Effect of elastic deformation on partial arc bearing with couple stress lubrication

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Abstract. A combined effect of deformation in bush liner and non-Newtonian (couple stress(CSF)) fluid is studied here for the system of partial arc journal bearing. A Reynolds equation with modifications along with elasticity equation, solved to satisfy the conditions of boundary. The effect of using CSF with bearing deformation on load carrying capacity, maximum pressure, static and dynamic characteristics is observed. An iterative procedure is adopted to solve system of equations by using FEM approach.

1. Introduction

The journal bearing is a machine element plays an important role to fulfil the emerging increased demand of high speed of rotation and heavy loads with machine compactness. It is very important to understand the exact geometry of the journal bearing systems to be used in required application area to provide a good service life of system. From decades a journal bearing used most preferably for better performances is journal bearing with hydrodynamic action. For practicability the elasticity of the journal and bearing effects the performance of the system. As material used for journal is found to be more rigid in comparison to the material used for bearing, it is assumed to be considered the journal to be rigid and the bearing as elastic for the analysis. The application areas of partial journal bearings are in numbers to withstand the loads during operation. An analytical solution rather than numerical to obtain the various properties of a liquid lubricated partial journal bearing [1] showed good agreement to the data available by Raimodi. The effect produced by deformation in bearing liner on various static properties for 60° and 120° centrally loaded partial bearings showed that the influence of flexibility parameter on short arc(60°) bearing is higher in comparison of long arc (120°) bearings and the attitude angle reduces for both (60° and 120°) arc bearings with an increase in deformation [2]. The capacity of the system for carrying load for lower eccentricity was superior than for higher eccentricity for partial arc gas bearing which are loaded centrally [3]. The temperature effect on the characteristics of bearings statically and dynamically was observed significant during a parametric study [4] of geometrically similar and with different size, bearings with the flow regimes of laminar & turbulent both. For partial arc journal bearings with mixed lubrication and pad deformation, a deterministic numerical solution [5] showed very well similarity with the available stochastic solution by Patir and Cheng. In a thermohydrodynamic analysis [6] of partial journal bearing, the influence of different wear related parameters was studied with global thermal effects. The performance of bearing in cylindrical journal bearings, was influenced significantly by the flexibility of liner [7]. An approach proposed by combining boundary element method and FEM [8] used to analyse the bearing housing deformation. The factors Rs, C, t, μ, Ur and Em relatively define the deformation coefficient factor [9] to evaluate the impact of liner flexibility on the properties of journal bearing. An inverse model [10] used to calculate both the parameter of permeability and eccentric ratio in flexible porous journal bearing. As per the wide
applications journal bearings in automotive sector also, there was a significant influence of the combined effect of the dimensions of the bearing and housing stiffness studied [11] on diesel engines connecting rod. The proper prediction of pressure distribution in lubricant film needs a mathematical model, for which a numerical algorithm [12] to uncouple the elasto-hydrodynamic load constraint, was suggested by solving the Reylond’s: Koiter model. Conventionally the elastohydrodynamic analyses assumes the lubricant behaves as Newtonian. It was observed by some published results that by adding long chained additives to the Newtonian lubricant showed its effects in the benefit of system characteristics. The Newtonian fluid blended with long chain polymer additives is known as couple stress fluid. Many theories of micro-continuum were used to explain the rheological behaviour of lubricants which are non-Newtonian. On the basis of theory of small micro-continuum, a model [13] (Stokes model) explained the impact of couple stress fluids. Addition of polymer particles to lubricants reduces the coefficient of friction and improves the capacity of carrying load [14-15]. The stokes model of micro-continuum was used to study the squeeze film characteristics for a couple stress lubricated partial arc bearing with long arc [16-18]. The combined effect of liner flexibility and couple stress parameter (CSP) on statical and dynamical properties of a cylindrical bearing showed that capacity of carrying load enhanced as such as an improvement in characteristics of system statical and dynamical [19]. A three dimensional (3-D) model of elasticity along with FEM approach was used to examine the performance characteristics, dynamic coefficients and stability parameters for varied ranges of CSP and elasticity parameter [20]. In slider bearings the ratio of film thickness at inlet-outlet and highest capacity of carrying load governed by mainly three key factors, geometry of bearing, couple stress factor and magnetic parameters [21]. A more stable behaviour of circular step thrust bearing pressurized externally was predicted while non Newtonian lubricant against Newtonian lubricant [22]. The use of CSF for lubricant in double layered journal bearings [23], results showed the reduced friction whereas increased load carrying capacity. Effect of CSF on contacts of finite-line showed that wear and friction decreased near edges [24].

In this paper the combined effect of the deformation in bearing liner and CSP is studied for a system of a partial arc journal bearing. The arc length considered for the analysis is 120° and the aspect ratio is 1.0. The load carrying capacity, variation in pressure profile and characteristics (static and dynamic) are analysed. The FEM analysis is performed with an iterative approach to find the solution.

2. Analysis
Consider a system of bearing with partial arc and lubrication of CSF and having the effect of flexibility of bearing liner on the system characteristics. The bearing is explained geometrically by Figure 1. The bearing with length L and an arc length of 120° is used for the analysis. It is assumed that the system is lubricated with an incompressible non Newtonian fluid, where forces and couples on body are found absent. Considered system is formulated with a set of equations which are explained below.

![Figure 1. Representation of partial arc journal bearing geometrically](image-url)

2.1. Modified Reynolds Equation
The Reynolds’ equation with modification by inclusion of CSP in the form of non-dimensional is as follows:

\[
\frac{\partial}{\partial \alpha} \left[ \frac{\partial \tilde{p}}{\partial \alpha} \right] + \frac{\partial}{\partial \beta} \left[ \frac{\partial \tilde{p}}{\partial \beta} \right] = 6 \frac{\partial \tilde{h}}{\partial \alpha} + \frac{\partial \tilde{h}}{\partial \tau} \tag{1}
\]

Where:

\[
G(\tilde{h}, \tilde{I}) = \tilde{h}^3 - 12\tilde{I}^2\tilde{h} + 24\tilde{I}^3 \tan \left( \frac{\tilde{h}}{2\tilde{I}} \right) \tag{2}
\]

\[
\alpha = \frac{x}{R}, \quad \beta = \frac{y}{R}, \quad \tilde{h} = \frac{h}{c}, \quad \tilde{p} = \frac{p_{\mu_0}^2}{\mu_0 \omega R^2}, \quad \tilde{\mu} = \frac{\mu}{\mu_0}; \quad \tilde{t} = \omega t; \quad U = R\omega; \quad \tilde{l} = \frac{l}{c} \tag{3}
\]

The interactions between the base lubricant and additives are represented by function G.

As discussed earlier also the flexibility of bearing liner effects oil film thickness which is explained by an expression below.

\[
\tilde{h} = 1 - \left( \tilde{x}_j + \tilde{X} \right) \cos \theta - \left( \tilde{z}_j + \tilde{Z} \right) \sin \theta + \delta \tag{4}
\]

Where:

\[
\tilde{x}_j = \varepsilon \sin \phi ; \quad \tilde{z}_j = -\varepsilon \cos \phi \quad \text{and} \quad \tilde{X} = \frac{x}{c}, \tilde{Z} = \frac{z}{c} \tag{5}
\]

According to equation (4), \( \delta \) is used as deformation in radial component of liner, \( \delta \) is a function of deformation coefficient \( (C_d) \) and pressure,

\[
\delta = C_d \tilde{X} \tilde{p} \tag{6}
\]

where, \( C_d = \psi \times \gamma_0 \); such that \( \psi = \left( \frac{\mu_0 \omega}{E} \right) \left( \frac{t_0}{R} \right)^3 \frac{R + h}{c} \), and \( \gamma_0 = \frac{(1+\gamma)(1-2\gamma)}{(1-\gamma)} \tag{7} \)

2.2 Elastic Deformation

The flexibility of bearing liner is an important property which increases the clearance space for the thickness of fluid film and effects the pressure variation for the new generated film thickness. A 3-D elasticity equation is used for the calculation of effect due to bearing deformation.

\[
[K] \{\delta\} = \psi [F] \tag{8}
\]

where \([K]\) = stiffness matrix,

\( \{\delta\} = \) deformation matrix, and

\( \{F\} = \) force matrix of flexibility model.

2.3 Solution Procedure

The Finite Element Method (FEM) approach is applied for the solution of system equations. A specified model considered for analysis is discretized into a mesh size of 50 X 40 (nodex X nodez), and film thickness is calculated at each node point by using an iterative approach.

The system is analysed by considering the boundary conditions which are as follows:

\[
\left( \frac{\partial \tilde{p}}{\partial \alpha} \right)_{\alpha_2} = 0, \quad \text{at the trailing edge of the film}
\]

\[
\left( \frac{p_i^k - p^{k-1}_i}{p^k_i} \right) < \left( 1 \times 10^5 \right), \quad \text{pressure convergence criteria,}
\]

Where, \( i \) is index for iteration and \( k = 1, 2, 3, \ldots n \).

The film thickness calculated is used to find the pressure profile of the film thickness within the clearance space to satisfy the above mentioned boundary conditions. The procedure is repeated for the values of eccentricity 0.2 & 0.3 by varying the parameter for couple stress \( (l^*) \) as 0.0 and 0.1.

3. Results and Discussions
The discretized model of partial arc journal bearing is solved to find the characteristics of the system statically and dynamically. The system is analysed for the eccentricity ratio of 0.2 & 0.3, such that the couple stress parameter considered for solution is 0.0 (rigid bearing) and 0.1.

To check whether the approach used to solve the partial arc bearing of arc length 120°, with aspect ratio (L/D) of 1.0 is validated or not, the results are cross checked with the published results of literature available (as in Table 1) below.

The comparison shows the good agreement of results with the published results by S. C. Jain et al., and the only difference is due to the approach used to solve the problem is different.

Table 1. Comparison of Load and pressure profile at Arc Length = 120°, Cd=0.0 and Aspect ratio = 1.0

| Validation | Load for Arc Length = 120°, Cd=0.0 and Aspect ratio = 1.0 |
|------------|-------------------------------------------------------------|
| Eccen= 0.1 | Eccen = 0.2 | Eccen = 0.3 | Eccen = 0.4 | Eccen = 0.5 | Eccen = 0.6 | Eccen = 0.7 |
| Load@Ref.(2) | 0.36 | 0.64 | 1.05 | 1.75 | 2.5 | 3.95 | 6.3 |
| Load@Present | 0.294598678 | 0.634921731 | 1.057034318 | 1.646169757 | 2.526919162 | 3.930878482 | 6.461995914 |

| Validation | Pressure for Arc Length = 120°, Cd=0.0 and Aspect ratio = 1.0 |
|------------|---------------------------------------------------------------|
| Eccen= 0.1 | Eccen = 0.2 | Eccen = 0.3 | Eccen = 0.4 | Eccen = 0.5 | Eccen = 0.6 | Eccen = 0.7 |
| Pressure@Ref.(2) | 0.2 | 0.4 | 0.65 | 1.105 | 1.7 | 2.755 |
| Pressure@Present | 0.172155961 | 0.376823593 | 0.644508865 | 1.039398577 | 1.667251688 | 2.760056022 | 4.927330106 |

Figure 2. Variation of load with Cd for ε = 0.2

Figure 3. Variation of stiffness coefficients (K_{xx}) with Cd for ε = 0.2

Figure 4. Variation of stiffness coefficients (K_{xy}) with Cd for ε = 0.2

Figure 5. Variation of stiffness coefficients (K_{yx}) with Cd for ε = 0.2
Figure 6. Variation of stiffness coefficients ($K_{yy}$) with $C_d$ for $\varepsilon = 0.2$

Figure 7. Variation of Damping coefficients ($C_{xx}$) with $C_d$ for $\varepsilon = 0.2$

Figure 8. Variation of damping coefficients ($C_{xy} = C_{yx}$) with $C_d$ for $\varepsilon = 0.2$

Figure 9. Variation of damping coefficients ($C_{yy}$) with $C_d$ for $\varepsilon = 0.2$

Figure 10. Variation of maximum pressure ($P_{max}$) with $C_d$

Figure 11. Variation of load ($W$) with $C_d$ for $\varepsilon = 0.3$

Figure 12. Variation of stiffness coefficients ($K_{xx}$) with $C_d$ for $\varepsilon = 0.3$

Figure 13. Variation of stiffness coefficients ($K_{xy}$) with $C_d$ for $\varepsilon = 0.3$
It is observed from the results for eccentricity 0.2 in Figure 2 that capacity of carrying load is enhanced by using the couple stress fluid but decreased by an increment in the value of deformation factor.

Figure 3-9 showed that the value of direct stiffness and direct damping in improved by increasing the couple stress parameter flexibility parameter both, whereas the value of cross stiffness and cross damping is reduced by increasing couple stress parameter but increased by increasing flexibility parameter.

In the same way, for eccentricity 0.3, it is observed from Figure 11, the capacity of carrying load is improved by using the couple stress fluid but decreased by an increment in the value of deformation factor.
For an eccentricity of 0.3 the results are explained by Figure 12-18, which shows that the value of direct stiffness and direct damping in improved by increasing the couple stress parameter flexibility parameter both, whereas the value of cross stiffness and cross damping is reduced by increasing couple stress parameter but increased by increasing flexibility parameter.

From Figure 10 and Figure 12 it is clear that the maximum pressure for partial arc bearing with an arc length of 120° is enhanced by using the couple stress parameter but decreased by increasing the deformation parameter for both the eccentricities of 0.2 and 0.3.

4. Conclusion

The system analysed here is a partial arc journal bearing with an arc length of 120° and a lubrication of couple stress fluid to predict a combined effect of deformation of bearing liner and couple stress parameter. It is founded that by the lubrication of couple stress fluid, the capacity of carrying load, maximum pressure and static and dynamic properties are improved for all the different values of deformation factor and both the eccentricities 0.2and 0.3.

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