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Performance of hydrostatic thrust pad bearings operating with electrically conducting lubricant

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Abstract
This paper theoretically investigates the influence of transverse magnetic field and recess shape on the performance of hydrostatic circular thrust pad bearing. The lubricant is assumed to be electrically conducting in nature. Navier-Stokes equation is modified by adding Lorentz force, so as to describe the flow of electrically conducting lubricant between bearing surfaces. A source code based on FE formulation of modified Reynolds equation is developed to compute the numerically simulated results. The bearing performance is presented in terms of load carrying capacity, fluid film stiffness and damping coefficient as a function of magnetic field (Hartmann number) and recess shape. Annular and elliptical recessed bearing operating with electrically conducting lubricants, are observed to be provide maximum increase in load carrying capacity (43.25%) and damping coefficient (67.7%) respectively.
Keywords: Hydrostatic thrust pad bearing, electrically conducting lubricant, Recess shape, FEM

1. Introduction

Hydrostatic thrust pad bearings are used for axial positioning of rotors in turbomachinery such as vertical pumps and hydraulic turbine rotors or supporting enormous structures such as telescopes, observatory domes, etc. Hydrostatic thrust pad bearings operates with a relatively thick fluid film between bearings usually maintain by external lubricant supply pumps. Initially, Fuller [1] and Brown [2] performed theoretical and experimental investigation to analyze the performance characteristic of hydrostatic thrust pad bearing operating with Newtonian lubricant. Later on, Sinhasan and Jain [3] numerically analyzed the influence of bearing shell elasticity on the performance indices of these bearings. Commercially available lubricants used in such bearings are often blended with traces of viscosity improvers or other additives to enhance the lubricating performance. Experimental studies carried out by Spikes [4] and Scott [5] reported that addition of viscosity improvers enhance the anti-friction and anti-wear characteristics of the lubricant. Adding traces of additives in base oil, makes lubricant exhibiting non-Newtonian behavior. Many researcher [6-10] in the past have used different non-Newtonian lubricant model to predict the behaviour of lubricant blended with traces of additives/viscosity improvers. Application of electrically conducting lubricant [10-16] have received considerable attention for enhancing the performance of this class of bearing. Recently, Kumar and sharma [16], theoretically investigated the performance of elliptical pad hydrostatic thrust pad bearing operating with electrically conducting lubricant. However, all of the abovementioned studies [11-16] have been carried out using either circular shape recess in hydrostatic thrust pad bearings or to predict the squeeze film performance of non-recessed uncompensated parallel plates. The geometric shape of recess significantly affect the pressure gradient and associated performance characteristics of the hydrostatic thrust pad bearings. Therefore, through this investigation an attempt have made to examine the influence of geometric shape of recess on the performance of capillary compensated hydrostatic thrust pad bearing operating with electrically conducting lubricant.

2. Mathematical Formulation

Electrically conducting lubricant is fed into the pocket of hydrostatic thrust pad bearing [3,15] via capillary restrictor. The modified Reynolds equation, governing the flow of electrically conducting lubricant is expressed as [11-16]:

$$\frac{\partial}{\partial \bar{x}}\left(\bar{h}^3 \emptyset(\bar{h},H)\frac{\partial p}{\partial \bar{x}}\right)+\frac{\partial}{\partial \bar{y}}\left(\bar{h}^3 \emptyset(\bar{h},H)\frac{\partial p}{\partial \bar{y}}\right)=\frac{\partial \bar{h}}{\partial \bar{t}} \qquad (1)$$

Where $\emptyset(\bar{h},H)$ [16] refers to the magnetohydrodynamic (MHD) function which itself depends on non-dimensional fluid film thickness $(\bar{h})$ and Hartmann number (H). After following usual finite element procedure and Galerkin's technique, the global modified Reynolds equation (Eq.1) in the form of matrix can be expressed as:

$$[\bar{F}]\{\bar{p}\}=\{\bar{Q}\}+\dot{\bar{h}}[\bar{R}_t] \qquad (2)$$

Steady state ($\dot{\bar{h}}=0$) performance characteristics i.e. fluid film pressure and load carrying capacity can be obtained by coupled solution equation (2) and capillary restrictor flow equation [9]. Dynamic characteristics i.e. fluid film stiffness ($\bar{S}=\partial\bar{F}/\partial\bar{h}$) and damping ($\bar{C}=\partial\bar{F}/\partial\dot{\bar{h}}$) coefficient of the bearing can be evaluated using finite element approach [9].

3. Result and discussion

Finite element numerical technique have been used to develop a source code based on Gauss–Siedel iterative method to obtain numerical solution of modified Reynolds equation (Eq.1). The bearing geometric and operating parameters are imported from the available literature [3, 9,14-15]. Bilinear isoparametric quad elements are used for discretization of land area in the thrust pad. Influence of geometry shape of recess is examined by using circular (CR), annular (AR), sectorial (SR) and elliptical (ER) pockets/recess in thrust pads. All recess/pocket shapes and thrust pad have identical area. The effect of magnetic field on bearing performance have been investigated by varying Hartmann number (*H*) from 0 to 10. Here *H=0* corresponds to Newtonian lubricant and *H=1-10* for electrically conducting lubricant. Mesh sensitivity test has been performed to obtain and use an appropriate mesh size for thrust pads employing different recess shapes. To terminate iterative numerical scheme, a convergence of order of 10^{-06} has been set on nodal pressure vector between two successive iterations. The developed numerical model has been validated (in ref. [15]) with numerical results from Fatima et al. [14]. After validation, the numerical results are computed in terms of non-dimensional physical quantities such as pocket pressure ($\bar{p}_{oc}$), load carrying capacity ($\bar{F}_o$), fluid film stiffness ($\bar{S}$) and damping ($\bar{C}$) coefficients.

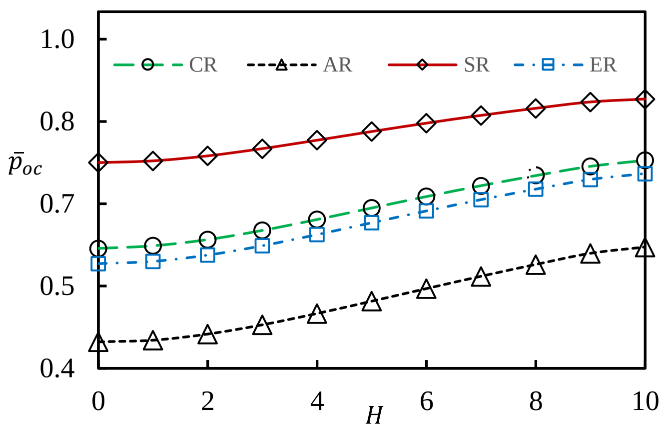

Fig.1 Pocket Pressure vs. Hartmann number

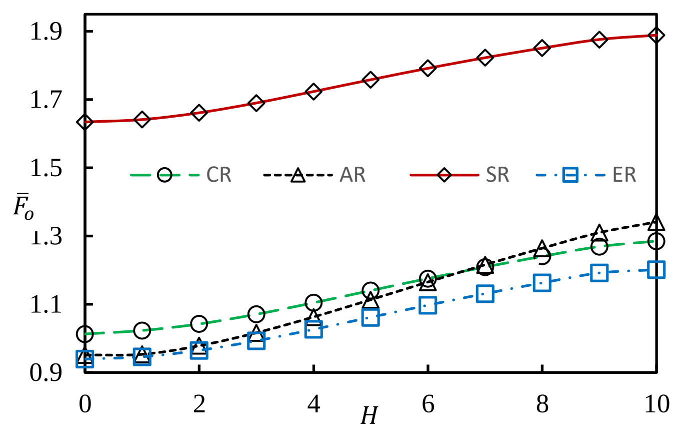

Fig.2 Load carrying capacity vs. Hartmann number

Figure 1 present the influence of recess geometric shape and strength of magnetic field on the numerical value of pocket pressure. It has been observed that as the value of Hartmann number/strength of magnetic field increases progressively, higher value of pocket pressure and pressure gradients are observed in pocket and land area respectively. Presence of magnetic field in electrically conducting lubricant induced Lorentz forces on the fluid particles. As a result of Lorentz forces, the velocity profile of lubricant shrinks, leading to accumulation of higher quantity of lubricant between the bearing surfaces which will eventually leads to higher fluid film pressure in pocket and land area. This increment in fluid film pressure have a direct effect on the load carrying capacity of bearing. As can be seen from Fig.2, there have been an increase in load carrying capacity with increase in Hartmann number. However, fluid film pressure and its gradient also depend on the geometric shape of recess. Therefore, increment in $\bar{F}_o$ owing to presence of magnetic field is different for each recess shape. Maximum and minimum load carrying capacity is observed for bearing with sectorial and elliptical recess shape respectively. However, a maximum of 43.25% increment in $\bar{F}_o$ (due to presence of magnetic field) has been reported for annular recess bearing. Figure 3 depicts variation in the fluid film stiffness coefficient w.r.t to the strength of magnetic field. It can be noticed that presence of magnetic field have an adverse on

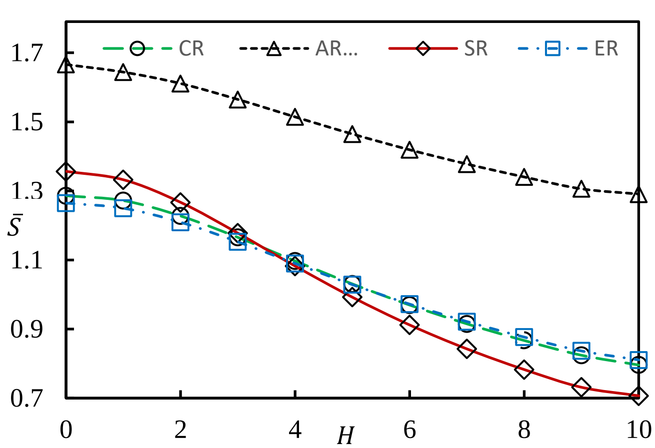

Fig.3 stiffness coefficient vs. Hartmann number

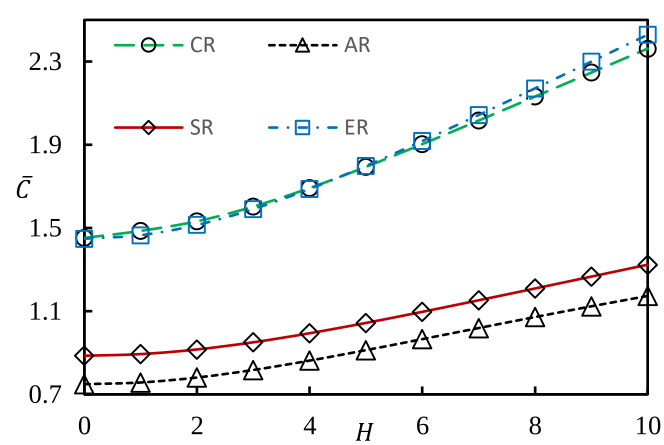

Fig.4 damping coefficient vs. Hartmann

the stiffness capabilities of the bearing, irrespective of geometric shape of recess. Among various recess shapes employed, the annular recessed bearing is observed to be providing maximum value of fluid film stiffness coefficient, invariant to the strength of magnetic field. Figure 4 presents variation in fluid film damping coefficient w.r.t to Hartmann number. It can be noticed that presence of magnetic field substantially enhance the damping coefficient of the bearing. This enhancement in damping capabilities of bearing is again noticed to be a strong function of geometric shape of the recess. Quantitatively, a maximum of 67.7% increase in damping coefficient has been noticed for elliptical recessed hydrostatic thrust pad bearing, owing to the presence of magnetic field.

4. Conclusions

Presence of magnetic field is observed to have a profound influence on the static and dynamic performance of the capillary compensated hydrostatic circular thrust pad bearings operating with electrically conducting lubricant. Application of magnetic field is noticed to be improving load carrying capacity and damping abilities of the bearings. The fluid film pressure and its gradient strongly depend on geometry of recess and thrust pad. Therefore, to achieve maximum gain in static and dynamic performance by application of magnetic field, a proper selection of geometric shape of recess is essential. Under simulated condition, the presence of magnetic field had enhanced the load carrying capacity for all recess shapes considered in the bearing. However, maximum increment (43.25%) in load carrying capacity has been observed for annular recess bearing. Use of electrically conducting lubricant vis-à-vis Newtonian lubricant is also noticed to be significantly enhancing the damping capabilities (67.7%, elliptical recess) of the hydrostatic thrust pad bearing.

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