6000	rpm
Reference temperature of lubricant	T_r	40	°C
Ambient temperature of lubricant	T_a	40	°C
Supply temperature of lubricant	T_s	40	°C

4. Results and Discussion

The variation of groove angle on performance characteristics of two-axial groove journal bearing have been found out. One dimensional heat conduction equation was used for temperature distribution in journal. The journal temperature was revised after obtaining the converged temperature for fluid and bush. The energy and Fourier conduction equations were simultaneously solved with revised journal temperature. All the above steps were repeated until the convergence was achieved. Numerical calculations were performed by writing a computer code in C.

Non dimensional governing equations were discretized for numerical solution. The global iterative scheme was used. A mesh discretization for fluid film and bush with, 10 nodes in the axial direction, 34 nodes in the circumferential direction, and 10 nodes across the film thickness and 10 nodes across the radius of bush thickness were used. Finite Element Method for generalized Reynolds equation and Finite Difference Method for energy equation were used for solving equations

Table 1 depicts the input parameters used for analysis of two-axial groove journal bearing. The present analysis assumes aligned two-axial groove journal bearing.

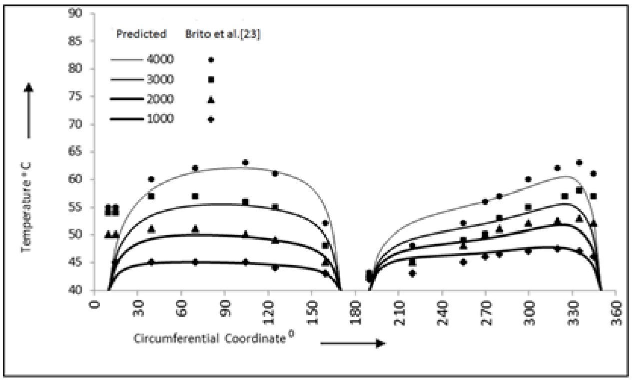

Figure 3. Comparison between the present analysis and experimental temperature profiles Brito F. P. et al. [23] at the mid-plane of the inner bush surface of the bearing for 6000N at 1000, 2000, 3000 and 4000 rpm speeds

Temperature of fluid equal to temperature of bush at the fluid–bush interface was considered for the parabolic distribution for calculating the values of coefficient. Journal temperature was equal to the fluid–journal interface. The condition of mixing the recirculating fluid with the supply fluid has also been included. For performance characteristics of two-axial groove journal bearing at two different speeds and loads variation from 1kN to 10kN have been carried out for varying groove angles.

Figure 3 predicts the circumferential temperature distribution in the mid-plane of fluid-bush interface for two-axial grooves. The authors compared their theoretical results with the published results of Brito F. P. et al. [23]. Theoretical predictions and results published by Brito F. P. et al. [23] exhibit a similar pattern, the predicted maximum temperature value and their locations are reasonably very close to the measured values of Brito F. P. et al. [23].

The performance of attitude angle on two-axial journal bearing has been carried out for different groove angle as depicted in Figure 4.

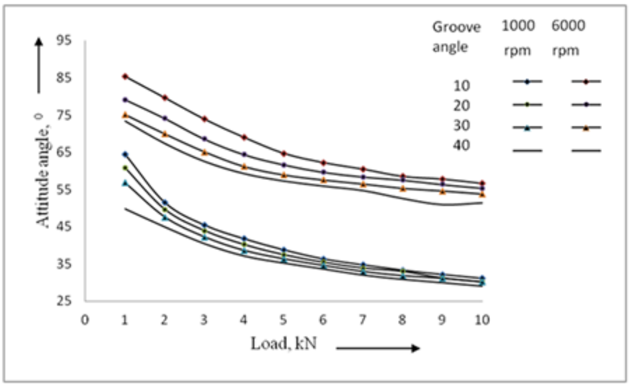

Figure 4. Comparison between attitude angle for different groove angles at different loads and at 1000 rpm and 6000 rpm

At smaller load values, it is observed that the attitude angle decreases rapidly. At 1kN, there is a drastic reduction observed in attitude angle almost to the extent of 29.32% and at higher speed there is 16.25% at lower load. As groove angle increases the variation of attitude angle decreases to 7.38% with increase in the load values. The maximum variation of attitude angle is observed at lower rpm of journal and the minimum is observed at higher load values (10kN). From Figure 5 it is clear that as the load increases the minimum fluid film thickness decreases and effect of variation of groove angle is very less

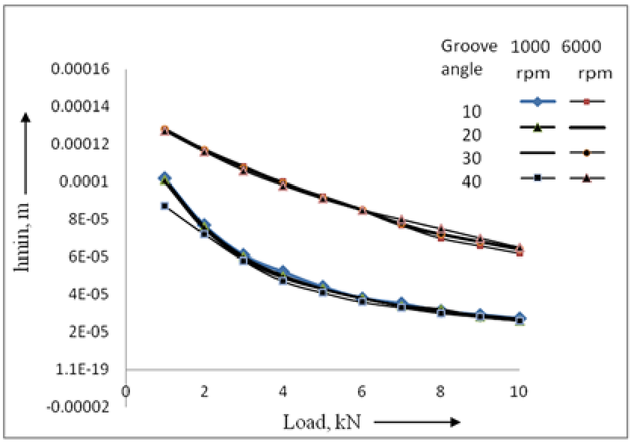

Figure 5. Comparison between minimum film thickness for different groove angles at different loads and at 1000 rpm and 6000 rpm

It has been observed that the oil groove angle has very a smaller amount effect on eccentricity ratio also. Eccentricity ratio decreases as the groove angle increases at higher speed shown in Figure 6.

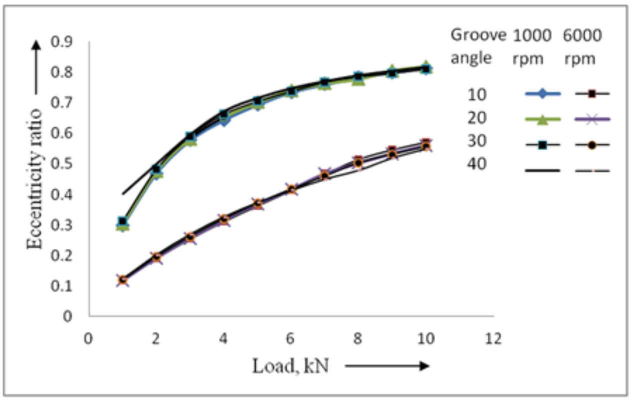

Figure 6. Comparison between eccentricity ratio for different groove angles at different loads and at 1000 rpm and 6000 rpm

At higher speed and lower load value, it is observed that the maximum bush temperature decreases. At large load value and higher speed there a variation in bush temperature is 3.032%. With increase in load value and with the corresponding increase in groove angle at higher speed, the variation observed in bush temperature is from 2.051% to 3.032%. There is a very small variation in bush temperature at lower speed shown in Figure 7.

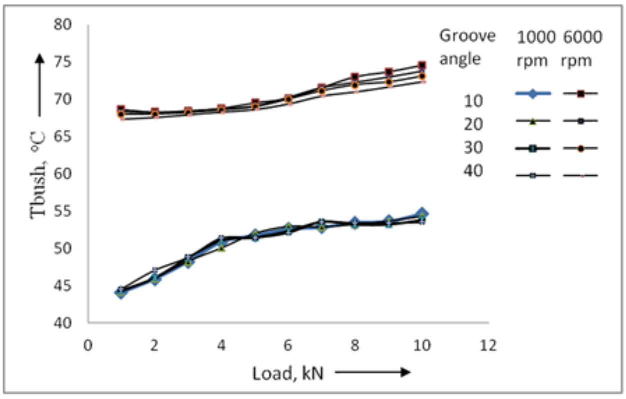

Figure 7. Comparison between bush temperature for different groove angles at different loads and at 1000 rpm and 6000 rpm

At lower speed, it is observed that there is maximum rise in the temperature of bush from 1kN to 5kN. From 5kN onwards rise in bush temperature is very small. At higher speed there is very less increase in bush temperature.

As depicted in Figure 8 the temperature of journal increases as the groove angle increases. Temperature of journal varies from 3.53% to 3.72% at lower speed with increase in the groove angle. At higher speed the variation is 4.77% at 4kN load and 6000rpm speed of journal. The maximum variation is 4.97% in journal temperature is observed at 1000 rpm and 4kN load values.

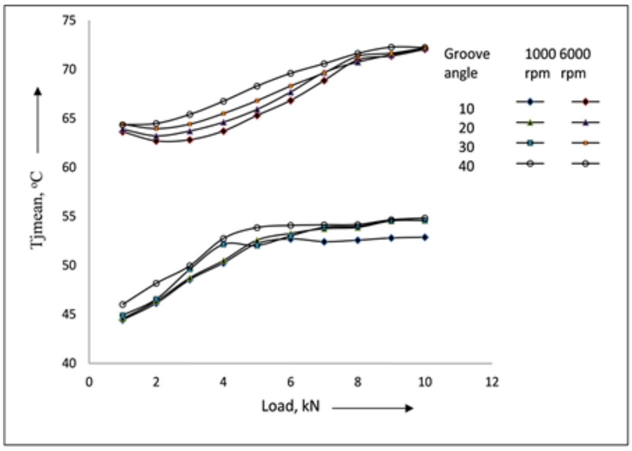

Figure 8. Comparison between mean journal temperature for different groove angles at different loads and at 1000 rpm and 6000 rpm

The maximum pressure values increases due to increase in groove angle. The pressure of fluid is maximum at lower speed and higher at smaller speed as depicted in Figure 9.

There is maximum variation 10.97% at lower speed and lower load whereas 3.07% variation at higher load and higher speed. The variation in pressure at 6000 rpm is 4.56 % at lower load and 10.85% at higher load values.

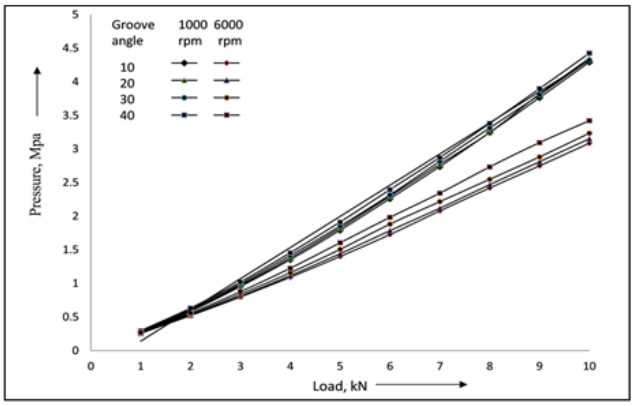

Figure 9. Comparison between for maximum pressure for different groove angles at different loads and at 1000 rpm and 6000 rpm

As the groove angle increases, it is observed that the side flow increases with increases in load values as shown in Figure 10.

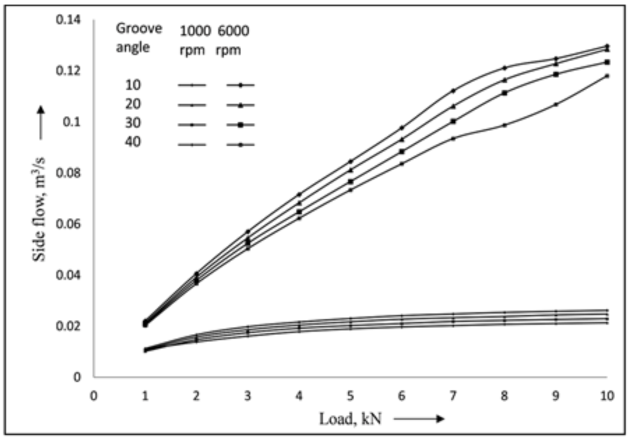

Figure 10. Comparison between side flow for different groove angles at different loads and at 1000 rpm and 6000 rpm

At larger load value, there is maximum reduction observed in the side flow almost to the extent of 22.82%. With increase in load value and corresponding increase in groove angle, the variation in side flow increases from 2.83% to 22.82%. The maximum variation of side flow is observed at lower speed of journal and for maximum load value. Beyond 6kN load the variation in the side flow is very less at 1 1000 rpm speed. At 8 kN load and higher speed the maximum variation observed is to the extent of 22.65%.

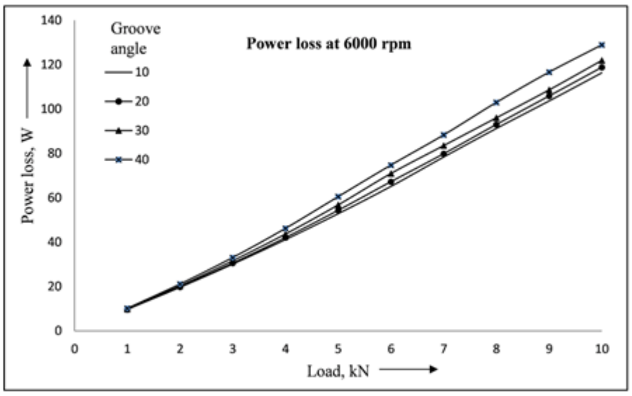

Figure 11. Comparison between power loss for different groove angles at 10kN load and at 6000 rpm

Figure 11 clearly brings out the disadvantage of increasing the groove angle. The variation in power loss is maximum at higher load and higher speed of jounal. It shows that with the increase in groove angles and the load value the power is very high as compare to lower groove angle, hence it can be easily concluded that the lower groove values are recommended for two-axial groove journal bearing.

5. Conclusions

The variation in groove angle from on performance characteristics of two-axial groove journal bearing has been studied. On the basis of results and discussions presented in above sections, following major conclusions are drawn:

- At smaller load and low speed value, there is a drastic reduction observed in attitude angle almost to the extent of 29.32% and at higher speed there is 16.25% at lower load. The variation decreases with the increase in the groove angle.
- The maximum variation of attitude angle is observed at lower rpm of journal and the minimum is observed at higher load values.
- The oil groove angle has very less effect on minimum fluid film thickness and eccentricity ratio.
- The performance characteristics drastically affected due to the variation in groove angles on attitude angle, bush temperature, journal temperature, pressure of fluid and side flow values. There are some changes observed in case of higher bush temperature and high journal temperature and side flow.
- The maximum variation of side flow is observed at lower speed of journal and for maximum load value. There is small variation of side flow at higher speed and larger load values.
- At large load value, there is a reduction observed in the bush temperature to the extent of 3.032%. With increase in load value and with the corresponding increase in groove angle, the variation observed in bush temperature is from 2.051% to 3.032%.
- There is an increase in journal temperature to the extent of 3.72% with increase in the groove angle.
- There is maximum variation in pressure 10.97% at lower speed and lower load values, whereas 3.07% variation at higher load and higher speed. The variation in pressure at larger speed is 4.56 % at lower load and 10.85% at higher load values.
- This study recommends lower value of groove angle.

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Appendix I

A. Reynolds Equation

A Reynolds equation in the following dimensionless form governs the flow of incompressible isoviscous fluid in the clearance space of a journal bearing system. This equation in the Cartesian coordinate system is written as,

$$\frac{\partial}{\partial\alpha}\left(\bar{h}^3\,\bar{F}_2\,\frac{\partial\bar{p}}{\partial\alpha}\right)+\frac{\partial}{\partial\beta}\left(\bar{h}^3\,\bar{F}_2\,\frac{\partial\bar{p}}{\partial\beta}\right) = \frac{\partial}{\partial\alpha}\left(\bar{h}-\bar{h}\,\frac{\bar{F}_1}{\bar{F}_0}\right)+\frac{\partial\bar{h}}{\partial\bar{t}} \quad (1)$$

where the non-dimensional functions of viscosity $\overline{F_0}$, $\overline{F_1}$ and $\overline{F_2}$ are defined, by,

$$\bar{F}_0=\int_0^1\frac{d\bar{z}}{\bar{\mu}};\ \bar{F}_1=\int_0^1\frac{\bar{z}}{\bar{\mu}}\,d\bar{z};\ and\ \bar{F}_2=\int_0^1\frac{\bar{z}}{\bar{\mu}}\left[\bar{z}-\frac{\bar{F}_1}{\bar{F}_0}\right]d\bar{z} \quad (2)$$

The non-dimensional functions of viscosity $\overline{F_0}$, $\overline{F_1}$ and $\overline{F_2}$ report for the effect of variation in fluid viscosity across the film thickness. Non dimensional minimum film thickness is given by,

$$\bar{h}=1-\bar{X}_j\cos\alpha-\bar{Z}_j\sin\alpha \quad (3)$$

In this work a finite element method is used to solve Eq. (1). The techniques of reducing differential equations to a set of algebraic equations. The above equation (1) was solved to satisfy the following boundary and complementary conditions:

i. On the bearing side boundaries

$$(\beta=\pm\lambda),\quad \bar{p}=0 \quad (4)$$

ii. On the supply groove boundaries

$$\bar{p}=\overline{p_s} \quad (5)$$

iii. In the positive pressure region, Positive pressures will be generated only when the film thickness is thin,

$$\bar{Q}=0,\quad \bar{p}>0 \quad (6)$$

iv. In the cavitated region

$$\bar{Q}<0,\ \bar{p}=0,\ \frac{\partial\bar{p}}{\partial\alpha}=0 \quad (7)$$

Solution of Eq. (1) gives pressure at each node with above boundary and complementary conditions.

B. Three Dimensional Energy Equation

Three dimensional energy equation is used to determine the temperature distribution in journal bearing.

$$\bar{h}^2\left(\bar{u}\,\frac{\partial\overline{T_f}}{\partial\alpha}+\bar{v}\,\frac{\partial\overline{T_f}}{\partial\beta}+\frac{1}{\bar{h}}\,(\bar{w}-\bar{z}\,\bar{u}\,\frac{\partial\bar{h}}{\partial\alpha})\frac{\partial\overline{T_f}}{\partial\bar{z}}\right) = \overline{D_e}\,\bar{\mu}\left[\left(\frac{\partial\bar{u}}{\partial\bar{z}}\right)^2+\left(\frac{\partial\bar{v}}{\partial\bar{z}}\right)^2\right]+\overline{P_e}\,\frac{\partial^2\overline{T_f}}{\partial\bar{z}^2} \quad (8)$$

Peclect number ($\overline{P_e}$) and Dissipation number ($\overline{D_e}$) are in non-dimensional form, as follows:

$$\overline{P_e}=\frac{k_f}{\left(C_p\ \rho\ \omega_j\ c^2\right)},\ \overline{D_e}=\frac{\bar{\mu}\,\omega_j}{\left(C_p\,\rho\,T_r\ c^2\right)} \quad (9)$$

Values of the non-dimensional velocity components in circumferential and axial direction are as follow:

$$\bar{u}=\bar{h}^2\,\frac{\partial\bar{p}}{\partial\alpha}\left[\int_0^{\bar{z}}\frac{\bar{z}}{\bar{\mu}}\,d\bar{z}-\frac{\bar{F}_1}{\bar{F}_0}\int_0^{\bar{z}}\frac{d\bar{z}}{\bar{\mu}}\right]+\frac{1}{\bar{F}_0}\int_0^{\bar{z}}\frac{d\bar{z}}{\bar{\mu}} \quad (10)$$

$$\bar{v}=\bar{h}^2\,\frac{\partial\bar{p}}{\partial\beta}\left[\int_0^{\bar{z}}\frac{\bar{z}}{\bar{\mu}}\,d\bar{z}-\frac{\bar{F}_1}{\bar{F}_0}\int_0^{\bar{z}}\frac{d\bar{z}}{\bar{\mu}}\right] \quad (11)$$

The continuity equation is partially differentiated with respect to $\bar{z}$ to determine the non-dimensional radial component of velocity ($\bar{w}$) as:

$$\frac{\partial^2\bar{w}}{\partial\bar{z}^2}+\bar{h}\,\frac{\partial}{\partial\bar{z}}\left[\frac{\partial\bar{u}}{\partial\alpha}+\frac{\partial\bar{v}}{\partial\beta}\right]-\frac{\partial}{\partial\bar{z}}\left[\bar{z}\,\frac{\partial\bar{u}}{\partial\bar{z}}\,\frac{\partial\bar{h}}{\partial\alpha}\right]=0 \quad (12)$$

Integrating the above equation with finite difference method considering the following boundary conditions:

$$\bar{w}=0\ \text{at}\ \bar{z}=0\ \text{and}\ \bar{w}=\frac{\partial\bar{h}}{\partial\alpha}\ \text{at}\ \bar{z}=1 \quad (13)$$

The three dimensional energy equations have been solved with the following boundary conditions:

(i) On the fluid–journal interface $(\bar{z}=1)$

$$\overline{T_f}=\overline{T_j} \quad (14)$$

(ii) On the fluid–bush interface $(\bar{z}=0)$

$$\overline{T_f}=\overline{T_b} \quad (15)$$

C. Heat Conduction Equation for Bush-Housing

Heat conduction analysis was performed to determine the bush temperatures. The Fourier heat conduction equation in the form of non-dimensional cylindrical coordinate form has been solved for the temperature distribution in the bush and is given below:

$$\frac{\partial^2\overline{T_b}}{\partial\bar{r}^2}+\frac{1}{\bar{r}}\,\frac{\partial\overline{T_b}}{\partial\bar{r}}+\frac{\partial^2\overline{T_b}}{\partial\beta^2}+\frac{1}{\bar{r}^2}\,\frac{\partial^2\overline{T_b}}{\partial\alpha^2}=0 \quad (16)$$

Using the following boundary conditions, heat conduction equation was solved.

i. On the interface of fluid–bush $(\bar{z}=0,\bar{r}=\overline{R_1})$:

Continuity of heat flux gives,

$$k_b \left(\frac{\partial \overline{T_b}}{\partial \overline{r}}\right)_{|\overline{r}=\overline{R_1}} = -\frac{k_f}{\overline{c}\,\overline{h}}\left(\frac{\partial \overline{T_f}}{\partial \overline{z}}\right)_{|\overline{z}=0} \quad (17)$$

ii. On the outer part of the bush housing ($\overline{r} = \overline{R_2}$): The free convection and radiation hypothesis gives

$$\left(\frac{\partial \overline{T_b}}{\partial \overline{r}}\right)_{|\overline{r}=\overline{R_2}} = -\frac{h_{ab}\,R}{k_b}\left(\overline{T_b}_{|\overline{r}=\overline{R_2}} - \overline{T_a}\right) \quad (18)$$

ii. On the lateral faces of the bearing ($\beta = \pm\lambda$):

$$\left(\frac{\partial \overline{T_b}}{\partial \beta}\right)_{|\beta=\pm\lambda} = -\frac{h_{ab}\,R}{k_b}\left(\overline{T_b}_{|\beta=\pm\lambda} - \overline{T_a}\right) \quad (19)$$

iv. At the outlet edge of bearing pad, free convection of heat flow from bush to fluid in the supply groove gives

$$\left(\frac{\partial \overline{T_b}}{\partial \alpha}\right)_{|\alpha=\alpha_e} = -\frac{h_{fb}\,R}{k_b}\left(\overline{T_b} - \overline{T_s}\right) \quad (20)$$

α_e = Circumferential coordinate of the outlet edge of bearing.

v. At the inlet edge of the bearing ($\alpha = \alpha_i$) and at the fluid supply point on the outer surface

$$\overline{T_b}_{|\overline{r}=\overline{R}_2} = \overline{T_s} \quad (21)$$

In addition, a free convection of heat between fluid and housing has been assumed

$$\left(\frac{\partial \overline{T_b}}{\partial \alpha}\right)_{|\alpha=\alpha_i} = \frac{-h_{fb}\,R}{k_b}\left(\overline{T_b} - \overline{T_s}\right) \quad (22)$$

Where α_i = circumferential coordinate of the inlet edge of bearing.

D. Heat Conduction Equation for Journal

For finding the temperature distribution in journal, the following assumptions were made,

i. Conduction of heat in the axial direction.
ii. Journal temperature does not vary in radial or circumferential direction at any section.
iii. Heat flows out of the journal from its axial ends.

Hence the following steady state unidirectional heat conduction equation was used for a journal:

$$k_j\left(\frac{\partial^2 \overline{T_j}}{\partial y^2}\right)\Delta y\, A_j + \Delta q = 0 \quad (23)$$

Where Δq = the heat input to the element $(q\,\Delta y)$; Δy = the length of element

The above equation reduces to the following non-dimensional form:

$$\pi\left(\frac{\partial^2 \overline{T_j}}{\partial \beta^2}\right) + \overline{q} = 0 \quad (24)$$

where $\overline{q}$ is the non-dimensional heat input to journal per unit length.

The above equations have been solved with the following boundary condition:

At the axial ends, i.e. $\beta = \pm\lambda$

$$\left(\frac{\partial \overline{T_j}}{\partial \beta}\right)_{|\beta=\pm\lambda} = -\frac{h_{aj}R}{k_j}\left(\overline{T_j}_{|\beta=\pm\lambda} - \overline{T_a}\right) \quad (25)$$

E. Mixing Of Oil in a Groove

Incoming temperature of fluid is less as compared to recirculating temperature of fluid. Thermal mixing analysis of hot recirculating and incoming cold fluid from supply groove was used to calculate the fluid temperature at the inlet of the groove. Energy balance equation is used to estimate the mean temperature of the fluid in a groove.

In this work, the overall energy balance equation is expressed in terms of mean temperature, T_m

$$\overline{Q}\;\overline{T_m} = \overline{Q_{re}}\;\overline{T_{re}} + \overline{Q_s}\;\overline{T_s} \quad (26)$$

Where $\overline{T_{re}}$ - recirculating hot fluid, for the unit length of bearing

$$\overline{Q} = \int_0^1 \left(\overline{h}\;\overline{u}\right) d\overline{z} \quad (27)$$

$$\overline{Q_s} = \overline{Q} - \overline{Q_{re}} \quad (28)$$

$$\overline{Q_{re}} = \int_0^1 \left(C_L\;\overline{h}\;\overline{u}\right) d\overline{z} \quad (29)$$

$$\overline{T_{re}}\;\overline{Q_{re}} = \int_0^1 \left(C_L\,\overline{h}\;\overline{u}\;\overline{T_f}\right) d\overline{z} \quad (30)$$

Mean temperature T_m related to the assumed temperature distribution, $\overline{T_f}\,(\overline{z})$ across the fluid film at the inlet of the bearing pad as below:

$$\overline{T_m} = \int_0^1 \overline{T_f}\,(\overline{z})\,d\overline{z} \quad (31)$$

Nomenclature

A_j Cross-sectional area of the journal (πR^2)

c Radial clearance, (m); $\overline{c} = c / R$

C_L Coefficient of contraction, C_L is unity in positive pressure region

$$C_L = \int_0^1 \left(\overline{u}\,\overline{h}\right)_{|te} d\overline{z} \Big/ \int_0^1 \left(\overline{u}\,\overline{h}\right)_{|\alpha} d\overline{z}$$

C_p Specific heat of fluid, (J/kg °C)

D Diameter of Journal, (m)

D_e Dissipation number

e Journal Eccentricity, (m); $\varepsilon = e / c$

$\overline{F_0}, \overline{F_1}, \overline{F_2}$ Non dimensional Integration functions of Viscosity

h Thickness of fluid-film,(m); $\overline{h} = h / c$

h_{ab} Convective heat transfer coefficient bush, (W/ $m^{2\,0}C$)

h_{aj} Convective heat transfer coefficient of journal, (W/$m^{2\,0}C$)

h_{fb} Convective heat transfer coefficient from bush to fluid in groove, (W/$m^{2\,0}C$)

h_{fj} Convective heat transfer coefficient from fluid to journal in groove, (W/$m^{2\,0}C$)

k_0, k_1, k_4, k_3 Coefficient of Viscosity

k_f, k_b, k_j Thermal conductivity of fluid, bush and journal, (W/m °C)

L Length of bearing, (m)

p Pressure , $\bar{p} = p/p_S$ (N/ m^2)

p_S Supply pressure, (N/ m^2)

P_e Peclet number,

q Heat input per unit length

Δq the heat input to the element

$\bar{q}$ non-dimensional heat input to journal per unit length

Q Fluid-flow, (m^3/s) $\overline{Q_S} = Q/(\omega_j \, c \, R^2)$

r Radial coordinate; $\bar{r} = r / R$

R Radius of journal, (m)

R_1, R_2 Inner and outer radius of bush, m $\bar{R}_1 = R_1 / R$, $\bar{R}_2 = R_2 / R$

T_r Reference temperature, (°C)

T_a Ambient temperature, (°C); $\bar{T}_a = T_a / T_r$

T_b Bush temperature, (°C); $\bar{T}_b = T_b / T_r$

T_f Fluid film temperature, (°C); $\bar{T}_f = T_f / T_r$

T_j Journal temperature,(°C); $\bar{T}_j = T_j / T_r$

T_S Supply temperature , (°C); $\bar{T}_S = T_S / T_r$

$\bar{T}_m$ Mean temperature, , (°C)

$\bar{T}_{re}$ recirculating hot fluid, for the unit length of bearing

t Time; $\bar{t} = t / \omega_j$

u, v, w Fluid velocity components, in circumferential, axial and radial directions respectively (m/s) $\bar{u} = \frac{u}{(\omega_j / R)}$, $\bar{v} = \frac{v}{(\omega_j / R)}$, $\bar{w} = \frac{w}{(\omega_j / R)}$

Cartesian Coordinate in circumferential, axial and radial direction, $\bar{z} = z / h$

X_j, Z_j Coordinates of journal centre, (m); $\overline{X}_j = \varepsilon \sin\phi$, $\overline{Z}_j = -\varepsilon \cos\phi$

W_x, W_y fluid-film reactions in x and z directions, N

x, y, z coordinate axes with origin at geometric center of bearing

α Circumferential cylindrical coordinate; x/R

α_G Groove angle (°)

β Axial cylindrical coordinate; y/R

ε Eccentricity ratio;

λ Aspect ratio; L/D

ϕ Attitude angle; (°)

μ Viscosity of fluid, (N.s/m^2); $\bar{\mu} = \mu / \mu_0$

μ_0 Reference viscosity of fluid,(N-s/m^2)

ρ Mass density of fluid, (kg/m^3)

ω_j Angular speed of the journal, (rad/s)