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Magnetic Fluid Lubrication of Rough Thrust Bearings considering a Sine Film Profile

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Abstract. The purpose of this research article is to study theoretically the influence of transverse surface roughness on the performance of a magnetic fluid based sine film profile in thrust bearings. In many lubrication situations it is required to place the lubricant at a desired position and then retain it there. By applying an external magnetic field, ferrofluids can be confided, positioned, shaped and, controlled at desired places. To explore the effects of magnetic fluid to the bearings, Neuringer-Rosensweig model has been invoked. As the affections of the surface roughness to the bearing, the stochastically averaged Reynolds equation has been solved with appropriate boundary conditions to achieve the appearance for film pressure, which is then used to calculate the load support. The simulated results based on a non-Newtonian lubricant with surface roughness has been presented in terms of fluid film thickness (inlet and outlet ratio), fluid film pressure and load support. The presented results have valuable in case of transversely rough surface compensated with load support. In addition, it will provide more messages in bearing selection and designing.

Keywords: Sine film profile, Roughness, Magnetic fluid

1 Introduction

Amongst the hydrodynamic bearings, thrust bearing is the simplest and frequently encountered because, the expression of film thickness is simple and boundary conditions to be required zero at the bearing ends are not complicated. Moreover, it is used in various fields likes clutch plates, automobile transmissions and domestic appliances. Lubrication performances of thrust bearings with an incline plane profile have been discussed by many investigators (Pinkus and Sternlicth[1], Bhusan[2] and Khonsari and Booser[3]).Launder and Leschziner[4] studied the inertia effects on thrust bearings by applying the method of integral-differential. It is found that the fluid inertia effects result in an increase in the film pressure and load support.

As man instigated to walk around space, it became relevant to build up efficient techniques to use and store rocket engine propellants under zero gravity conditions. For this reason the NASA Research Center developed in the 1960s a kerosene based ferro fluid that could be accumulated at a craving location by the use of a magnetic field. Magnetic fluids are stable colloidal suspensions of magnetic metal nana particles in a carrier liquid such as hydrocarbon, dieter, water, mercury. The study and applications of colloidally stable magnetization fluids as an active discipline is about approximately 57 years old and cross many disciplinary lines. Magnetic fluids are well controlled by an external magnetic field that gives broad possibilities for technical and biomedical applications. Examples of possible applications are microscopy wafer/chip inspection and pick and place machines. More recently, its application in magnetic fluid hyperthemia(MFH). MFH promises to be available alternative in the treatment of localized cancerous tumors.

Chawala[5] investigated the magnetohydrodynamic inclined slider bearing with an azimuthally magnetic field. It was found that capacity of lad increased significantly. Zahn and Rosenweigh [6]described the motion of magnetic fluids through porous media under the influence of obliquely applied magnetic field. This paper became the basis for many research analyses dealing with the performance of porous bearings working with a magnetic fluid as lubricant. Use of ferro fluid as lubricant modifying the performance of the bearing has now been recognized.Huang and Tian [7] deal with the problem using a new derivation to learn the steady performance of hydrostatic thrust bearing considering Rabinowitch fluid model. Here, basic parameters were studied to obtain fluid pressure and load capacity under deliberation of Gauss-Legendre integral formulae. Moreover, findings were well approved with the results of Singh et al.[8]

The roughness profile of metal-to-metal contacting parts is one of the most important characteristics in making mechanical system more durable and energy efficient. For life span of the bearing, the roughness aspects must be considered carefully while designing the system. One of the first investigation was made by Patir and Cheng [9] who proposed an ensemble averaged Reynolds equation in which roughness effects were built into a number of flow factors. The method derived here was more versatile than stochastic theories, and also applicable for three dimensional structure of roughness. Zhang et al. [10] outlined a new approach for optimizing the film shape of rough slider bearing using the multiphysics software with the method of moving asymptotes.

As a result, the roughness with a magnetic fluid lubricant both together plays vital role for increasing the life period of bearing. Shukla and Deheri [11] illustrated a theoretical analysis of the effect of surface roughness on tilted pad bearing lubricated with a magnetic fluid. Even, if the effect of transverse roughness was adverse in general, this investigation offered some indications for obtaining better performance in the case of negatively skewed roughness by suitably choosing the magnetic strength. Recently, Shukla and Deheri [12] extended the above analysis under the influence of porosity by taking in to account slip effect. Although, there are many parameters reducing the bearing load, the ferro fluid lubrication turns in a better situation.

2 Derivation of the Mathematical Model

One problem is to decide which way to make the runner move. In the given problem, the runner has to move towards the X-direction with velocity U. The physical configuration of the bearing structure is represented in Fig.1.

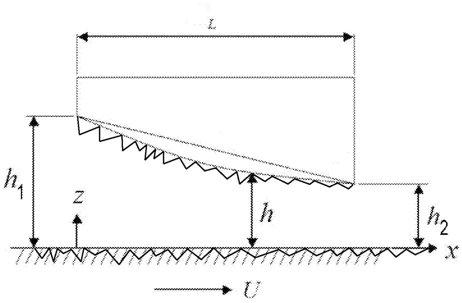

Fig. 1. The geometry configuration of the bearing system

$$h(x) = h_2 + h_t(x) \tag{1}$$

where the sine film profile is expressed by

$$h_t(x) = (h_1 - h_2)[1 - t(\frac{\pi(0.5)x}{L})] \tag{2}$$

Before deriving the full equation the assumptions that are to be made must be considered. These can be listed and commented on as follow: 1.Body forces are neglected, i.e. there are no extra field of forces acting on the fluids.

2.The pressure is constant through the thickness. As the film is only one or two thousands of an inch thick it is always true.

3.The curvature of surface is large compared with film thickness. Surface velocities need not be considered as varying in direction.

4.***There is no slip at the boundaries.***

5.The lubricant is iso-viscous Newtonian, i.e. stress is proportional to rate of shear.

6.Flow is laminar

7.Fluid inertia is neglected.

8.The viscosity is constant through the film thickness.

With these assumptions the development of the equations can start. The Reynolds equation for the lubricant flow in the thrust bearing as usually quoted by Lin[13],

$$\frac{d}{dx}(h^3\frac{dP}{dx}) = 6\eta U\frac{dh_t}{dx} \tag{3}$$

The magnitude of the magnetic field if given by considering $k = 10^14A^2m^-4$ (Bhat[14])

$$H_1^2 = kx(x - L) \tag{4}$$

By applying Neuringer and Rosensweig [15] model based ferrofluid lubrication transfer to equ. (3)

$$\frac{d}{dx}(P - \frac{\mu_0\mu^- H_1^2}{2}) = \frac{6\eta U}{h^3}\frac{dh_t}{dx} \tag{5}$$

Now, with the aid of stochastic average model of Christensen and Tonder [16-18], one is inclined to obtained

$$\frac{d}{dx}(P - \frac{\mu_0\mu^- H_1^2}{2}) = \frac{6\eta U}{a(h)}\frac{dh_t}{dx} \tag{6}$$

where $a(h) = h^3 + 3h^2\alpha + 3(\alpha^2 + \sigma^2)h + \epsilon + 3\sigma^2\alpha + \alpha^3$ In order to analyze the bearing performance, it is conveniently to introduce the non-dimensional schemes. $x^* = \frac{x}{L}$, $H = \frac{h}{h_2}$, $H_t = \frac{h_t}{h_2}$, $\beta = \frac{h_1}{h_2}$ $P^* = \frac{Ph_2^2}{\eta UL}$, $W^* = \frac{Wh_2^2}{\eta UL^2B}$ $\epsilon^* = \frac{\epsilon}{h_2^3}$, $\sigma^* = \frac{\sigma}{h_2}$, $\alpha^* = \frac{\alpha}{h_2}$, $\mu^* = \frac{k\mu^-\mu_0 Lh_2^2}{\eta U}$ $A(h) = \frac{a(h)}{h_2^3}$

Now, non-dimensional film height in both the form can be represented from equ. (1) and (2) are:

$$H(x^*) = 1 + H_t(x^*) \tag{7}$$

$$H_t(x^*) = (\beta - 1)[1 - t(\pi 0.5x^*)] \tag{8}$$

The boundary conditions for the film pressure:

$P^* = 0$ at the inlet position $x^* = 0$

$P^* = 0$ at the outlet position $x^* = 1$ Integrating the dimensionless Reynolds equation and using given appropriate boundary conditions, one can achieve the fluid film pressure in non-dimensional form,

$$P^* = \frac{\mu^*}{6}x^*(x^* - 1) + 6P_1 + c_1P_2 \tag{9}$$

where c_1 is an integral constant, and

$$P_1 = \int_o^{x^*} \frac{H_t}{A(h)}dx^* \tag{10}$$

$$P_2 = \int_o^{x^*} \frac{1}{A(h)}dx^* \tag{11}$$

Then expression for non-dimensional load support is given by

$$W^* = \int_o^1 P^* dx^* \tag{12}$$

Applying the expression of the film pressure and integrating the equation, the dimensional less supporting load can be obtained by

$$W^* = \frac{\mu^*}{12} + 6W_1 + c_2 W_2 \tag{13}$$

where c_2 is an integral constant, and

$$W_1 = \int_o^1 P_1 dx^* \tag{14}$$

$$W_2 = \int_o^1 P_2 dx^* \tag{15}$$

3 Graphical Results and Discussion

It is put forward from equ. (9) that the non-dimensional pressure increases by $\frac{\mu^*}{6}$ while equ. (13) establishes that the dimensionless load support gets enhanced by $\frac{\mu^*}{12}$. It is manifested that the expression for the load support is linear with respect to μ^*, and hence the non-dimensional load increases with respect to μ^*. The magnetization parameter results in an improved performance because it increases the viscosity of the lubricant leading to an increase in pressure; thus giving increased load support as display in Fig. (2)-(5) and tabular form (a).However, for smaller values of the standard deviation, the effect on load support with respect to the magnetization parameter can be raised sharply as in the Fig.2. The effect of standard deviation is negligible up to 0.05.

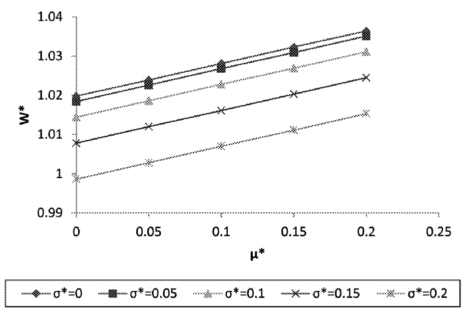

Fig. 2. Deviation of supporting load with μ^* and σ^*

The effect of standard deviation on the variation of load support is summarized in Fig. (6)-(8) as well as tabular form (b) and (c).It is easily noticed that the supporting capacity of load decreases considerably owing to the standard deviation. Holding up the motion of the lubricant due to the surface roughness

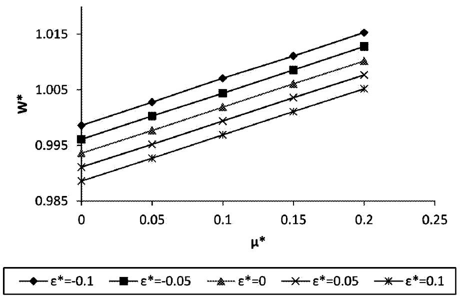

Fig. 3. Deviation of supporting load withμ^* and ϵ^*

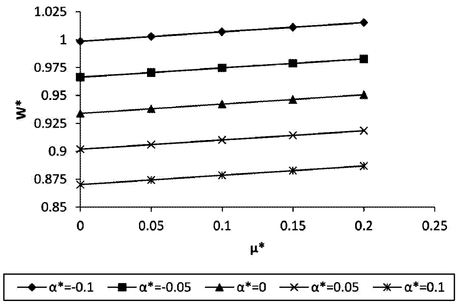

Fig. 4. Deviation of supporting load with$\mu^*$and α^*

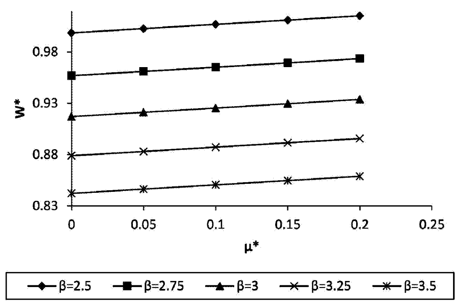

Fig. 5. Deviation of supporting load withμ^*andβ

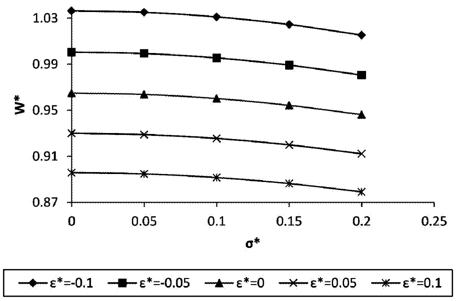

Fig. 6. Deviation of supporting load with $\sigma^*$and ϵ^*

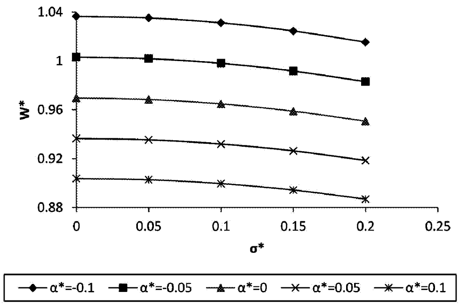

Fig. 7. Deviation of supporting load with $\sigma^*$and α^*

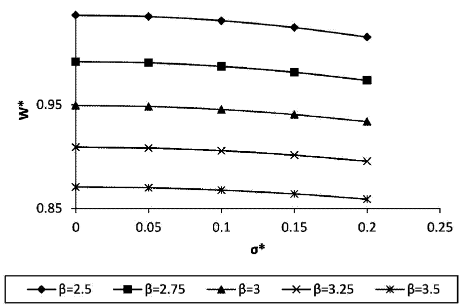

Fig. 8. Deviation of supporting load with $\sigma^*$and β

causes abridged pressure resulting in decrease the load. The skewness remains a very crucial parameter which has a profound impact on the behavior the bearing system. Furthermore, it can be visualized that the combined effect of negative skewness and variance negative is significantly positive in most of the situations. This effect has been depicted in Fig. (9) and (10). The variance is the most com-

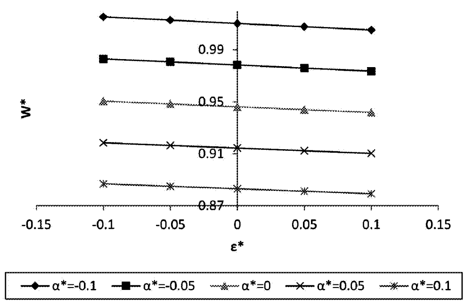

Fig. 9. Deviation of supporting load with $\epsilon^*$and α^*

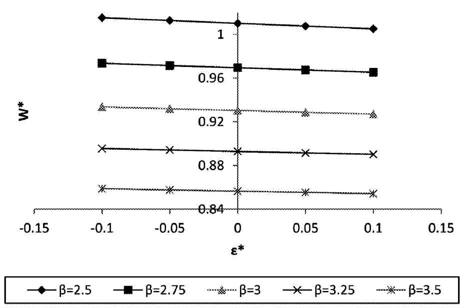

Fig. 10. Deviation of supporting load with $\epsilon^*$and β

monly specified parameter for surface finish measurement. The effect of mean and ratio is displayed in Fig. (11).

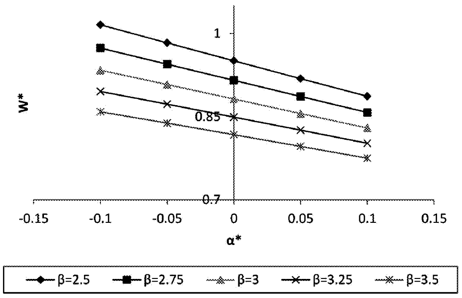

Fig. 11. Deviation of supporting load with $\alpha^*$and β

Tabular form (a):variation of load support with $\mu^*$and α^*					
	$\alpha^* = -0.1$	$\alpha^* = -0.05$	$\alpha^* = 0$	$\alpha^* = 0.05$	$\alpha^* = 0.1$
0	0.9986	0.9663	0.9339	0.9018	0.8701
0.05	1.0028	0.9704	0.9381	0.9060	0.8743
0.1	1.0071	0.9746	0.9423	0.9102	0.8785
0.15	1.0111	0.9788	0.9464	0.9143	0.8826
0.2	1.0153	0.9829	0.9506	0.9185	0.8868

Tabular form (b):variation of load support with $\sigma^*$and ϵ^*					
	$\epsilon^* = -0.1$	$\epsilon^* = -0.05$	$\epsilon^* = 0$	$\epsilon^* = 0.05$	$\epsilon^* = 0.1$
0	1.0364	1.0005	0.965	0.9301	0.8959
0.05	1.0351	0.9993	0.9638	0.9289	0.8948
0.1	1.0311	0.9955	0.9602	0.9256	0.8916
0.15	1.0245	0.9892	0.9543	0.9200	0.8864
0.2	1.0153	0.9806	0.9462	0.9123	0.8792

Tabular form (c):variation of load support with$\sigma^*$and β					
	$\beta =$ 2.5	$\beta =$ 2.75	$\beta =$ 3	$\beta =$ 3.25	$\beta =$ 3.5
0	1.0364	0.9916	0.9493	0.9091	0.8706
0.05	1.0351	0.9905	0.9483	0.9082	0.8699
0.1	1.0311	0.9870	0.9454	0.9057	0.8676
0.15	1.0245	0.9813	0.9405	0.9014	0.8640
0.2	1.0153	0.9735	0.9337	0.8955	0.8589

4 Conclusion

One of the conclusions of this study is that even if there is a suitable magnetic field, the roughness must be treated carefully. The present optimal geometries are rather easy to manufacture, which makes them an attractive choice for the design of thrust bearings operated in a wide range of rotor speeds.

From this investigation, remarkable resorts have been pointed out:

* The magnetization has a limited option in lowering the adverse effect of roughness.

* Although, there are many parameters reducing the bearing load, the ferro fluid lubrication turns in a better situation when the effect of standard deviation is at considerably low level.

* The inlet and outlet fluid film height ratio plays an important role for improving the performance of the bearing system.

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Table 1. Notation

symbol	Meaning
h	local film height
t	sine film profile
h_1	inlet film height
h_2	outlet film height
B	width of the pad
L	length of the bearing
P	pressure of fluid film
U	Velocity of the runner
W	load support
H_1	magnitude of magnetic field
P^*	non-dimensional pressure
W^*	non-dimensional load support
σ	standard deviation
ϵ	skewness
α	variance
β	film thickness ratio
η	viscosity of lubricant
μ^*	magnetization parameter
σ^*	dimensionless standard deviation
ϵ^*	dimensionless skewness
α^*	dimensionless variance
μ^-	magnetic susceptibility
μ_0	permeability of the free space