Simulation Studies on Vertical Offset Non-Circular Journal Bearing Profile

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In the present simulation study, the performance characteristics of vertical offset non-circular journal bearing have been evaluated. During study, the thermal pressures have been found of increasing nature with increase in speed in both the lobes of the bearing. A considerable rise in oil-film temperature and thermal pressure has been found in lower lobe and comparatively a low rise in oil-film temperature and thermal pressures has been found in upper lobe of the bearing. The trend for variation of oil-film temperature with speed has been found of increasing nature. The load capacity, power loss in the bearing and bearing characteristics Sommerfeld number has been observed to be of increasing nature with speed. It has been concluded that more than one lobe have been reported in the bearing profile under study, thus the profile can be considered as an alternate to circular journal bearing for better stability applications.

Keywords: vertical offset bearing, oil-film temperature, thermal pressure, Sommerfeld number, load capacity

1. Introduction

High horsepower and high load machinery, such as steam turbines, centrifugal compressors, pumps and motors, find the application of journal bearings. The purpose of providing a bearing is to provide low friction environment to support and guide a rotating shaft. The journal bearing operates on the concept of hydrodynamic lubrication theory in which a thick film of lubricant separates the surfaces. The separation of surfaces at load occurs as a result of pressure generation in film of fluid. The combined effect of hydrodynamic action and fluid film results in pressure generation and the load carrying capacity of fluid film depend upon the magnitude of pressure generated within it. Usually hydrodynamic plain journal bearings experience a considerable variation in temperature due to viscous dissipation under normal operating conditions. The rise in temperature can significantly reduce the bearing performance as the lubricant viscosity is strong dependent of temperature. Moreover, excessive temperature can cause oxidation of the lubricant and, consequently, lead to failure of the bearing because of degraded lubricating oil. Hence, it is of great importance to know temperature distribution in bearing and related characteristics of the bearing. Hence, temperature rise can be regarded as one of bearing design

criteria. The lobed bearings such as elliptical, multi-lobe, and orthogonally displaced profiles results in the reduction of lubricant temperature rise and therefore can be termed as alternate to circular bearings. Each lobed bearing may have different applications based on the configuration, but all lobed bearing has enhanced stability (common characters) when compare to circular profile.

Vertical Offset Journal bearings (Fig. 1) is commonly known as a lobed bearing in which two lobes are obtained by vertically displacing two halves of a cylindrical bearing and can be used in gear boxes connecting turbine and generator for power generation industries. The importance of thermal effects in hydrodynamic journal bearings has been recognized since long. During literature review, the authors has found that different investigations has been carried out experimentally as well as theoretically by many researchers especially for circular and to some extent for non-circular journal bearing profiles. However, only a limited experimental work and even theoretical work has been reported for thermal effects in lobed bearing configurations such as orthogonally displaced bearings, two lobe bearing and elliptical/lemon bore bearings which leads to the non availability of standard procedures for their design. Some recent experimental

Fig. 1 Schematic diagram of vertical offset journal bearing

and theoretical work related to non-circular journal bearing has been reviewed here. Hopf and Schuler [2] have carried out an experimental investigation into large turbine bearings including an elliptical bearing and a tilting pad bearing of 500 mm journal diameter under transitional conditions between laminar and turbulent flow. It has been found during the study by authors that under laminar flow conditions, circumferential temperature excursions and temperature differences across the film could be very significant. Read and Flack [3] have established a test apparatus on which an offset-halves journal bearing of 70 mm diameter journal was tested. The author has reported some experimental data for the same bearing. Fitzgerald and Neal [4] have presented some experimental data for 76 mm diameter two-axial groove circular bearings. The authors have reported that the axial temperature variation was negligible but the circumferential temperature variations are very significant. Basri and Gethin [5] have carried out some experimental investigations into the thermal behavior of three lobe profile bore journal bearing. The authors have considered three geometries: symmetric, tilted, and tilted with side rail. Global behavior is also presented by the authors, which demonstrates the trends similar to those with a cylindrical geometry. The results by the authors confirm the effectiveness of the side rail in reducing lubricant side flow while having only a marginal effect on thermal excursions. Hussain et al. [6] has predicted the temperature distribution in various non-circular journal bearing profiles namely two-lobe, elliptical, and orthogonally displaced. The work reported is based on a two-dimensional treatment following Mc Callion's approach (1970) in which the Reynolds and energy equations in oil film are decoupled by neglecting

all pressure terms in the energy equation. It has been concluded by the authors that the temperature rise is highest for the two-lobe bearing followed by elliptical and orthogonally displaced bearings. Ma and Taylor [7] have experimentally investigated the thermal behaviour of a two-axial-groove circular bearing and an elliptical bearing, both 110 mm in diameter, at specific loads upto 4 MPa and rotational frequencies up to 120 Hz. Power losses and flow rate were measured directly and detailed temperature information was collected and the results reported by them shows that the thermal effects are significant in both bearings. Sehgal et al. [8] has presented a comparative theoretical analysis of hydrodynamic journal bearing profiles namely: circular, axial groove, and offset-halves. It has been observed by the authors that the offset bearing runs cooler than an equivalent circular bearing with axial grooves. Mishra et al. [10] has considered non-circularity in bearing bore to be elliptical and made a comparison with circular one. It has been shown that with increasing non-circularity, pressure gets reduced and temperature rise is less in case of bearing with high non-circularity. Chauhan et al. [12] has presented a solution of Reynolds equation along with the energy equation and heat conduction equation by Finite Difference Method and used Parabolic Temperature Profile Approximation technique for the solution of Energy equation to achieve faster computation. A comparative study for the rise in oil temperatures, thermal pressures and load carrying capacity for three different commercially available oils have been carried out. Computed oil temperature rise, thermal pressures as well as load carrying capacity have been found maximum for Mak Multigrade oil and minimum for Mak 2T oil. Sehgal et al. [13] has presented experimental study on oil-film temperature of elliptical journal bearing at different loads for various speeds. The authors has carried out the study for three lubricating grade oils and observed that the oil-film temperature rise is observed in both the lobes of the bearing and rise is low for the Oil 2 under the study.

The literature review reveals that almost all previous experimental/theoretical work pertains to non-circular journal bearings like elliptical, three lobe, and orthogonal bearings with a horizontal shift. Just one paper [6] related to vertical offset journal bearing has been found by the author while reviewing the literature. However, such bearing can be made useable as they possess excellent stability characteristics due to lobed nature. Thus, here the thermal parameters such as pressure, oil-film temperature, power loss, Sommerfeld number and load capacity in a vertical offset journal bearing have been determined. The variation of these parameters with speed has been presented and in addition to this the three-dimensional variation of thermal pressure has also been shown in the current simulation study.

2. Governing equations

Reynolds equation:

The Reynolds equation for steady-state and incompressible flow given by the equation [6]:

$$\frac{\partial}{\partial x} \left(\frac{h^3}{\mu} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial z} \left(\frac{h^3}{\mu} \frac{\partial p}{\partial z} \right) = 6U \frac{\partial h}{\partial x} \quad (1)$$

Equation (1) gives isothermal pressures as well as thermal pressures. The viscosity variation with temperature and pressure has been obtained using the following viscosity relation:

$$\mu = \mu_{ref} e^{\alpha P - \gamma(T - T_0)} \quad (2)$$

Energy equation:

The energy equation for steady-state and incompressible flow is given as [11]:

$$\rho C_p \left(u \frac{\partial T}{\partial x} + w \frac{\partial T}{\partial z} \right) = \frac{\partial}{\partial y} \left(K \frac{\partial T}{\partial y} \right) + \mu \left[\left(\frac{\partial u}{\partial y} \right)^2 + \left(\frac{\partial w}{\partial y} \right)^2 \right] \quad (3)$$

In eqn. (3), x-axis represents the axis along the circumference of bearing, y-axis represents the axis along the oil-film thickness and z-axis represents the axis across the width of bearing.

The variation of temperature across the film in eqn. (3) has been approximated by parabolic temperature profile. The expressions for velocities u and w in above equation are given as:

$$u = \left(\frac{y^2 - yh}{2\mu} \right) \frac{\partial P}{\partial x} + u_L \left(\frac{h - y}{h} \right) + u_U \left(\frac{y}{h} \right) \quad (4)$$

$$w = \left(\frac{y^2 - yh}{2\mu} \right) \frac{\partial P}{\partial z} \quad (5)$$

The temperature profile across the film thickness is represented by a second order polynomial as:

$$T = a_1 + a_2 y + a_3 y^2 \quad (6)$$

The constants appearing in equation has been obtained using the boundary conditions

$$\text{At } y = 0, T = T_L \quad (a)$$

$$\text{At } y = h, T = T_U \quad (b)$$

$$\text{and } T_m = \frac{1}{h} \int_0^h T dy \quad (c)$$

Using these relations, the eqn. (6) can be written as:

$$T = T_L - (4T_L + 2T_U - 6T_m) \left(\frac{y}{h} \right) + (3T_L + 3T_U - 6T_m) \left(\frac{y}{h} \right)^2 \quad (7)$$

Where, T_L , T_U , and T_m represent temperatures of the journal, bearing and mean temperature across the film respectively.

The energy equation after integration from the limit '0' to 'h' gives the following form:

$$\begin{aligned} & 6T_L + 6T_U - 12T_m \\ & - \frac{\rho C_p h^4}{120K\mu} \frac{\partial P}{\partial x} \left(\frac{\partial T_L}{\partial x} + \frac{\partial T_U}{\partial x} - 12 \frac{\partial T_m}{\partial x} \right) \\ & - \frac{\rho C_p h^4}{120K\mu} \frac{\partial P}{\partial z} \left(\frac{\partial T_L}{\partial z} + \frac{\partial T_U}{\partial z} - 12 \frac{\partial T_m}{\partial z} \right) \\ & - \frac{\rho C_p h^2 (u_L + u_U)}{2K} \frac{\partial T_m}{\partial x} \\ & - \frac{\rho C_p h^2 (u_U - u_L)}{12K} \left(\frac{\partial T_U}{\partial x} - \frac{\partial T_L}{\partial x} \right) \\ & + \frac{h^4}{12K\mu} \left[\left(\frac{\partial P}{\partial x} \right)^2 + \left(\frac{\partial P}{\partial z} \right)^2 \right] + \frac{\mu (u_U - u_L)^2}{K} \\ & = 0 \end{aligned} \quad (8)$$

Heat conduction equation:

The Laplace equation has been used to evaluate the bush temperature within the bearing material given below [9]. During analysis variation of temperature in rotating member has been taken constant since it is insignificant in magnitude [1].

$$\frac{\partial^2 T_b}{\partial x^2} + \frac{\partial^2 T_b}{\partial y^2} + \frac{\partial^2 T_b}{\partial z^2} = 0 \quad (9)$$

The above equation has been solved by using central difference technique.

Film thickness equation:

The film thickness equations for vertical offset journal bearing are given as [6]:

$$h = c_m [1 + E_M + \varepsilon_1 \cos(\theta + \varphi - \varphi_1)], \quad 0 < \theta < 180$$

$$h = c_m [1 + E_M + \varepsilon_2 \cos(\theta + \varphi - \varphi_2)], \quad 180 < \theta < 360 \quad (10)$$

The various parameters used in film thickness equation are given as:

$$\varepsilon_1 = (E_M^2 + \varepsilon^2 - 2E_M \varepsilon \cos \varphi)^{\frac{1}{2}};$$

$$\varepsilon_2 = (E_M^2 + \varepsilon^2 + 2E_M \varepsilon \cos \varphi)^{\frac{1}{2}}$$

$$\varphi_1 = \pi - \tan^{-1} \left(\frac{\varepsilon \sin \varphi}{E_M - \varepsilon \cos \varphi} \right);$$

$$\varphi_2 = \tan^{-1} \left(\frac{\varepsilon \sin \varphi}{E_M + \varepsilon \cos \varphi} \right);$$

$$E_M = \left(\frac{C_h - C_m}{C_m} \right)$$

In the expression of h, the first equation gives lower lobe profile whereas second equation gives film profile of upper lobe of the bearing.

3. Simulation procedure

Finite difference technique has been used to get the numerical solution of Reynolds, energy and heat conduction equations for vertical offset journal bearing profile. The number of nodes has been taken as 100, 50

and 10 in x , y , and z directions of the bearing respectively. The procedure adopted for obtaining the thermohydrodynamic solution is discussed below.

Reynolds equation:

A suitable initial value of attitude angle is assumed and film thickness equations are solved. Then the solution of equation (1) has been carried out to find isothermal pressures. Here, over relaxation with relaxation factor of 1.7 has been taken for error convergence. The initial viscosity value has been taken equal to inlet oil viscosity of the lubricating oil.

Energy equation:

The numerical solution begins with the known pressure distributions obtained by solution of Reynolds equation. Viscosity in the fluid film domain has been calculated using equation (2) corresponding to temperatures computed. With new value of viscosity i.e. after incorporating thermal effects, equation (1) has been solved for thermal pressure. Using the thermal effects, energy equation (8) has been further solved. Temperature profile obtained by solving equation (8) is substituted in equation (7) to find the temperature in the oil film. Now, this temperature is used to solve the equation (9) to obtain the temperature profile in the bush. Under relaxation factor of 0.7 has been taken in numerical solution of energy equation for error convergence. The boundary conditions used in the solution of governing equations has been taken from the Chauhan et al. [12]:

The convergence criterion for the solution of governing equations has been given below:

For pressure:

$$\frac{|\left(\sum P_{i,j}\right)_{n-1} - \left(\sum P_{i,j}\right)_n|}{\left|\left(\sum P_{i,j}\right)_n\right|} \leq 0.0001 \quad (11)$$

For temperature:

$$\frac{|\left(\sum T_{i,j}\right)_{n-1} - \left(\sum T_{i,j}\right)_n|}{\left|\left(\sum T_{i,j}\right)_n\right|} \leq 0.0001 \quad (12)$$

Where, n represents number of iterations.

4. Results and discussion

The input parameters considered to carry out the simulation study has been given in Table 4.1 (Appendix-I). The simulation results for oil temperature have been observed in good agreement to those reported by Hussain et al. [6]. The following observation has been made from the simulation carried on vertical offset journal bearing.

1. Figs. 2 (a, b and c) represent the thermal pressure distribution throughout the journal bearing of the profile under study. The thermal pressure has been found to be increasing with increase in speed in both the lobes of the bearing under study which clear from Figs. 2 (a, b and c). The simulation

(a)

(b)

(c)

Fig. 2 (a, b, and c) Thermal pressure for offset journal bearing profile at 3000, 4000 and 5000 rpm

results show a considerable pressure rise in lower lobe whereas small pressures rise in upper lobe of the bearing profile.

2. The oil-film temperature has also been found to be increasing with increase in speed in both the lobes of the bearing under study (Fig. 3). During simulation results, a considerable increase in oil-film temperature in lower lobe and comparatively small oil-film temperature rise in upper lobe has been observed. The trend for

Fig. 3 Variation of oil-film temperature along circumferential angle for speeds 3000, 4000 and 5000 rpm

variation of oil-film temperature with speed has been found of increasing nature.

3. The variation of load capacity, power loss in the bearing and the bearing characteristics Sommerfeld number has been presented in the form of Table 4.2 in Appendix-I. It has been found the all the parameters considered are of increasing nature with an increase in speed.

5. Conclusions

The simulation studies on the vertical offset non-circular journal bearing have been carried out. During the study, the effect of speed on thermal pressure, oil-film temperature, load capacity, power loss and Sommerfeld number has been analyzed. It has been found that the considered characteristics are significantly influenced by the variation in speed. Therefore, while selecting a suitable design, attention must be focused on all these characteristics since the rise in oil-film temperature can cause reduction in life of the bearing. However, the authors have observed two lobes in the vertical bearing profile which adds to stability and low rise in oil temperature. Hence, the vertical offset bearing can be considered as alternate for circular journal bearing from among the available non-circular journal bearings.

Nomenclature

C, C_h	Radial clearance, μm
C_p	Specific heat of lubricating oil, $\text{J/Kg}^\circ\text{C}$
C_m	Minimum clearance when journal centre is coincident with geometric centre of bearing, μm
e	Eccentricity, m
h	Film thickness for vertical offset bearing, m
g	Groove Angle
l	Length of the bearing, mm
N	Journal speed, rpm
O_B	Bearing centre
O_J	Journal centre
O_L	Lower-lobe centre
O_U	Upper-lobe centre

P	Film pressure, Pa
$P(i, j)_{iso}$	Isothermal pressure, Pa
$P(i, j)_{th}$	Thermal pressure, Pa
R	Journal radius, mm
r	Bush radius, mm
T	Lubricating film temperature, $^\circ\text{C}$
T_a	Ambient temperature, $^\circ\text{C}$
T_b	Bush temperature, $^\circ\text{C}$
T_0	Oil inlet temperature, $^\circ\text{C}$
U	Relative velocity between journal and bearing surface, m/s
u, w	Velocity components in X- and Z-directions, m/s
u_L	Velocity of lower bounding surface, m/s
u_U	Velocity of upper bounding surface, m/s
μ	Absolute viscosity, $\text{Pa}\cdot\text{s}$
μ_{ref}	Absolute viscosity at oil inlet temperature, $\text{Pa}\cdot\text{s}$
θ	Angle measured from the horizontal split axis in the direction of rotation
ω	Angular velocity of the shaft, rad/s
φ	Attitude angle
α	Barus viscosity-pressure index, Pa^{-1}
ρ	Density of the lubricating oil, kg/m^3
ε	Eccentricity Ratio
γ	Temperature viscosity-coefficient of the lubricant (K^{-1})

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Appendix-I

Table 4.1 Input parameters:

Shaft Radius	50 mm
External bush radius, D	100 mm
Length, l	100 mm
Radial Clearance, C	50 μ m
Initial Viscosity	0.04986 Ns/m ²
Oil inlet temperature, T_0	33°C
Speed	3000, 4000 and 5000 rpm
Eccentricity ratio, ε	0.7
Lubricant Specific heat	2000 J/kg.°C

Table 4.2 Variation of bearing Characteristics with speed:

	Speed		
	3000 rpm	4000 rpm	5000 rpm
Load Capacity	6.923	8.4585	9.6849
Power Loss	1.8522	3.1846	4.7875
Sommerfeld Number	0.352	0.368	0.3856