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RESEARCH ARTICLE

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EXPERIMENTAL INVESTIGATION OF PRESSURE PROFILE AND STABILITY OF SHAFT IN MAGNETORHEOLOGICAL FLUID HYDRODYNAMIC JOURNAL BEARING

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ABSTRACT

The Magnetorheological fluid varies its apparent viscosity when it's comes under the magnetic field. The experimental set up is prepared to study the effect of varying magnetic field on the pressure profile in Magnetorheological fluid hydrodynamic bearing. The Magnetorheological fluid bearing is tested for the different eccentricity ratios. For analytical calculations, the viscoplastic lubrication theory for Heschel-Bulkley fluid model is considered. The stability of the shaft is also analyzed when the Maagnetorheologial effect is on and off.

Key Words— Magnetorheological fluid, Hydrodynamic journal bearing, Herschel-Bulkley model, shear stress, viscoplastic lubrication theory.

The hydrodynamic lubricated bearings are common in industry applications. The oil film produced by journal rotation is the key factor for generating forces to support the shaft load and avoid touching the shaft to the bearing surfaces. The theories and experiments show that the one crucial property of the lubricant itself is in determining load carrying capacity of bearing is its viscosity. In industrial applications, common lubricants like oil are usually considered to have constant viscosities at constant temperature. Change of some factors like temperature and pressure can lead to the change of viscosity, but it is passive and uncontrollable.

In current industrial applications, it does require that the hydrodynamic bearing should operate in different loading conditions and in different environments. Bearings lubricated with smart lubricants are therefore studied, aiming to achieve the different working loading conditions. The detail investigation, it can be seen that there are three stages of operation in hydrodynamic journal bearing. First stage is during starting, due to low speed there is no hydrodynamic action. For second stage, during the running, due to high speed, there is hydrodynamic action. And for the third stage, during stopping, there is low hydrodynamic action, due to decreasing speed.

Due to no hydrodynamic actions, during starting and stopping, there is possibility of metal to metal contact and during the running condition, the oil viscosity decreases due to temperature rise and also possibilities of metal to contact; this can be avoided by changing the viscosity of MR fluid. Because of their quick response and reliability, electrorheological (ER) fluid and magnetorheological (MR) fluid are mostly used as smart lubricant. When there is no external electric field or magnetic field (off state), both fluids can be treated as Newtonian fluids. When external electric or magnetic field is applied, these two fluids exhibit viscoplasticity, categorizing them as non-Newtonian fluids. The alteration of viscosity in two different states could thus be used to actively control both static and dynamic behavior of hydrodynamic bearings. However, since the application of ER fluid requires very high currents and the yield stress of it at on state is relatively low, MR fluid is more applicable.

In present study, by using viscoplastic lubrication theory, the load carrying capacity of bearing determined by analyzing shear stresses, pressure profile and verified with the experimental results.

Abbreviations

B	-Bingham number
B_L	-Width of bearing (m)
C_b	-Radial clearance (m)
D	-Integral region
e	-Eccentricity ratio
h	-Oil film thickness (m)
H	-Characteristics length in Y- direction (m)
K	-Herschel-Bulkley model parameter
L	-Characteristics length in X-direction (m)
n	- Herschel-Bulkley model parameter
P	- Oil film pressure (Pa)
Q	- Dimensionless flux at θ=0
R_b	- Bearing radius (m)
R_j	- Journal radius (m)
t	- time (s)
U	- Velocity in X-direction (m/s)
x_0	- Point at which oil film pressure and pressure gradient both reach zero
θ	- Bearing Angle (rad)
θ_0	- Point at which oil film pressure and pressure gradient both reach zero (rad)
τ	- Shear stress (Pa)
τ_0	- Yield shear stress
μ	- Dynamic viscosity of Newtonian fluid (Pa.s)
σ	- Indicative Shear stress (Pa)
1	- Subscript, indicates the inner surface of bearing
2	- Subscript, indicates the journal surface

Literature Review

Cohen and Oren [8] and Milne [9] to investigated the grease lubricated bearings with plug formation in the oil film. Wada et al. [7] described theory for the core behavior in a finite length bearing and then applied to a one dimension situation. Tichy [6] developed a simplified Bingham-based Reynolds equation to model behavior of Bingham fluid in thin films, and then applied it to journal bearing and squeeze film damper. Nikolakopoulos [1] analyzed a ER fluid lubricated bearing experimentally and analytically in a macroscopic way, and concluded that ER fluid could be used as smart bearing lubricant to control vibration. Hessbelch and Abel-Keilhack [4] explained MR fluid lubricated hydrostatic bearing. Gertzos et al. [3] and Bompos and Nikolakopoulos [2] used the CFD codes to xplain static characteristics of the bearing. Urreta et al. [10] differentiate between ER fluid and MR fluid and concluded that at low speed MR fluid achieved good results compared to ER fluid in hydrodynamic bearing. Amalraj [11] analyzed a pressurized thrust bearing lubricated with MR fluid using HB model, taking into consideration of inertia effect. He concluded that high HB model numbers enhanced the bearing performance, and the fluid inertia effect was only significant when HB model numbers were low. Hewitt and Balmforth [5] developed a general viscoplastic lubrication theory with HB model and applied it to bearing and washboard instability of a plate, achieving good agreement with experiments.

Objectives

The objectives of this research work is-

- To investigate the use of the smart lubricant in hydrodynamic bearing in which viscosity can be varied according to the requirement.
- By varying the viscosity, to vary the load carrying capacity and speed of bearing.
- To improve overall life of the bearings.

Scope

As smart lubricant various lubricants are available, they are also known as non-Newtonian fluids. Following are the various non Newtonian fluids available –

- Ferro fluid
- Electro rheological fluid
- Magneto rheological fluid

All these fluids vary their apparent viscosities as per the electric and magnetic field variation. The scope of work limited to Magneto rheological fluid to be used in hydrodynamic journal bearing.

Methodology

The Magneto rheological fluid behaves like non Newtonian fluid and follows the two types of model.

- Bingham plastic flow model
- Herschel-Bulkley fluid model.

In this research work, the Herschel-Bulkley fluid model is considered in hydrodynamic journal bearing. The fluid model is investigated theoretically and validates it by experimentally. The viscoplastic lubrication theory is used for the analysis.

Analytical investigation

In order to simplify analysis and calculation, the following assumptions are made, as suggested by Hewitt and Balmforth [5].

- The MR fluid is incompressible;
- Body forces are ignored, and
- Variables (pressure, strain, stress, etc.) along axial direction are constant.

Under these assumptions, in the oil film coordinates system shown in Figure 1.

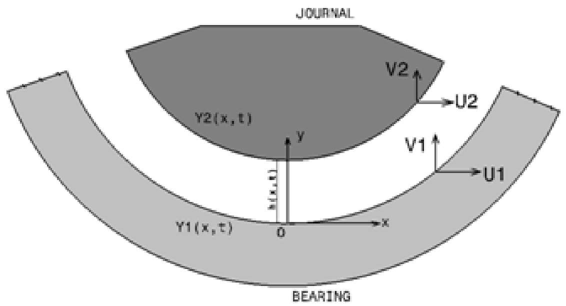


Figure 1 - Geometry of part of the oil film of a journal bearing.

For the further analytical investigation, the derivations and formula suggested by Hewitt and Balmforth [5], is used.

The Herschel-Bulkley model can be represented as:

$$\tau = \tau_0 + K\frac{du^n}{dy} \qquad (1)$$

Where τ is the shear stress, τ_0 is the yield stress , du/dy is the shear rate, and K, n are model constants.

The relationship given by Hewitt and Balmforth [5], used for the pressure analysis distribution.

$$\frac{U}{h}=\frac{n\,(s_2\,\sigma_2^{\frac{n+1}{n}}-s_1\,\sigma_1^{\frac{n+1}{n}})}{(n+1)[(s_2-s_1)B+s_2\sigma_2-s_1\sigma_1\,]} \qquad (2)$$

$$\frac{2(n+1)}{nh^2}(Q-q)=\frac{n\left(s_2\,\sigma_2^{\frac{2n+1}{n}}-s_1\,\sigma_1^{\frac{2n+1}{n}}\right)}{(2n+1)[(s_2-s_1)B+s_2\sigma_2-s_1\sigma_1]^2}-\frac{(\sigma_2^{\frac{n+1}{n}}-\sigma_1^{\frac{n+1}{n}})}{[(s_2-s_1)B+s_2\sigma_2-s_1\sigma_1\,]} \qquad (3)$$

Where

$\sigma_1=|\tau_1|-B\,,\ \sigma_2=|\tau_2|-B\,,$

$s_1=\mathrm{sgn}(\tau_1),\ \ s_2=\mathrm{sgn}(\tau_2),$

By defining the Reynolds boundary conditions, the pressure can be determined for given value of Q, by using the Reynolds boundary condition, therefore, can be represented as:

$$\int_D \frac{dP}{dx}d\bar{x}=\int_D \frac{\tau_2-\tau_1}{h}=0 \qquad (4)$$

$$h = c_b(1+e\cos\theta) \qquad (5)$$

The above equations are directly used for the calculations of pressure and pressure gradient analytically.

experimental Procedure

For experimental verification, the bearings dimensions are taken as:

Journal Radius = R_j = 0.04m

Length of bearing = B_L=0.08m

Clearance = $c_b = 8\times10^{-5}$m

Herschel-Bulkley model parameter= K= 35×10^{-3}

Speed= 500 R.P.M.

The rest of the fluid properties are taken from supplier of Lord MRF-132 fluid.

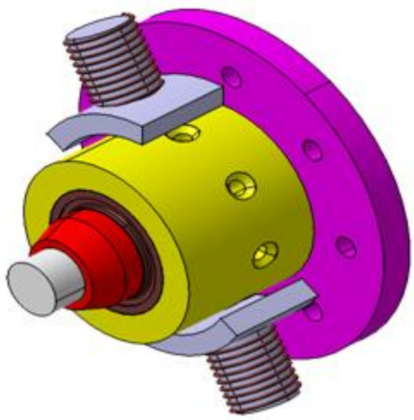


Figure 2 – Cad model concept of Magneto-Rheological fluid bearing.

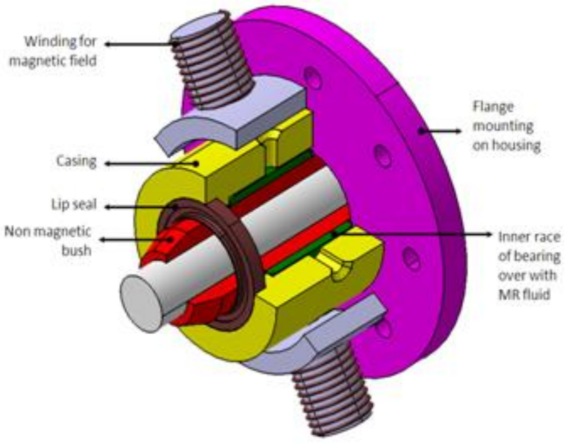


Figure 3 - Cross sectional view of MR fluid bearing.

As shown in Figure 3, the bearing is mounted by using flange, the magnetic winding is supported on different frame the lip seal is provide on both side to avoid leakage and intrusion of foreign particles into MR fluid. The MR fluid is supplied by the Perealistic pump at pressure of 1×10^5 to 3×10^5 Pa.

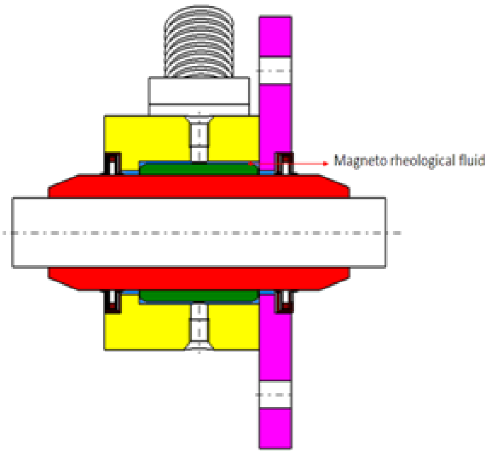


Figure 4 – Magnetorheological fluid in Bearing.

The MR fluid is entering through small hole which is exactly in the center width of the bearing at three different locations. The MR Fluid is supplied by peralistic pump, which is particularly designed for non Newtonian fluid and its commercially available. The flow rate measurement is continuously monitored by the flow meter.

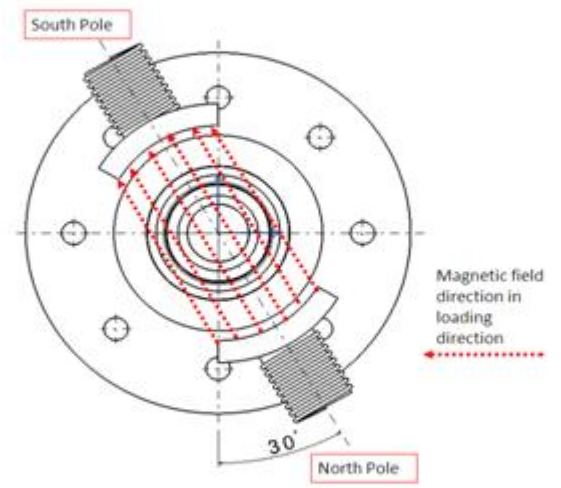


Figure 5 – Magnetic field direction along the attitude angle, so that magnetic particle aligns in loading direction.

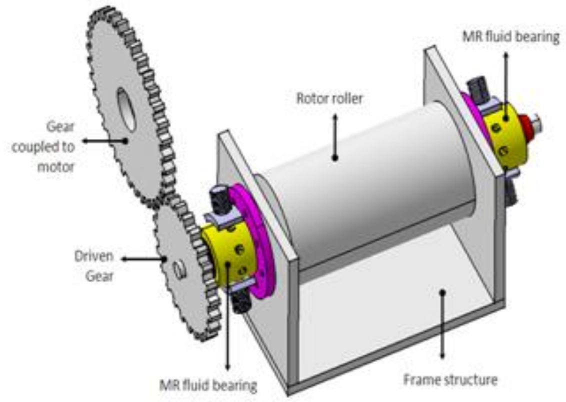


Figure 6- Experimental set up for the MR fluid bearing (Indicative arrangement of actual experiment set up)

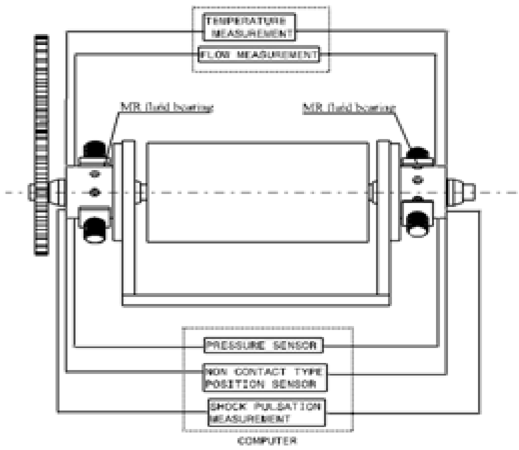


Figure 7 – Typical arrangement of all the sensors.

On the MR fluid bearings various types of sensors used. Pressure sensors are used along the circumference to measure the pressure. The shock pulsation sensor measurement device used to measure the if there is any metal to metal contact, it will be get recoded by the shock pulsation measurement device, this is especially required to see at the starting, is there any metal contact or not, when there is high value of yield shear stress. The non contact laser type position sensor is used to find out the stability of the shaft, there is one point marked on the shaft end faces, that point is traced by the non contact position sensor to determine the stability of the shaft, when MR effect is on and off. The hydrodynamic bearing set up is used for testing, as shown in Figure 6, on the frame two MR fluid hydrodynamic bearings are mounted for testing. The extra weight attached onto the cylinder. All the data measured from different sensors (as shown in Figure -7) feed to the computer for analysis. All the data, of different bearings get recorded separately on computer.

Results and discussions

The equations mentioned in numerical investigation are solved numerically. First, Equations (2) and (3) are solved for τ_1 and τ_2 using assumed flux Q. Then Q is solved on trial and error basis, based on Reynolds boundary condition Equations (4), where $\bar{x}_0$ is the point at which $\tau_1 = \tau_2$. Every value of Q is used to solve Equation (2) and (3) in the first step and the iteration goes on.

When the values all values of shear stress, discharge, and pressure gradient are converged, the static condition of the bearing is determined. From the given dimension of bearing, the analytical values are calculated for shear stress and pressure at different eccentricities when n=0.75.

Table 1- Analytical data of shear stresses, pressure for different eccentricity ratios.

Eccentricity ratio	Speed (R.P.M.)	τ_1 (N/m2)	τ_2 (N/m2)	Pressure (Pa)
0.2	500	4	-0.5	5
0.6	500	12	-6	50

For the analytical data it will be assumed that the pressure and pressure gradient will be acting in positive pressure region only that is from 0 to 180 degree. The negative pressure and pressure gradient is ignored in this case.

From the table 1 analytical data, the pressure curve is plotted along the circumference. The experimental values are taken by 12 pressure sensors which are placed at an equal angle of 30 deg. along the periphery of the bearing. The comparison between analytical pressure curve and experimental pressure curve as in the Figure 8.

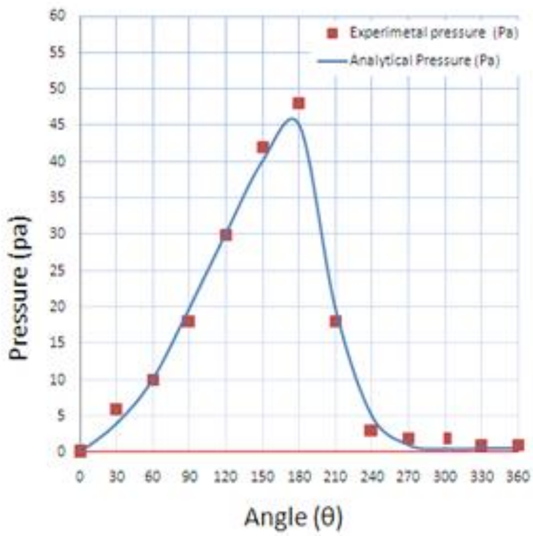


Figure 8 – Oil film pressure variation along the circumference when eccentricity ratio e=0.6.

From Figure 8, it shows that the analytical results are in certain agreement with experimental results.

By increasing the electromagnetic field current the yield stress value τ_0 get increased. Following table shows the effect of yield stress on pressure when other parameter remains constant, For Speed N=500 R.P.M., Eccentricity ratio e=0.6, and n=0.75. All the readings are taken after 30min running of bearing and stabilization of the bearing.

Table 2 – Effect of yield stress variation on Pressure.

Yield stress (Pa)	Analytical Pressure (MPa)	Experimental Pressure (MPa)
100	3.15	2.9
200	3.28	3.4

300	3.48	3.5
400	3.62	3.4
500	3.80	3.9
600	4.05	4.2
700	4.23	4.1
800	4.41	3.9
900	4.59	4.6
1000	4.8	4.7

All the experimental data are within accuracy of +/-15 percent to analytical values. From table 2, it is clear that, as the yield stress increased, the pressure and load carrying capacity of bearing also get increased.

From figure 9, shows the pressure distribution along the circumference at different yield stress values.

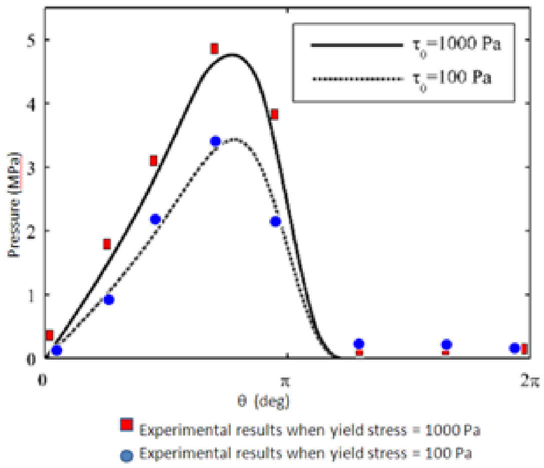
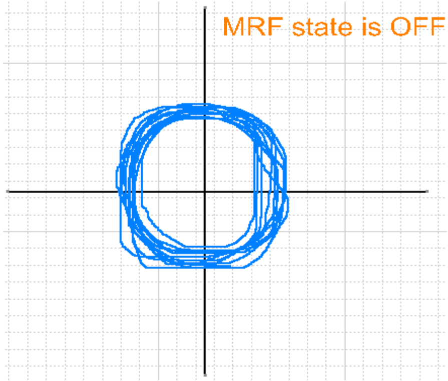


Figure 9 - Oil film pressure variation for different values of yield stress along the circumference for eccentricity ration e=0.6



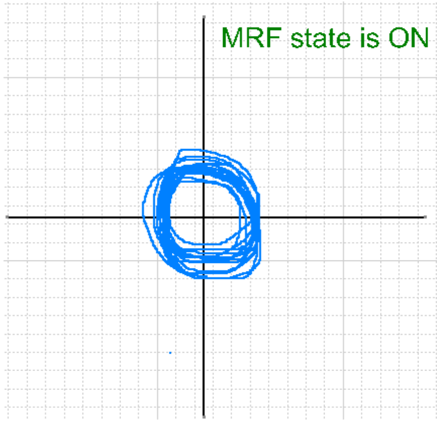


Figure 10 – Journal orbit position from position sensor data when MRF state effect is on and off.

From figure 10, the tracing circles are the results from position sensor data. When MR effect is OFF state, there are bigger circular diameter lines, which show instability of the shaft in bearing, but when the MR effect is ON state, the circular lines are smaller in diameter, this shows that the shaft get more stable due to yield stress.

Conclusion

As the yield stress values increases, the apparent viscosity increases, and ultimately the load carrying capacity of the bearing also increases, due to increase in pressure. During the testing, higher values of yield stress maintained at the starting, to check is there any metal contact, as there is no signal from shock pulsation measurement, there was no metal to metal contact. So, there is no wear of bearing surface. From position sensor data, it is clear that, the shaft is more stable when MR effect is ON state. The calculated pressure profile is experimentally verified. It can be concluded that, by using MR fluid the hydrodynamic bearing performance get improved.

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