RESEARCH ARTICLE

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Numerical Analysis of Tilting Pad Thrust Bearing

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Abstract

Tilted pad thrust bearings are used in various rotating machinery that has to withstand high thrust loading. The thrust load is transferred from a sliding part to a stationary tilted part through hydrodynamic oil films. The pressure developed in the lubricant between the sliding surface and the tilted pad counteracts the external load applied to the sliding surface and thus prevents contact between the two surfaces. The present work studies hydrodynamic interaction between the sliding surface and tilted pad for a typical thrust bearing. Numerical analysis was conducted using COMSOL Multi-physics for symmetric boundary conditions, where elastic deformation along the thrust pad is neglected. Variation in pressure concerning different angular velocities is estimated. The effect of load on lubricant pressure developed and fluid film h eight is also studied. The results obtained are comparable with the analytical solutions available in the literature.

Keywords: Thrustpad, Load, Pressure, Film height, Angular velocity

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Introduction

Tilted pad thrust bearings are used in various rotating machinery that has to withstand high thrust loading. The thrust load is transferred from a sliding part to a stationary part through hydrodynamic oil films. The pressure developed in the lubricant between the sliding surface and the tilted pad counteracts the external load applied to the sliding surface and thus prevents contact between the two surfaces. The hydrodynamic interaction of lubricant film and its effect on bearing performance had been of interest to many investigators. Yang et al. (2019) have analyzed the load capacity of hydrodynamic thrust bearings using numerical techniques involving Fourier series decomposition. Qin et al. (2018) studied the structural effects of bearing element interaction in thrust bearings using a fluid-structural coupling model. SiyuGao et al. (2015) studied using the computational fluid dynamics approach that the performance characteristics of thrust bearings under various operating conditions. Remy et al. (2015) introduced a transient modified Reynolds equation by considering a detailed model representing multi-grade engine oils. Wasilczuk et al. (2014) considered the systems required for forced cooling of the lubricant in thrust bearings. Sunami et al. (2013) explored the application of oil-lubricated thrust bearings for hard disk applications. Variationalstudies to analyze the influence of changing parameters such as load or speed of operation of bearing. The present study employs a numerical model of tilting pad thrust bearing to find the pressure distribution and change in film thickness. Further, changing loads and angular velocities on fluid pressure developed and film height are analyzed.

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Numerical Modelling

COMSOL Multiphysics is employed for the numerical study of the problem. A single thrust pad is modeled for analysis purposes considering the symmetrical arrangement of thrust pads around the bearing axis. The physical model of the thrust pad developed in COMSOL Multiphysics is shown in Figure 1. Material properties of Nylon are assigned to the model.

The surface of interest is that of the thrust pad. Load is applied on the pad as distributed over the entire pad surface. Therefore, the elastic deformation and change in viscosity along the thrust pad are not accounted for in the analysis. Various parameters used for incorporating the operating conditions of the bearing onto the model are given in Table 1. A thin-film flow shell application module of COMSOL Multiphysics was employed to solve the problem.

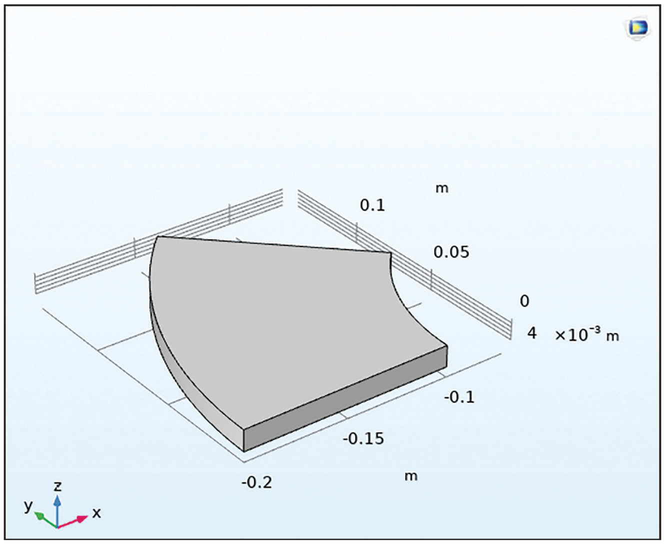

Figure 1: Physical model of thrust pad

Table 1: Operating characteristics of the bearing

Parameters	*Range of values*
Radius of thrust pad	0.2 m
Sector angle	45º
Rotation angle	135º
Layer thickness of thrust pad	0.1 m
Externalload	5000 N
Angularvelocity	1 rad/s
Density	0.33 kg/m^3
Fluid layer thickness	10 µm

The edges of the top surface of the tilted pad are assigned a boundary condition of ambient pressure.

$$p = 0 \text{ ---------------(1)}$$

The top face of the tilted pad has a face load.

$$Fr = -p \text{ -------------- (2)}$$

The bottom boundary is fixed with zero displacements in the *x*, *y*, and *z* directions. All the other boundaries are free.

The pressure in the lubricant (engine oil) is governed by Reynolds equation. For an incompressible fluid with no-slip condition, Reynolds equation in the continuum range is given by:

$$\nabla_T - \left(\frac{-\rho h^3}{12\eta}\nabla_T p + \frac{\rho h}{2}(v_a + v_b)\right) - \rho((\nabla_T b.v_b) - (\nabla_T a.v_a)) = 0......(3)$$

∇_T= tangential derivative operator
ρ = density (kg/m^3)
h= lubricant thickness
η =viscosity (Pa.s),
p = pressure (Pa)
a= location (m) of the channel base
v_a= tangential velocity of the channel base (m/s)
b = location of the solid wall (m)
v_b = tangential velocity of the solid wall (m/s)

The Reynolds equation is solved on the surface of the tilted pad. The external load (W_{ext}) applied to the sliding surface is counter-balanced by the pressure in the lubricant. This is imposed as the constraint:

$$\int_{\partial\Omega} p\,dS - W_{ext} = 0 \text{ ---------------- (4)}$$

Results And Discussions

Steady-state analysis was conducted on the model in COMSOL Multiphysics for the operating conditions depicted in Table 1. The effect of varying load on fluid pressure developed, and film height was analyzed. Typical plots of pressure developed and film height for a load of 200 kN are shown in Figures 2 and 3.

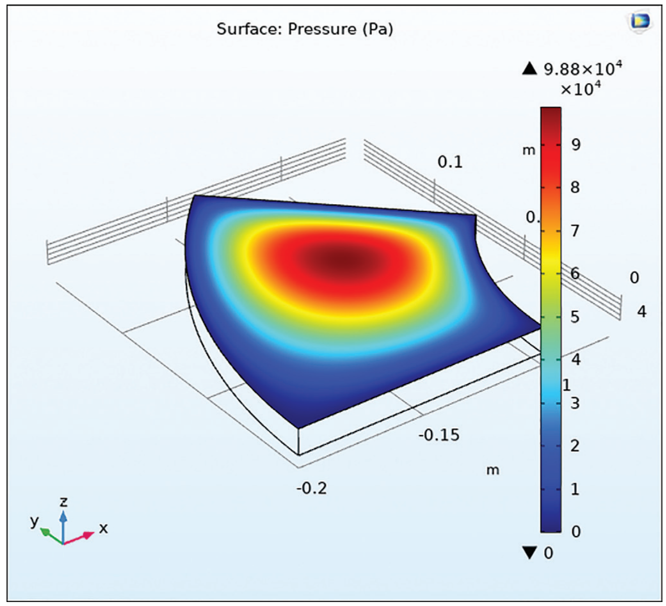

Figure 2: Fluid pressure distribution for 200 kN

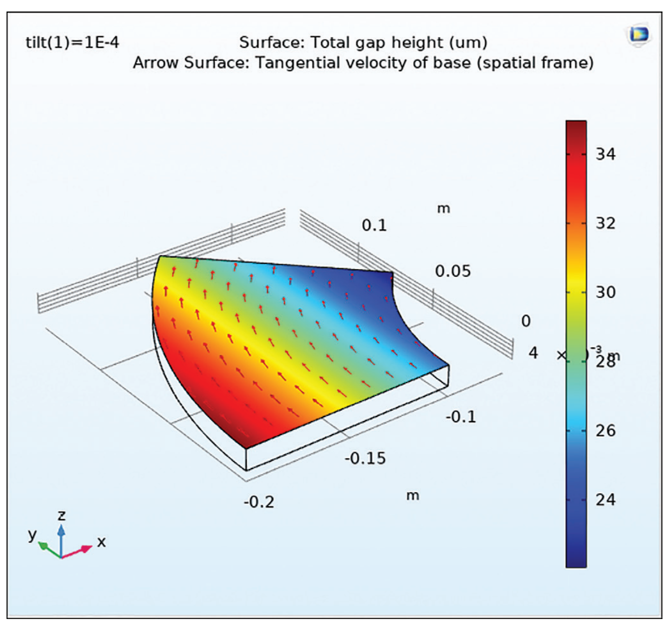

Figure 3: Film height distribution for 200 kN

Maximum pressure is developed at the pad center, whereas the film height changes from maximum to minimum from the leading corner of the pad to its trailing corner in the direction of the velocity of bearing rotation. The variation of maximum fluid pressure and maximum film height concerning change in applied thrust load is shown in Table 2 and Figure 4.

Table 2: Variation of maximum pressure and max. film height w.r.t load

Force (kN)	*Maximum pressure (kPa)*	*Fluid film thickness (μm)*
100	60.3	40
200	98.8	34
300	178	32
400	187	30

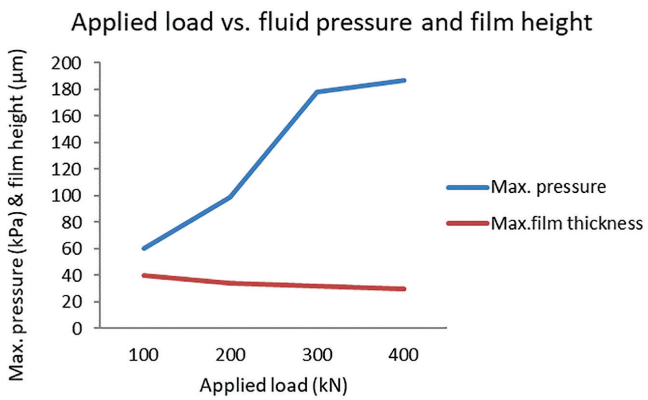

Figure 4. Effect of load on fluid pressure and film height

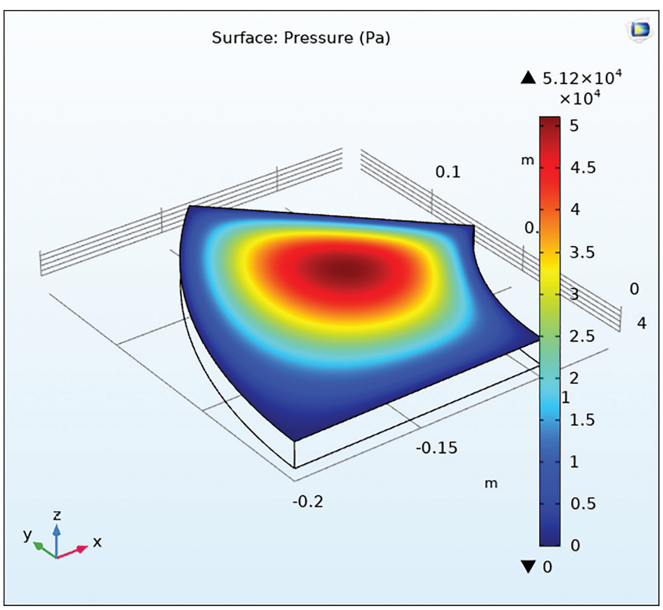

Figure 5: Pressure distribution over thrust pad for the angular velocity of 20 rad/s

As shown in Figure 4, as applied thrust load increases, fluid pressure increases when the fluid film thickness reduces. For example, when the film thickness reduces by 10 μm, the load pressure developed in the hydrodynamic film increases to about thrice the initial value. This indicates the load-carrying capacity of the thin lubricant film.

As the distribution of fluid film developed over the surface of the thrust pad showed a directional trend along the direction of the angular velocity of the bearing, the effect of varying angular velocity on pressure developed is also analyzed. The angular velocity varies from 10 to 50 rad/s keeping the applied load constant at 10 kN, and corresponding pressure distributions are obtained. For example, the pressure plot obtained for the angular velocity of 20 rad/s is shown in Figure 5.

The variation of maximum pressure developed at the center of the pad with the change in angular speed is explained in Table 3, and the variational plot is illustrated in Figure 6.

Table 3: Variation of maximum pressure with change in angular velocity

Angularvelocity (rad/s)	*Maximum Pressure (Pa)*
10	25600
20	51200
30	76800
40	102000
50	128000

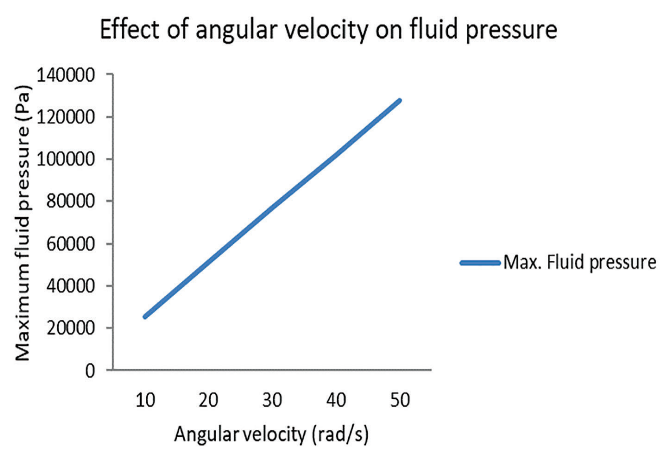

Figure 6: Variation of fluid pressure w.r.t angular velocity

Angular velocity of the bearing is seen to increase the fluid pressure in a linear fashion. However, as the fluid is modeled as a continuum medium, the breakage of the fluid film under tremendous pressure cannot be visualized. Therefore, the module cannot analyze the effects of film breakage and subsequent wear, which is a limitation of the study.

Conclusions

A numerical analysis is conducted to simulate the operating conditions inside a tilted pad thrust bearing. A simple physical model of a single thrust pad alone is employed assuming symmetric conditions. The operating parameters are incorporated by providing suitable load and boundary conditions. The numerical model can capture the pressure distribution and lubricant film thickness which are the deterministic factors indicating the performance of thrust bearings. The model can also predict changes in pressure distribution and lubricant film thickness in response to varying loads and operating velocities. However, as the modeling is done based on continuum thin-film flow assumption, breakage of the fluid film under high load cannot be predicted by the model. This can be termed a limitation of the model. Nevertheless, for simulating normal operating conditions of a thrust pad bearing, the present model proves to be quite capable.

Acknowledgments

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