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Effect of Groove Location on Pressure Profile of Twin Axial Groove Journal Bearing

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The main aim of current work is to analyze the influence of groove location on the pressure profile of the hybrid journal bearing. The objective is to develop solution algorithm and computer program which will provide pressure profile developed in the space of clearance of bearing having a given value of Somerfield number. The solution algorithm shall also provide three dimensional pressure profiles of the bearing for different groove location. It was found that groove location with respect to the loading line strongly influences the parameters of performance because of the stronger groove interference in the pressurized hydrodynamic field. The comparison of circumferential pressure with angular location from +X-axis is shown for the ratio of length to diameter which comes out to be 0.5, the clearance ratio (C/R) is found to be 0.00294 while 1125 rpm is the speed. The groove axes optimal location lies between 60° to 90° ($\propto_1$ g =0°) to the line of loading.

Keywords: Groove, Hybrid journal bearing, Pressure profile, Somerfeld number, Super laminar flow

NOMENCLATURE

P_s Supply pressure
C/R Clearance ratio
$\overline{K}_{\alpha}$, $\overline{K}_{\beta}$ Non dimensionless tturbulence coefficients in X, Y direction
W Width of groove
λ Aspect ratio (L/D)
α_g Location of groove
Ω Rotational velocity of journal
H Film thickness between clearance space
S_n Somerfield number
$\dot{X}_j, \dot{Z}_j$ Journal centre velocity
a a Length of groove
L Length of bearing
D Diameter of bearing
μ Lubricant Viscosity

Introduction

Hybrid journal bearings are widely used in turbo-machinery subjected to high loads and high journal speeds. The lubricant is often supplied at a prescribed pressure from a supply hole at the inlet through one or two grooves. Groove location and their orientation have significant effect on pressure profile. These grooves are cut into the bearing's internal surface. The supply condition[1] of oil has a great influence on the pressure profile generated between the clearance spaces. They[2,3] further reported that double groove log bearings are commonly utilized particularly if the shaft is supposed to take rotation in both directions. Researchers[4,5] recommended the usage of the circumferential groove in the bearing centre, with the oil supply hole opposite the load bearing region. Groove position[6–8] has a significant influence on the dynamic properties of many diary bearings lubricated by axial groove liquid.[9] The dynamic output of the rear bearing and pressure distribution is affected by fluid flow regime[10,11] and working condition[12] of the bending. Researchers[13,14] have also worked on the stability analysis of journal bearing used for rocket propulsions.

Theoretical Model/ Analysis

Current segment explains the theoretical framework designed for the study of dual "groove journal bearing" results. Fundamental geometry of the bearing and two-dimensional unwrapped geometry are shown in Fig. 1.

The degree of positive fluid film pressure is shown by α_1 and α_2 in Fig 1. α_{1g} and α_{2g} reflect the middle line position of the grooves.

Flow Field Equation

The un-dimensional Rey equation that regulates the flowing of lubricating oil in space clearance of the

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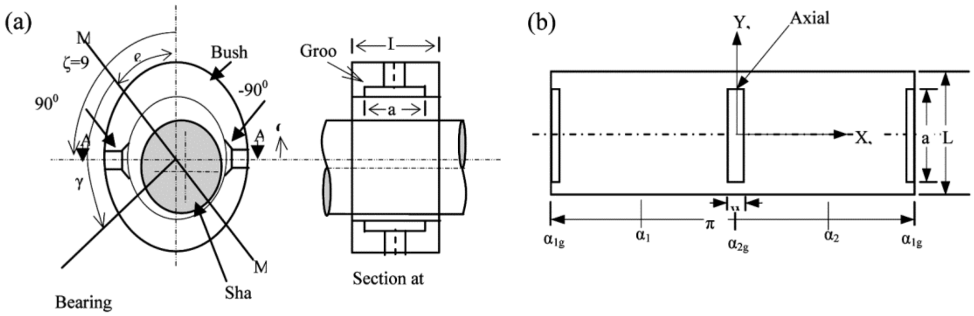

Fig. 1 — (a) basic bearing; (b) unwrapped geometry in two dimensions

"hydrodynamic bearings of journal" that uses the linear theory of turbulence given by Constatantinescu[10] in 1967 as provided in Eq. (1).

$$\frac{\partial}{\partial\alpha}\left[\frac{\bar{h}^3}{\bar{\mu}\bar{K}_\alpha}\frac{\partial\bar{p}}{\partial\alpha}\right]+\frac{\partial}{\partial\beta}\left[\frac{\bar{h}^3}{\bar{\mu}\bar{K}_\beta}\frac{\partial\bar{p}}{\partial\beta}\right]=\frac{1}{2}\bar{\Omega}\left(\bar{X}_j\sin\alpha-\bar{Z}_j\cos\alpha\right)-\left(\dot{\bar{X}}_j\cos\alpha-\dot{\bar{Z}}_j\sin\alpha\right) \quad \dots (1)$$

The thickness $\bar{h}$ of "non dimensional" fluid film for the case of the parallel axes is as mentioned in Eq. (2)

$$\bar{h}=1-\bar{X}_j\cos\alpha-\bar{Z}_j\sin\alpha \quad \dots (2)$$

Approximation for Short Bearings

Let us suppose that the bearing is made infinitely small keeping the gradient of pressure in the direction of circumference in quite smaller as compared to the "axial direction" such that $\frac{\partial\bar{p}}{\partial\alpha}\ll\frac{\partial\bar{p}}{\partial\beta}$.

The resulting Reynolds equation becomes (converges to)

$$\text{“}\frac{\partial}{\partial\beta}\left[\frac{\bar{h}^3}{\bar{\mu}\bar{K}_\beta}\frac{\partial\bar{p}}{\partial\beta}\right]=f(\alpha)\text{"} \quad \dots (3)$$

Such that $f(\alpha)=\frac{1}{2}\bar{\Omega}\left(\bar{X}_j\sin\alpha-\bar{Z}_j\cos\alpha\right)-\dot{\bar{X}}_j\cos\alpha-\dot{\bar{Z}}_j\sin\alpha$

Eq. (3) has been solved with the help of newly provided conditions of boundary as mentioned in the section below.

Conditions for Boundary

The model parameters "boundary conditions" related to the study were provided as

$$\text{(i) } \bar{p}=0 \text{ at } \beta=\pm\frac{L}{D}=\pm\lambda \quad \dots (4)$$

Integrating Eq. (3) with respect to β and using boundary conditions, Eq. (4) pressure distribution around the journal is obtained as

$$\bar{p}(\alpha,\beta)=\left(\frac{\bar{\mu}\bar{K}_\beta}{\bar{h}^3}\right)\left(\frac{1}{2}f(\alpha)(\beta^2-\lambda^2)\right) \quad \dots (5)$$

After obtaining pressure distribution all negative pressure is made zero. Further, following conditions were also applied.

$$\text{(i) } \bar{p}=\bar{p}_s \text{ at } \left[\left(-\frac{a}{D}<\beta<\frac{a}{D}\right) \text{ and } \left(\alpha_{1g}-\frac{w}{D}\right)<\alpha<\left(\alpha_{1g}+\frac{w}{D}\right)\right] \quad \dots (6a)$$

$$\text{(ii) } \bar{p}=\bar{p}_s \text{ at } \left[\left(-\frac{a}{D}<\beta<\frac{a}{D}\right) \text{ and } \left(\alpha_{2g}-\frac{w}{D}\right)<\alpha<\left(\alpha_{2g}+\frac{w}{D}\right)\right] \quad \dots (6b)$$

where α_{1g} and α_{2g} are the center lines of the angular position of grooves in radian

Load Carrying Capacity

The load-bearing capability of the bearing is calculated by incorporating the pressure from over positive field of pressure. Load ability in circumference direction determined as

$$\bar{F}_X=-\int_{-\lambda}^{\lambda}\int_0^{2\pi}\bar{p}.\cos\alpha\, d\alpha.\, d\beta \quad \dots (7)$$

and similarly load carrying capacity in radial direction (Z) is written as

$$\bar{F}_Z=-\int_{-\lambda}^{\lambda}\int_0^{2\pi}\bar{p}.\sin\alpha\, d\alpha.\, d\beta \quad \dots (8)$$

Eqs (7) and (8) are solved numerically using Gauss-Legendre numerical integration in the positive pressure zone

Computational Procedure

The equation of the Reynolds, which controls the flow of lubricant in a log-bearing clearance field, is adjusted by adding Turbulence Coefficients as well as the laminar, transformation and turbulent flow tests. A short bearing estimate had been used to overcome Reynolds equation. For the short bearing approximation, the closed form expression for pressure was obtained by combining the Reynolds equation twice and using new boundary conditions. Positive pressure area shall be defined by deletion of pressure below atmospheric (sub-ambient) and 100 kPa supply pressure (p_s) is taken at entire groove area. A computer program with modified boundary conditions was developed.

Results and Discussion

Solution and analysis algorithms have been employed to measure the profile for the pressure reaction of the fluid film, its stiffness and the coefficients of damping. Their works have done with the help of taking the aspect bearing ratio (L/D) 0.5 and 0.25. The assumptions that are taken for the bearings and the algorithm for solution have been used to find out the profile for the pressure, reaction of the fluid film and the capacity of loading, coefficient of damping and the stiffness of the fluid film. The aspect ratio of bearing during this experimentation work was kept at 0.25 & 0.5 (L/D). Taking into consideration the bearings and log axes parallel to the radius ratio of the sufficient clearance 0.001 (C/R = 0.001).

Validation of Results

The middle level circumferential pressure of the current short bearing approximation was contrasted with Brito *et al*. (2006)[2] findings in order to determine the validity of the study, solution algorithms and computer program. The L / D ratio of 0.50, C / R of 0.00294 and 1125 rpm are presented in Fig. 2 for comparison of circumferential pressure with angular direction.

Results well compare any slight unavoidable difference attributable to multiple methodologies for solutions. Following the validation of the solution process, findings are now provided in the following paragraph from the analysis of various groove lengths, location, and super-laminar flow conditions.

Influence of Location of the Grooves on Pressure Profile

The hydrodynamic pressure profile at the mid plane of the bearing is shown in Fig 3. The pressure profile is obtained by varying the angular location of

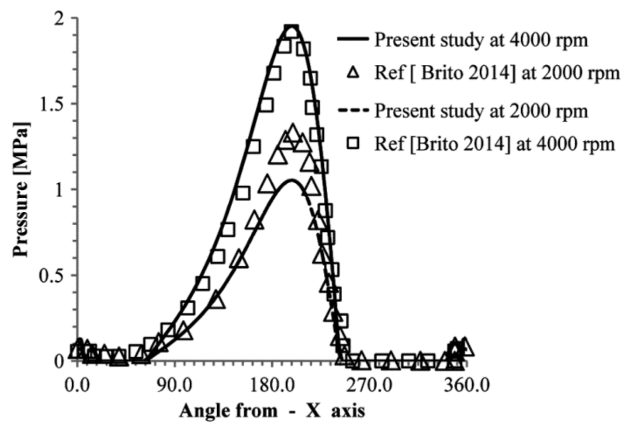

Fig. 2 — Comparison of pressure profile at the mid-plane of the bearing between present analysis with Brito *et al*. (2006)[2]

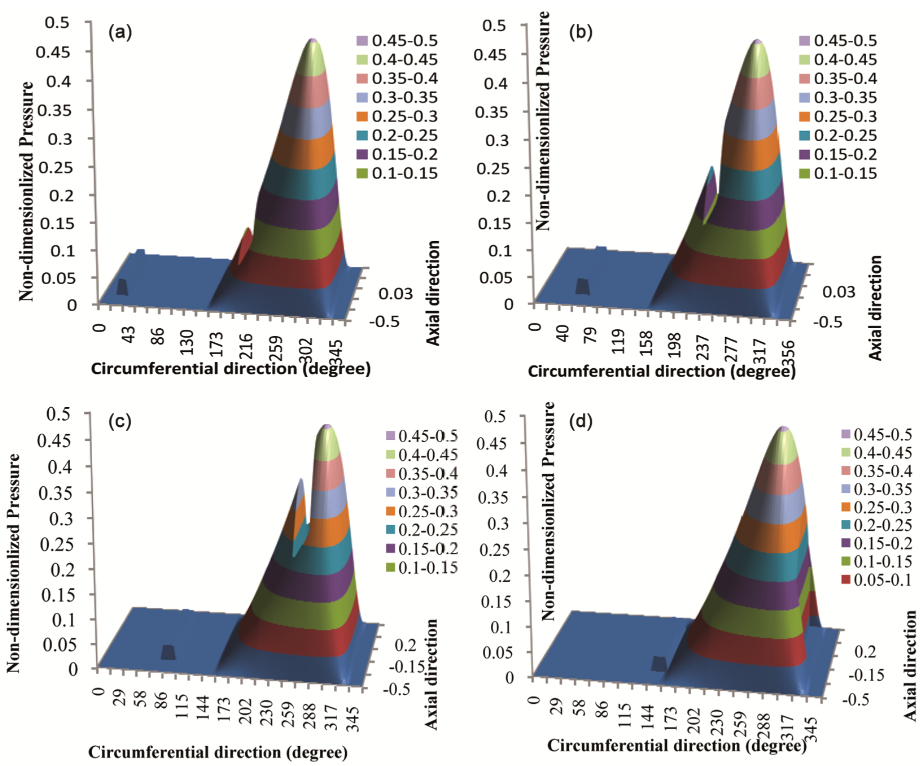

Fig. 3 — Variation of pressure profile for angular position of groove (a) α_{1g}=30° and α_{2g} =210°; (b) α_{1g}=60° and α_{2g} =240°; (c) α_{1g}=90° and α_{2g} =270°; (d) α_{1g}=150°and α_{2g} =330°

the grooves while length and width ratios are kept constant as a/L = 0.6 and w/d = 0.05.

Effect of angular location of the grooves on pressure profile is studied on both axial and circumferential directions. It was observed that maximum pressure decreases when location of Ist groove α_{1g} changes from 30° to 90° and then starts increasing when it changes from 90° to 150°. Maximum pressure is nearly equal for the location of Ist groove, α_{1g} equal to 30° and 150°. These variations are shown in Fig 3, the values of α_{1g}= 30°, 60°, 90° and 150° respectively and α_{2g}=210°, 240°, 270 and 330° respectively.

Conclusions

A parametric review has been conducted using the statistical / computational model of lubricant supply conditions on non-recessed hybrid rear log bearings. It is concluded that, owing to the high involvement of grooves in the hydrodynamic pressure area, the position of grooves in relation to a load line greatly affects most output parameters. The optimal position of the groove axis was reached at load lines varying from 60° to 90° (α_{1g} =0°).

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