Performance Behaviours of Flared Journal Bearings

P. Ganai, R. K. Pandey, J. K. Dutt, R. P. Singh

Abstract

A plain cylindrical bore journal bearing is designed for a particular operating condition by selecting suitable lubricating oil. If such bearing is operated with other lubricating oils having different viscosities, its performance behaviour is bound to off-set from the designed point, which is not a favourable situation. Thus, in order to keep the performance behaviour of the bearing at the designed point even with different lubricating oils, a journal bearing having flaredness has been conceived and analysed herein. Therefore, the objective of this study is to explore the role of flaredness on the performance characteristics of journal bearing supporting the aligned shaft using the thermohydrodynamic lubrication (THL) model. Based on the results reported herein, it is found that with a flaredness of 0.0688^0 in the bearing, about 25% increase in the load carrying capacity and approximately 15% reduction in friction coefficient have occurred in comparison to the conventional plain circular bore journal bearing. Moreover, substantial reduction in the lubricant's average temperature has also occurred.

Keywords: Flared journal bearing, THL analysis, load carrying capacity, friction coefficient, and temperature rise.

1. Introduction

Due to inherent good operational characteristics, the fluid film journal bearings are widely used in mechanical systems and machines in industries to efficiently support and guide the loaded rotors. The performance behaviours of a journal bearing are very strong function of lubricant's viscosity. Moreover, it is worth mentioning here that it is very difficult in practice to maintain a constant value of oil viscosity for longer duration once oil is circulating in the lubrication circuit. This normally happens due to the oil getting contaminated and commencement of oil oxidation in the presence of moisture through lapses of time. Moreover, the journal bearings

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employed in the centrifugal oil filters are lubricated with the different oils (having different viscosity), which are processed for cleaning. It is essential to mention here that a conventional circular bore cylindrical journal bearing is designed at particular operating condition (load, speed, *L*, *D*, surface finish etc.) selecting a suitable lubricating oil. If the journal bearing which is designed with particular viscous oil is operated with different lubricating oil (having different viscosity and its VI value), its static and dynamic performance behaviours are bound to deviate from the designed point, which is undesirable situation from the operational perspective. Thus a need arises to conceive and analyse a journal bearing, which can operate with different lubricating oils (i.e. having different viscosities and VI values) without having considerable change in its designed performance behaviours. In order to explore the existence of a journal bearing having the desired features as discussed herein, the relevant literature review has been carried out and reported in the paragraph to come.

To assess the status of research in the area of fluid film bearings, Khonsari [1] and Tanaka [2] have published literature review papers in 1987 and 2000, respectively. Based on the information provided in these papers, it is observed that no research study has been carried out for developing a journal bearing, which can operate with different viscous oils while keeping the designed performance parameters unchanged. Moreover, the researchers [3, 4] have also reviewed and explored the dynamic performance behaviours of journal bearings. But the concepts of journal bearings operating with any viscous oil without changes in it designed performance behaviours have not been reported. Recently published studies [5, 6] discuss the grooving and texturing in the journal bearings for enhancing the performance behaviours, however, it is observed that the role of the flaredness on the bearing performance behaviours has not been explored yet.

The primary objective of this study is to explore the role of flaredness on the static performance behaviours of journal bearing supporting the aligned shaft using the THL model. Moreover, the additional objective of this study is to compare the performance parameters of flared journal bearing with conventional journal bearing.

2. Mathematical model

The governing equations employed in the proposed numerical simulation are presented in this section. The schematic diagram of a flared journal bearing with the coordinate system is also provided herein. Figure 1 demonstrates the sectional and unwrapped views of a typical flared journal bearing. It is worth mentioning here that the flaredness is an axial tapering in the bore along the length of journal bearing as can be seen in Fig. 1. The flaredness leads in the variation of radial clearance at different cross-sections of the bearing, which accommodates different viscous oils at different portions of the bearing along the axial length depending on the oils' viscosity. The flaredness conceived in the journal bearing is different from the flaredness of bearing arising due to the operational issues such as due to wear, misalignment, and deformation. The flared journal bearing conceived herein has circular bore diameter at the mid length of bearing, which is equal to the clearance of conventional cylindrical journal bearing under comparison. However, one end of the flared bearing has marginally less and other end marginally large bores than mid-section bore.

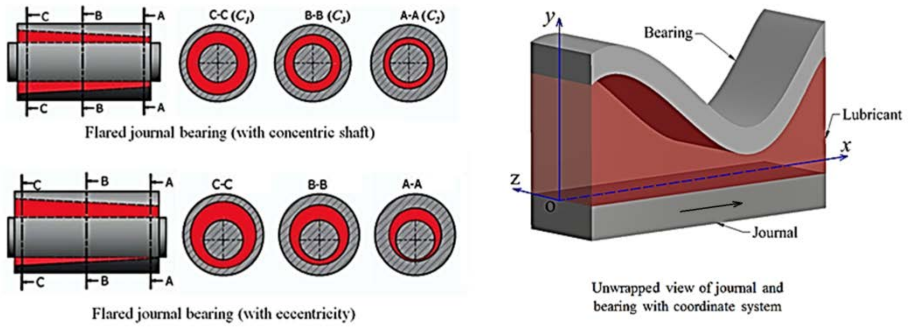

Fig.1 Geometry of flared journal bearing and description of the coordinate system

Pressure distributions in the conventional and flared journal bearings have been computed using the Reynolds equation applicable for the incompressible, non-inertial, steady laminar flow of Newtonian lubricant rheology. The normalised form of Reynolds equation is written as below:

$$\frac{\partial^2 \bar{p}}{\partial \theta^2} + \frac{3}{\bar{h}}\frac{\partial \bar{h}}{\partial \theta}\frac{\partial \bar{p}}{\partial \theta} + \left(\frac{2R}{L}\right)^2 \left(\frac{\partial^2 \bar{p}}{\partial Z^2} + \frac{3}{\bar{h}}\frac{\partial \bar{h}}{\partial Z}\frac{\partial \bar{p}}{\partial Z}\right) = \frac{\bar{\eta}}{\bar{h}^3}\frac{\partial \bar{h}}{\partial \theta} \tag{1}$$

where $Z = z/(L/2)$, $\bar{h} = h/C_3$, $\bar{p} = p/(6\eta_0 UR/C_3{}^2)$, $\bar{\eta} = \eta/\eta_0$

R: bearing bore radius (m), L: bearing length (m), h: film thickness (μm), p: pressure (Pa), U: journal speed (m/s), η_o: lubricant's viscosity (Pa-s), z: coordinate in axial direction, θ: circumferential angle (degree), C_3: radial clearance at mid-section (μm). For accounting the temperature increase in the lubricating film due to the viscous heat dissipation, the following form of normalised energy equation has been adopted in the computation:

$$\begin{aligned} &\left(\frac{1}{2} - \frac{1}{12}\frac{\bar{h}^2}{\bar{\eta}}\frac{\partial \bar{p}}{\partial \theta}\right)\frac{\partial \bar{T}}{\partial \theta} + \left(-\frac{1}{3}\frac{R}{L}\frac{\bar{h}^2}{\bar{\eta}}\frac{\partial \bar{p}}{\partial Z}\right)\frac{R}{L}\frac{\partial \bar{T}}{\partial Z} \\ &= \alpha \frac{\bar{\eta}}{\bar{h}^2}\left[1 + \frac{1}{12}\left(\frac{\bar{h}^2}{\bar{\eta}}\frac{\partial \bar{p}}{\partial \theta}\right)^2 + \frac{1}{3}\left(\frac{\bar{h}^2}{\bar{\eta}}\frac{R}{L}\frac{\partial \bar{p}}{\partial Z}\right)^2\right] \end{aligned} \tag{2}$$

where $\alpha = (2\pi N \eta_0 \beta / \rho C_p) \times (R/C)^2$, $\bar{T} = \beta(T - T_0)$, $\bar{\eta} = \eta/\eta_0$; N: Journal speed (rpm); β: temperature viscosity coefficient (K^{-1}); ρ: lubricant's mass density (kg/m^3); C_p: lubricant specific heat (J/kg-˚C); T: temperature (˚C); η: lubricant viscosity (Pa-s); $\bar{T}$ and $\bar{p}$ are normalised pressure and temperature, respectively.

Film thickness varying in the domain around and along the axial length of the journal is expressed in the normalised form by the following relation:

$$\bar{h} = \left[1 + (1 - Z) \times (C_1 - C_2)/(2C_3)\right] \times (1 + \varepsilon \times \cos\theta) \tag{3}$$

where ε is eccentricity ratio, C_1 and C_2 are radial clearances at large and small bore ends, respectively, $C_3\ \left[= (C_1 + C_2)/2\right]$ is the radial clearance at mid cross-section of the bearing.

Viscosity variation with temperature in the proposed mathematical model is expressed using the following relation:

$$\bar{\eta} = \exp\left[-\beta\left(T - T_0\right)\right] \tag{4}$$

The normalised load carrying capacity of journal bearing has been calculated using the expression as written below:

$$\bar{W} = \int_{-1}^{+1}\int_{0}^{2\pi} \bar{p}\, d\theta dZ \tag{5}$$

where $\bar{W} = W\big/(3\eta_0 U R^2 L / C_3^2)$

Using the following relation, the normalised friction force is computed:

$$\bar{F} = \int_{-1}^{+1}\int_{0}^{2\pi} \left(0.5 * \bar{\eta}/h + 1.5\bar{h}(\partial\bar{p}/\partial\theta)\right) d\theta dZ \tag{6}$$

where $\bar{F} = F/\left(\eta_0 U L R / C_3\right)$

The coefficient of friction has been evaluated using the following expression:

$$\mu = F/W \tag{7}$$

3. Computational procedure

The coupled solution of governing equations has been achieved by discretizing the partial differential equations using the finite difference method (FDM). Thereafter, the solution of the linearized algebraic equations at the nodes of domain has been obtained using the Gauss-Siedel (GS) iterative scheme. The flowchart employed in the computation is shown in Fig. 2. The grid size (151 x 51) employed in the solution has been arrived based on the grid independent test. In the coupled solution of governing equations, the following boundary conditions have been implemented:

For pressure in Reynolds equation:

$\bar{p} = 0$; At the edges of the domain and at the location of oil hole

$\bar{p} = d\bar{p}/d\theta = 0$; At the film rupture point in the domain

For temperature in energy equation:

$\bar{T} = 0$; At the edges of the domain and at the location of oil hole

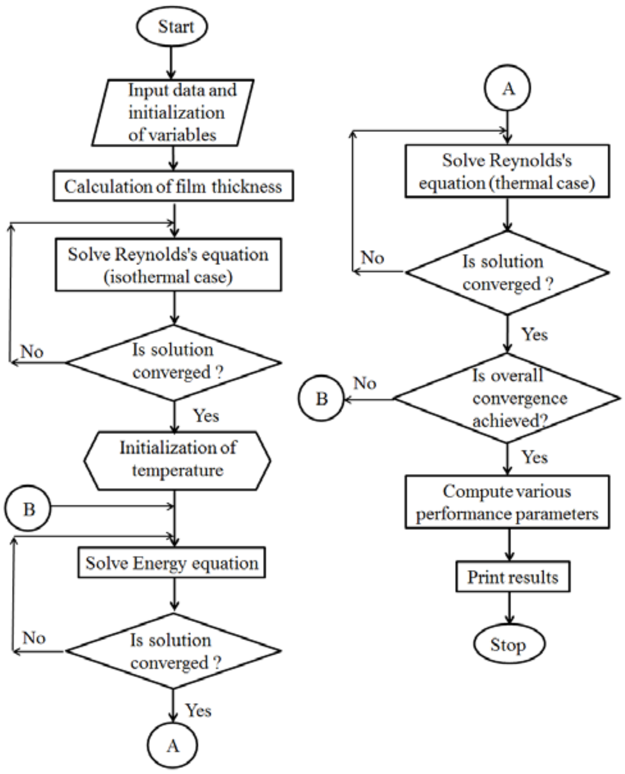

Fig. 2 Flowchart for computational procedure

Moreover in the solution of governing equations, the following criterions have been employed in achieving the convergence:

For pressure:

$$\left\{ \sum_{i=1}^{n} \sum_{j=1}^{m} | \bar{p}(i,j)^{N+1} - \bar{p}(i,j)^{N} | \right\} / \left\{ \sum_{i=1}^{n} \sum_{j=1}^{m} | \bar{p}(i,j)^{N+1} | \right\} \leq 10^{-4} \tag{8}$$

For temperature:

$$\left\{ \sum_{i=1}^{n} \sum_{j=1}^{m} | \bar{T}(i,j)^{N+1} - \bar{T}(i,j)^{N} | \right\} / \left\{ \sum_{i=1}^{n} \sum_{j=1}^{m} | \bar{T}(i,j)^{N+1} | \right\} \leq 10^{-4} \tag{9}$$

Where i, j are the symbols used for representing the nodes in θ and z directions, respectively. N is the number of the iterations.

4. Results and discussion

The numerical results achieved based on the proposed mathematical model have been compared with the published results of the researchers [7, 8] for developing the confidence in the present investigation. Figures 3(a) and 3(b) demonstrate the comparisons of the normalised circumferential pressure distributions at the mid-section of the bearing for the ε = 0.3 and ε = 0.9, respectively. Reasonably good matching between the results (present work and work of authors [7]) can be seen in these figures. Moreover, Figs. 3(c) and 3(d) also demonstrate the comparisons of circumferential normalised pressure profile and circumferential temperature in the lubricated film at the mid-section of the bearing for the adiabatic boundary conditions. Good matching of results (present work and work of authors [8]) can also be seen in these figures. Thus based on the comparisons of results provided in Fig. 3, it is concluded that parametric studies can be done using the proposed mathematical model.

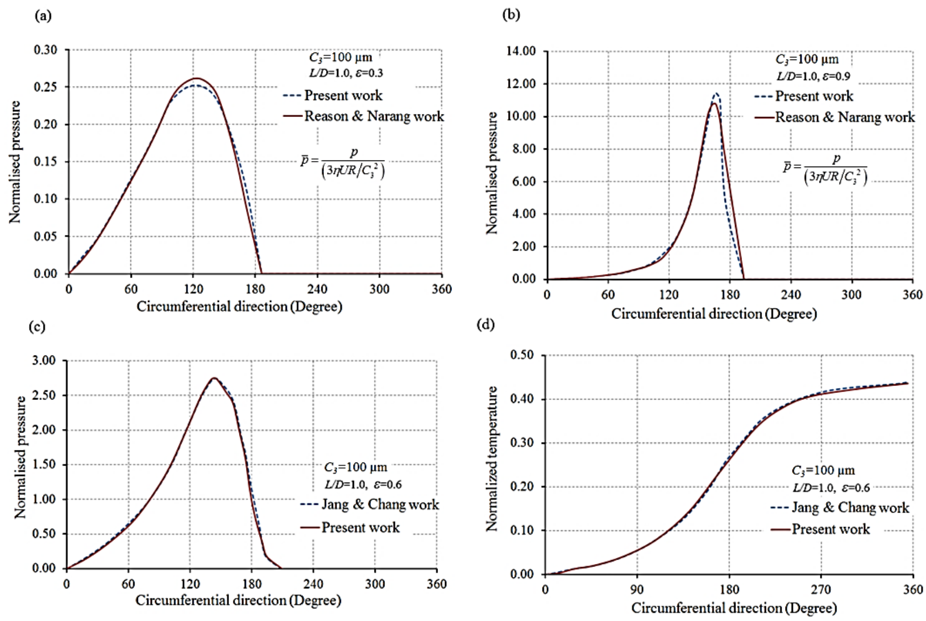

Fig. 3 Comparison of present numerical results with the study of researchers [7, 8]

The results reported in Figs. 4-7 have been generated for the lubricant data: viscosity (η_0) = 0.0416 Pa-s, density (ρ_0) = 881.6 kg/m^3, specific heat (C_p) = 1840 J/kg-^{0}C, temperature viscosity coefficient (β) = 0.0315 ^{0}C^{-1}, and inlet oil temperature (T_0) = 38^{0}C. Moreover, the additional data are also provided with the figures presented herein. Figures 4(a) and 4(b) demonstrate the comparisons of pressure distributions in the lubricating film in the circumferential and axial directions for the plain cylindrical and flared journal bearings, respectively. It can be seen that flared journal bearing develops high magnitude of pressure (refer Fig. 4(b)) in the zone where radial clearance happens to be less than C_3. Moreover at the mid-section of the bearing, the high magnitude of circumferential pressure profile in comparison to conventional journal bearing can also be seen in Fig. 4(a). This happens due to the less side leakage from the small clearance (C_2) end in the flared journal bearing in comparison to conventional bearing having C_3 clearance at every section of the bearing. Figure 1 can be referred for having the relative magnitudes of the clearances.

Comparisons of isothermal and thermal pressure distributions in the circumferential and axial directions in a flared bore journal bearing at different values of eccentricity ratios are illustrated in Figs. 5(a)-5(d). For the constant values of the operating parameters, drastic reduction in the magnitude of pressure values can be seen in Figs. 5(c) and 5(d) due to the thermal effect on the lubricating film. About 66% reduction in the value of maximum pressure (thermal case) is obtained with respect to the maximum pressure computed for the isothermal condition. The results reported in Figs. 6 and 7 reveal that with a flaredness of 0.0688^0 in the bearing, about 25% increase in the load carrying capacity and approximately 15% reduction in friction coefficient have occurred in comparison to the conventional plain circular bore journal bearing. Moreover, substantial reduction (about 5^{0}C) in the lubricant's average temperature has also occurred with flared journal bearing.

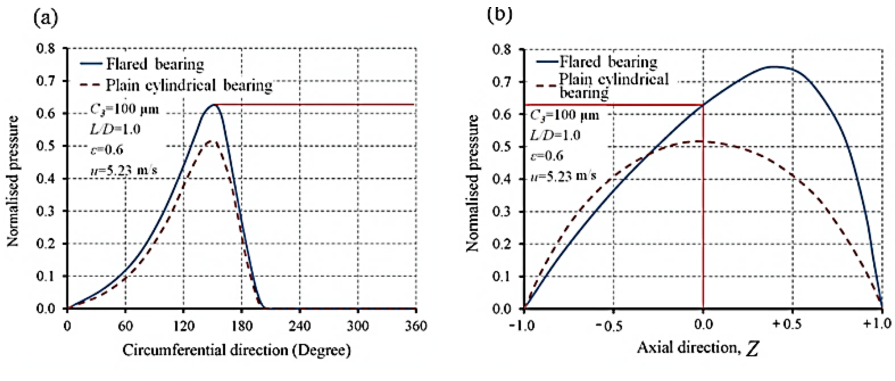

Fig.4 Comparisons of pressure distributions in the circumferential and axial directions in plain cylindrical and flared journal bearings [D=100 mm, flaredness (α) =0.0688^0]

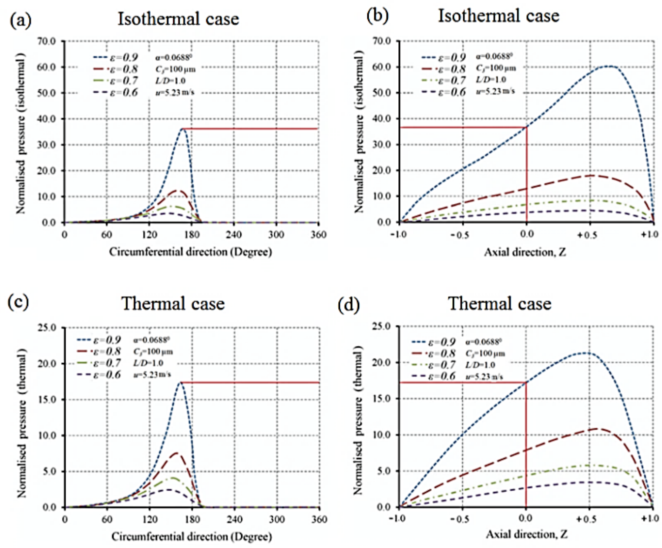

Fig. 5 Comparison of isothermal and thermal pressure distributions in flared bore journal bearing at different eccentricity ratios [*D*=100 mm]

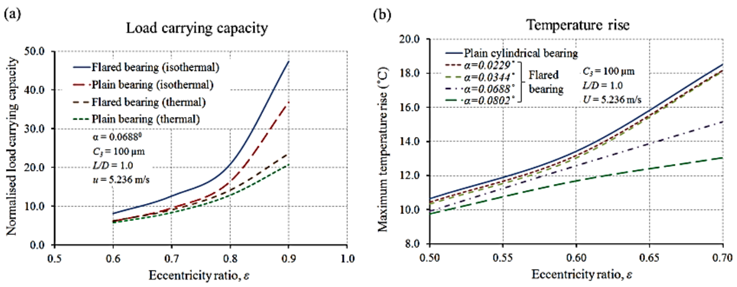

Fig.6 Comparison of load carrying capacity and temperature rise in the conventional and flared journal bearings [*D*= 100 mm]

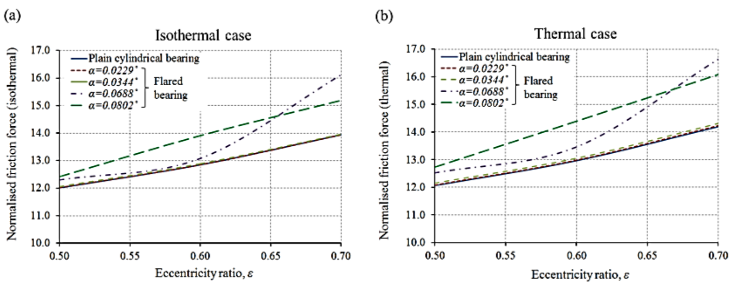

Fig.7 Friction force variation with eccentricity ratios for isothermal and thermal conditions for various values of flaredness [*L/D* =1.0, *D*=100 mm, Journal speed= 5.32 m/s]

5. Conclusions

Based on the investigations reported herein, it has been observed that a lightly loaded journal bearing having little flaredness and variation of lubricating oil viscosity shows comparable performance to conventional bearing designed with a lubricating oil having constant oil viscosity. Moreover, it is found that with a flaredness of 0.0688^0 in the bearing, about 25% increase in the load carrying capacity and approximately 15% reduction in friction coefficient have occurred in comparison to the conventional plain circular bore journal bearing. Moreover, substantial reduction in the lubricant's average temperature has also occurred.

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