Elastohydrostatic simulation of thrust bearing Compensated with orifice restrictor operation with non-Newtonian lubricant

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Abstract: In the present work influence of elastohydrostatic lubrication on orifice compensated bearing has been shown. The thrust pad has been modeled by using three-dimensional mesh and lubrication domain has been modeled by using two dimensional mesh. The analysis of Reynold equation is coupled with the analysis of solution of equilibrium equation of thrust pad domain. The present results shows that elasticity of thrust pad results a significant decrement in the fluid film Elasticity and it results an increase in fluid film stiffness. In the present case Circular, Elliptical, square and rectangular shape of recess has been taken from the analysis.

Keywords: Elastohydrostatic Lubrication, Non-Newtonian Lubricant.

1 Introduction

The thrust bearings support axial thrust coming from shaft. These bearing are able to give high value of stiffness and damping. Tilt in these bearing arises due to some assembly and manufacturing errors. It causes the reduction in the performance characteristics of the bearing. Therefore, in the past, many works have been done by researchers to analyze the behaviour of Hydrostatic thrust bearing under the tilt conditions [1, 2]. Further, consideration elasticity is also important for the exact prediction of bearing dynamic characteristic[3, 4]. Sharma et al.[4] carried out the study on thrust bearing in hydrostatic mode operation using capillary restrictors. The nonlinear finite element method was used for the analysis. In the present, analysis of tilted pad hydrostatic thrust bearing has been performed by considering the elasticity of thrust pad.

2 Analysis

The governing Reynolds equation in the clearance space shown in figure 1 is expressed by using the following expression [5, 6].
The orifice restrictor flow equation for lubricant is given by using following expression[7].

\[
\frac{\partial}{\partial \alpha} \left( \bar{h}^2 \bar{F} \frac{\partial \bar{v}}{\partial \alpha} \right) + \frac{\partial}{\partial \beta} \left( \bar{h}^2 \bar{F} \frac{\partial \bar{v}}{\partial \beta} \right) = \frac{\partial \bar{h}}{\partial t}
\]  

(1)

The orifice restrictor flow equation for lubricant is given by using following expression[7].

\[
\bar{Q}_r = \bar{C}_s \alpha_2 (1 - \bar{p}_{oc})^{\frac{1}{2}}
\]  

(2)

**Figure 1:** Schematic Diagram of Elastohydrostatic thrust bearing having orifice restrictor

Where, \(\bar{C}_s\alpha_2\) is parameter for restrictor design and \(\bar{p}_{oc}\) is pocket pressure

**2.1 Performance Characteristics of thrust bearing**

Non-dimensional bearing performance parameters have been calculated by using following expression[8].
2.1.1 Load carrying capacity

\[ F_0 = \sum_{e=1}^{n_e} \left\{ \sum_{i=1}^{n_i} \bar{p}_i \Omega_e \right\} + \bar{A}_{ac} \bar{p}_{ac} \]  

(3)

2.1.2 Fluid Film stiffness (\( \bar{S} \))

\[ \bar{S} = \sum_{e=1}^{n_e} \left\{ \int \int \left( \sum_{i=1}^{n_i} \frac{\partial \bar{p}_i}{\partial h} N_i \right) J d\xi d\eta \right\} + \sum_{i=1}^{n_p} A_{ac} \frac{\partial \bar{p}_{ac}}{\partial h} \]  

(4)

2.1.3 Fluid film damping Coefficient (\( \bar{C} \))

\[ \bar{C} = \sum_{e=1}^{n_e} \left\{ \int \int \left( \sum_{i=1}^{n_i} \frac{\partial \bar{p}_i}{\partial h} N_i \right) J d\xi d\eta \right\} + \sum_{i=1}^{n_p} A_{ac} \frac{\partial \bar{p}_{ac}}{\partial h} \]  

(5)

3 Solution procedure

In the present case fluid structure interaction problem has been solved for the bearing domain. In the present work it requires the nonlinear iterative method for the solution. The finite element method has been employed to solve fluid structure interaction problems. The solution of Reynolds equation incorporating the elasticity has been obtained.

Figure 2: Finite Element mesh for three dimensional elastohydrodynamics thrust pad Bearing for a) elliptical shape recess b) square shape recess c) square shape recess d) Circular shape recess
4 Results and discussion
A finite element mesh has been chosen on the basis of convergence study and presented in Figure 2. Three-dimensional mesh has been chosen for the solution of elasticity equation as shown in Figure 2. Four different type of recess has been chosen for the analysis. Following bearing operating parameter has been chosen in the simulation on the basis of simulation data available in literature.

Table 1: Operating parameter for the simulation

| S. No: | Geometric parameters                  | Value                                      |
|-------|---------------------------------------|--------------------------------------------|
| 1     | Tilt Parameter ($\lambda$)            | 0.0-0.8                                    |
| 2     | Type of compensating element          | Capillary                                  |
| 3     | Orifice Restrictor Coefficient ($C_{i2}$) | 5-40                                       |
| 4     | Poisson’s ratio                       | 0.3                                        |
| 5     | Non-Newtonian parameter               | 0 for Newtonian lubricant, 1 for psuedoplastic lubricant |
| 6     | Speed parameter ($\Omega$)            | 0.1                                        |
| 7     | Bearing Pad thickness                 | 0.2                                        |
| 8     | $A_y / A_x$                            | 4                                          |
| 9     | Lubricant Model                       | Cubic shear stress law                     |
| 10    | Element Type(Fluid Domain)            | Isoparametric quadrilateral Element        |
| 11    | Element Type(Elastic Domain)          | 8-noded brick Element                      |
|       | Coefficient of deformation ($C_d$)     | 0 For Rigid thrust pad, 0.5 For Flexible thrust pad |

4.1 Pocket pressure ($\bar{p}_{oc}$)
Pocket pressure in the bearing is key parameter for designing of bearing. Therefore, variation of pocket pressure has been shown in the Figure 3. As tilt in bearing increases the pocket pressure inside the bearing decrease monotonically. For consideration of tilt inside bearing, the following generalized pattern is noted for pocket pressure of the bearing.

$$\bar{p}_{oc}|_{\text{Rigid}} > \bar{p}_{oc}|_{\text{Flexible}} > \bar{p}_{oc}|_{\text{Crankle}} > \bar{p}_{oc}|_{\text{Square}} > \bar{p}_{oc}|_{\text{Elliptical}} > \bar{p}_{oc}|_{\text{Rectangular}}$$
Figure 3: Values of $\bar{n}_{uc}$ with $\lambda$

4.2 Load carrying capacity ($F_0$)

The variation of load capacity against tilt parameter is presented in Figure 4. This capacity of bearing reduces as the tilt parameter value increases. In case elastohydrostatic bearing, the load capacity is higher as compared to hydrostatic bearing. For this important parameter in bearing design, one can select the bearing parameter on the basis of following pattern.

\[
\begin{align*}
\bar{F}_0|_{Tilt} & < \bar{F}_0|_{Parallel}, \\
\bar{F}_0|_{Rigid} & < \bar{F}_0|_{Flexible}_{\kappa=0}
\end{align*}
\]
Variation of stiffness coefficient of the fluid film thrust bearing has been shown in Figure 5. Film stiffness constantly decreases for increasing value of tilt parameter. Elasticity of thrust pad increases the stiffness of bearing. Thrust bearing with circular recess has the least stiffness coefficient value and bearing by rectangular recess has the maximum film stiffness coefficient. Following important parameter has been obtained for stiffness coefficient of bearing.

\[ S_{\text{Tilt}} > S_{\text{Parallel}} \]

\[ S_{\text{Rigid}} < S_{\text{Flexible}} \]

**4.4 Damping coefficient of Bearing (\( \bar{C} \))**

The variation of damping coefficient of bearing has been shown in Figure 6. The damping in the bearing is an important parameter. It is capability of the system to absorb the vibration. The damping coefficient of bearing monotonically increases with an increase in tilt parameter value. The consideration of thrust pad elasticity in bearing shows the reduction in film damping coefficient. Following general pattern is noticed for damping coefficient.
\[ \bar{C}_{\text{Tilt}} > \bar{C}_{\text{Parallel}}, \quad \bar{C}_{\text{Rigid}} > \bar{C}_{\text{Flexible}}. \]

Figure 5: Values of $\bar{S}$ with $\lambda$

Figure 6: Values of $\bar{C}$ with $\lambda$
5 Conclusion

From the results presented above, the conclusions have been drawn, are:
1. Tilt parameter in the bearing results increment in stiffness as well as damping of bearing and it deteriorate load capacity of the bearing.
2. The consideration of thrust pad elasticity is important for the exact prediction of film stiffness and damping of bearing.

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