Shliomis Model Based Magnetic Jluirf Lubrication OjA Squeeze Jilm In Rotating Rougfi Curved Circular Plates

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ABSTRACT:
The study of ferrofluid-based squeeze film performance on revolving rough circular plates using the Shliomis model. When it comes to rough surfaces, researchers use the Christophersen and Tonder stochastic model. The load-carrying capacity is derived from the pressure distribution by solving the stochastically averaged Reynolds type equation. The Shliomis model based on ferrofluid lubrication is preferable than the Neuringer-Rosensweig model for this kind of bearing system. The rust can be reduced by curvature factors, at least in the case of negatively skewed roughness.
Keywords: Shliomis Model, Magnetic Jluirf, Squeeze Jilm, Rotating Rougfi
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INTRODUCTION:
Exhaustive study reports that a brief discussion of the manufacturing and stability difficulties of magnetic colloids was held. [1-3] In this paper, many theoretical and experimental research on the effects of a magnetic field on suspension equilibria and the nature of their motion were discussed. An anisotropy in both viscosity and magnetism, as well as a rotation of the particles, were all taken into consideration while calculating the kinetic coefficients. [4-7] When a film was squeezed between two circular discs, researchers in 1980 investigated how it behaved under a uniform magnetic field. In order to increase the bearing system's performance, it was shown that the slip parameter deserved to be reduced. [8-10]

ANALYSIS:
Each plate in the bearing, has an a-radius. The z-axis is rotated by the angular velocities £2t and ilu of the bottom and higher plates. With a constant hQ velocity, the top disc approaches the bottom disc properly.

Transverse roughness is assumed for the bearing surfaces. It also used stochastic modelling of roughness to determine the film thickness h of the lubricating film.

$$h = h + hs \qquad (1)$$

hs is the departure from the mean film thickness that characterises the random roughness of the bearing surfaces. The probability density function governs the deviation hs.

$$f(h_s) = \begin{cases} \frac{35}{32c^7}\left(c^2 - h_s^{\,2}\right)^3, -c \le h_s \le c \\ 0 \qquad\qquad\qquad , \ elsewhere \end{cases}$$

A departure of c from the mean film thickness is considered significant. There are three relationships that characterise the random variable hs's symmetry: its mean a, standard deviation a, and parameter $.

$$\alpha = E(h_s), \sigma^2 = E[(h_s - \alpha)^2], \varepsilon = E[(h_s - \alpha)^3]$$

where *E* denotes the expected value defined by

$$E(R) = \int_{-c}^{c} R f(h_s) dh_s$$

The details can be seen from Christensen and Tonder (1969a, 1969b, 1970). *

It is assumed that the rotating upper plate lying along the surface determined by the relation

$$z_u = h_0 exp(-\beta r^2);\ 0 \le r \le a$$

approaches with normal velocity *h0* to the rotating lower plate lying along the surface given by

$$z_l = h_0[exp(-\gamma r^2) - 1];\ 0 \le r \le a$$

where *(3* and *y* are the curvature parameters of the corresponding plates and *h0* is the central film thickness. The film thickness *h(r)* then is defined by (Bhat (2003), Abhangi and Deheri (2012))

h(r) = *h0[exp(—fir2) — exp(—yr2)* + 1]; 0 < r < *a*

In 1967, Shliomis (1972) noted that when the applied magnetic field changes, magnetic particles in a magnetic fluid may relax in two ways. In the first case, magnetic particles in the fluid rotate, but in the second case, magnetic particles themselves rotate. While the Brownian relaxation time parameters TB explain particle rotation, the intrinsic rotational process is defined by the relaxation time parameters rs. It is not necessary to take into account inertia and the second derivative of 5 in order to calculate the flow. 1709

$$-\nabla p + \eta \nabla^2 \bar{q} + \mu_0 (\bar{M}\ \nabla)\bar{H} + \frac{1}{2\tau_S} \nabla \times (\bar{S} - I\bar{\Omega}) = 0$$

$$\bar{S} = I\bar{\Omega} + \mu_0 \tau_S (\bar{M} \times \bar{H}) \qquad (2)\ \&\ (3)$$

$$\bar{M} = M_0 \frac{\bar{H}}{H} + \frac{\tau_B}{I}(\bar{S} \times \bar{M})$$

where *S* is the internal angular momentum, *I*is the sum of moments of inertia of the particles per unit volume,

$$\bar{\Omega} = \frac{1}{2} \nabla \times \bar{q}$$

together with

$$\nabla\ \bar{q} = 0, \nabla \times \bar{H} = 0, \nabla\ (\bar{H} + \bar{M}) = 0$$

The fluid viscosity in the film area is given by q, with rj denoting fluid viscosity and p0 denoting free

space permeability. [11] His external magnetic field, magnetic susceptibility of the magnetic field, film pressure, and p0 all contribute to q.
By making use of equation (5.3) in equation (5.2) and (5.4), one finds that,

$$-\nabla p+\eta\nabla^2\bar{q}+\mu_0(\bar{M}\,\nabla)\bar{H}+\frac{1}{2}\mu_0\nabla\times(\bar{M}\times\bar{H})=0$$

And

$$\bar{M}=M_0\frac{\bar{H}}{H}+\tau_B(\bar{\Omega}\times\bar{M}) \qquad (6)$$

Neglecting tb ts terms, substitution ofM in above equation, leads to

$$-\nabla p+\left(\eta+\frac{\mu_0}{4}\tau_B\bar{M}\,\bar{H}\right)\nabla^2\bar{q}+\mu_0(\bar{M}\,\nabla)\bar{H}$$

$$+\frac{1}{2}\mu_0\tau_B[\nabla(\bar{\Omega}\,\bar{H})\times\bar{M}+(\bar{\Omega}\,\bar{H})\,\nabla\times\bar{M}-\nabla(\bar{M}\,\bar{H})\times\bar{\Omega}]=0 \qquad (7)$$

From equation (6), it is easily observed that an initial approximation to Mis

$$\bar{M}=M_0\frac{\bar{H}}{H}$$

Substituting the value of *M* on the right side of equation (5.6), a second approximation to A? is found to be

$$\bar{M}=M_0\frac{\bar{H}}{H}+\frac{M_0}{H}\tau_B(\bar{\Omega}\times\bar{H})$$

Again, substituting this value of *M* on the right side of equation (6), third approximation to *M* is availed as

$$\bar{M}\,\bar{H}=M_0H+\frac{M_0}{H}\tau_B^2\{(\bar{\Omega}\,\bar{H})^2-\Omega^2H^2\}$$

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Shliomis' model of cylindrical polar [12] coordinates with uniform magnetic field, where both surfaces are solid but the top one spins, yields a film pressure controlling equation in accordance with

$$\frac{1}{r}\frac{d}{dr}\left(h^3r\frac{dp}{dr}\right)=12\eta_a\dot{h}_0+\frac{3}{10}\rho\Omega_u^2\frac{1}{r}\frac{d}{dr}(r^2h^3)+\frac{3}{320}\frac{N\tau_B^3}{\eta_a^3}\frac{1}{r}\frac{d}{dr}$$

$$\left[h^5r\left\{-\left(\frac{dp}{dr}\right)^3+\frac{27}{616}\rho^3r^3\Omega_u^6+\frac{13}{14}\left(\frac{dp}{dr}\right)^2\rho r\Omega_u^2-\frac{251}{756}\frac{dp}{dr}\rho^2r^2\Omega_u^4\right\}\right]$$

If r| is not included in the modified Reynolds equation when both plates are rotating, as Christensen and Tonder discussed in 1969a, 1969b, 1970 about the modelling of roughness, the modified Reynolds equation when both plates are rotating has the form of

$$\frac{1}{r}\frac{d}{dr}\left(g(h)r\,\frac{dp}{dr}\right) = 12\eta(1+\tau)\dot{h_0}$$

$$+\rho\left(\frac{3}{10}\Omega_r^2 + \Omega_r\Omega_l + \Omega_l^2\right)\frac{1}{r}\frac{d}{dr}\left(r^2 g(h)\ \right)$$

where,

$$g(h) = h^3 + 3h^2\alpha + 3(\sigma^2+\alpha^2)h + 3\sigma^2\alpha + \alpha^3 + \varepsilon.$$

The following non dimensional quantities are adopted

$$\bar{h} = \frac{h}{h_0}, R = \frac{r}{a}, P = -\frac{h_0^3 p}{\eta a^2 \dot{h_0}}, B = \beta a^2, C = \gamma a^2, \bar{\sigma} = \frac{\sigma}{h_0}, \bar{\alpha} = \frac{\alpha}{h_0},$$

$$\bar{\varepsilon} = \frac{\varepsilon}{h_0^3}, \eta_a = \eta(1+\tau), \Omega_r = \Omega_u - \Omega_l, S = -\frac{\rho\Omega_u^2 h_0^3}{\eta \dot{h_0}}, \Omega_f = \frac{\Omega_l}{\Omega_u}$$

The related boundary conditions are

$$P(1) = 0, \left(\frac{dP}{dR}\right)_{R=0} = 0 \qquad (11)$$

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Solving equation (10) in view of boundary conditions (11) tbe expression for non-dimensionalfluid film pressure is found to be

$$P = -6(1+\tau)\int_1^R \frac{R}{g(\bar{h})}dR - \frac{S}{20}\left(3\Omega_f^2 + 4\Omega_f + 3\right)(1-R^2) \qquad (12)$$

where

$$g(\bar{h}) = \bar{h}^3 + 3\bar{h}^2\bar{\alpha} + 3\,(\bar{\sigma}^2 + \bar{\alpha}^2)\bar{h} + 3\bar{\sigma}^2\bar{\alpha} + \bar{\alpha}^3 + \bar{\varepsilon}$$

The load carrying capacity of the bearing system then, is determined by

$$W = -\frac{h_0^3}{2\pi\eta a^4 \dot{h_0}} w = \int_0^1 RPdR \qquad (13)$$

which assumes the form

$$W = 3(1+\tau)\int_0^1 \frac{R^3}{g(\bar{h})}dR - \frac{S}{80}\left(3\Omega_f^2 + 4\Omega_f + 3\right) \qquad (14)$$

using the non-dimensional quantities.

RESULTS

Equation (12) calculates the dimensional pressure distribution, whereas Equation (5.14) determines the bearing system's non dimensional load capacity. [13] Clearly, there is a rise in the non-dimensional pressure distribution

$$6\tau \int_R^1 \frac{R}{g(\bar{h})} dR$$

while the increase in the load carrying capacity turns out to be

$$3\,\tau \int_0^1 \frac{R^3}{g(\bar{h})} dR$$

as compared to the case of traditional lubricant based bearing system.

Increases in the top plate curvature parameter increase load bearing capacity, whereas increases in the bottom plate curvature parameter reduce capacity. This is due to the fact that B and C have diametrically opposed effects. In order to have a satisfactory performance, the curvature parameters must be accurately specified. [14]

It depicts the skewness trends, and it shows how the variance effect works in a similar way. As a result, the combined impact of variance (-ve) and negatively skewed roughness (-ve) is quite beneficial.

Because of its negative influence, transverse surface roughness may be mitigated by a positive effect of magnetization and the proper configuration of curvature parameters, according to this research. The combined effects of magnetization and negatively skewed roughness, especially when variance (-ve) is present, may counteract the standard deviation and rotation to some extent. [15]

COMBINED EFFECT OF SURFACE ROUGHNESS AND SLIP VELOCITY ON JENKINS MODEL BASED MAGNETIC SQUEEZE FILM IN CURVED ROUGH CIRCULAR PLATES :

Squeeze film performance on circular plates with curved and rough surfaces is affected by slip velocity and surface roughness. The top plate's curvature is described by an exponential equation, whereas the bottom plate's curvature is described by a hyperbolic expression.

INTERPRETATION

The flow model for a ferrofluid was developed by Jenkins in 1972. As part of this study, a vector magnetization density was added to motion and temperature to complete an anisotropic fluid description. Using local magnetism as an independent variable, Jenkins could apply the same method to both static and dynamic conditions. He was able to discern between paramagnetic and ferromagnetic fluids because of this. Uniqueness theorem was devised for incompressible paramagnetic fluids and it was shown that the magnetization dissipated with the applied magnetic field. [16-18]

The steady flow model equations when the Maugin's corrections are applied are as follows:

$$\rho(\bar{q}\,\nabla)\bar{q} = -\nabla p + \eta\nabla^2\bar{q} + \mu_0(\bar{M}\,\nabla)\bar{H} + \frac{\rho A^2}{2}\nabla \times \left[\frac{\bar{M}}{M} \times \{(\nabla \times \bar{q}) \times \bar{M}\}\right]$$

Together with

$$\nabla\,\bar{q} = 0, \nabla \times \bar{H} = 0, \bar{M} = \bar{\mu}\bar{H}\,, \nabla\,(\bar{H} + \bar{M}) = 0$$

There are a number of variables that may be taken into consideration while discussing fluid density: p, q (the fluid velocity in the film area), H, p0 (the free space permeability), and A (the material constant). For further information on these criteria. According to the above equation, Jenkins model is an extension of the Neuringer-Rosensweig model with an additional term.

$$\frac{\rho A^2}{2}\nabla\times\left[\frac{\bar{M}}{M}\times\{(\nabla\times\bar{q})\times\bar{M}\}\right]=\frac{\rho A^2\bar{\mu}}{2}\nabla\times\left[\frac{\bar{H}}{H}\times\{(\nabla\times\bar{q})\times\bar{H}\}\right]$$

This affects the fluid's velocity. Neuringer-Rosensweig just affects pressure, but Jenkins' model affects both pressure and velocity of the Magnetic fluid simultaneously.

At every position between two solid surfaces, (u, v, w) is the velocity of the fluid. Using the hydrodynamic lubrication assumptions and keeping in mind that the flow is constant and axially symmetric, the equations of motion assume the form

$$\left(1-\frac{\rho A^2\bar{\mu}H}{2\eta}\right)\frac{\partial^2 u}{\partial z^2}=\frac{1}{\eta}\frac{d}{dr}\left(p-\frac{\mu_0\bar{\mu}}{2}H^2\right)$$

$$\frac{1}{r}\frac{\partial}{\partial r}(ru)+\frac{\partial w}{\partial z}=0$$

In view ofthe boundary conditions,

$u=0$ at $z=0,h,$

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the solution of equation (5.3) can be obtained as,

$$u=\frac{z(z-h)}{2\eta\left(1-\frac{\rho A^2\bar{\mu}H}{2\eta}\right)}\frac{d}{dr}\left(p-\frac{\mu_0\bar{\mu}}{2}H^2\right)$$

By substituting the value ofuin equation (5.4) and integrating it with respect to *z* over the interval (0, *h.)* one can get Reynolds type equation for obtaining the film pressure as

$$\frac{1}{r}\frac{d}{dr}\left(\frac{h^3}{\left(1-\frac{\rho A^2\bar{\mu}H}{2\eta}\right)}r\frac{d}{dr}\left(p-\frac{\mu_0\bar{\mu}}{2}H^2\right)\right)=12\eta\dot{h_0}$$

Here, the bearing surfaces are considered transversely rough. According to the stochastic model of Christensen and Tonder (1969a, 1969b, 1970), the thickness *h* of the lubricant film is assumed as

$$h=\bar{h}+h_s$$

For Christensen and Tonder, a divergence from the mean film thickness (hs) may be used to define the random roughness of the bearing surfaces.

$$z_u=h_0\,exp(-\beta r^2);\;0\le r\le a$$

approaches with normal velocity *ti0* tothe lower plate lying along the surface given by

$$z_l=h_0\left[\frac{1}{1+\gamma r}-1\right];\;0\le r\le a$$

where *p* and *y* are the curvature parameters of the corresponding plates and *h0* is the central film thickness.

$$h(r) = h_0\left[exp(-\beta r^2) - \frac{1}{1+\gamma r} + 1\right];\ 0 \le r \le a$$

Christensen and Tonder proposed stochastic averaging of the differential equation in question.

$$\frac{1}{r}\frac{d}{dr}\left(\frac{g(h)}{\left(1-\frac{\rho A^2\bar{\mu}H}{2\eta}\right)} r\frac{d}{dr}\left(p - \frac{\mu_0\bar{\mu}}{2}H^2\right)\right) = 12\eta\dot{h_0}$$

where

$$g(h) = (h^3 + 3h^2\alpha + 3(\sigma^2+\alpha^2)h + 3\sigma^2\alpha + \alpha^3 + \varepsilon)\left(\frac{4+sh}{2+sh}\right).$$

Introducing the non-dimensional quantities,

$$\bar{h} = \frac{h}{h_0}, R = \frac{r}{a}, P = -\frac{h_0^3 p}{\eta a^2 \dot{h_0}}, B = \beta a^2, C = \gamma a, H^2 = kr^2\frac{(a-r)}{a},$$

$$\mu^* = -\frac{k\mu_0\bar{\mu}h_0^3}{\eta\dot{h_0}}, \bar{A}^2 = \frac{\rho A^2\bar{\mu}\sqrt{k}a}{2\eta}, \bar{\sigma} = \frac{\sigma}{h_0}, \bar{\alpha} = \frac{\alpha}{h_0}, \bar{\varepsilon} = \frac{\varepsilon}{h_0^3}, \bar{s} = sh_0$$

and by making use ofthe equation (5.9), equation (5.8) reduces to 1714

$$\frac{1}{R}\frac{d}{dR}\left(\frac{g(\bar{h})}{(1-\bar{A}^2R\sqrt{1-R})}R\frac{d}{dR}\left(P - \frac{1}{2}\mu^* R^2(1-R)\right)\right) = -12$$

where,

$$g(\bar{h}) = \left(\bar{h}^3 + 3\bar{h}^2\bar{\alpha} + 3(\bar{\sigma}^2+\bar{\alpha}^2)\bar{h} + 3\bar{\sigma}^2\bar{\alpha} + \bar{\alpha}^3 + \bar{\varepsilon}\right)\left(\frac{4+\bar{s}\bar{h}}{2+\bar{s}\bar{h}}\right)$$

Solving the above expression, under the boundary conditions

$$P(1) = 0, \left(\frac{dP}{dR}\right)_{R=0} = 0$$

one can find the expression for dimensionless pressure as

$$P = \frac{1}{2}\mu^* R^2(1-R) - 6\int_1^R \frac{R}{g(\bar{h})}(1-\bar{A}^2R\sqrt{1-R})dR$$

In view of the classical result of Riemann, following the method, the load carrying capacity of the hearing system in non-dimensional form can be obtained from

$$W = -\frac{h_0^3}{2\pi\eta a^4 \dot{h_0}} w = \int_0^1 RPdR$$

$$= \frac{\mu^*}{40} + 3\int_0^1 \frac{R^3}{g(\bar{h})}\left(1 - \bar{A}^2 R\sqrt{1-R}\right)dR$$

The time t taken by the upper plate to reach a film thickness h_0 starting from an initial film thickness *h2* can be obtained in dimensionless form as

$$\bar{t} = \frac{h_2^2\, W\, t}{\eta a^4} = 3\left(\frac{1}{\bar{h}_0^2} - 1\right)\int_0^1 \frac{R^3}{g(\bar{h})}\left(1 - \bar{A}^2 R\sqrt{1-R}\right)dR \Big/ \left(\frac{1}{\pi} - \frac{\mu_1^*}{20}\right)$$

where

$$\bar{h}_0 = \frac{h_0}{h_2}, \mu_1^* = \frac{k\mu_0\bar{\mu}a^4}{W}$$

DISCUSSION

Equation of Load carrying capacity offers the suggestion that the load carrying capacity gets increased by

$$\frac{\mu^*}{40}$$

bearing system that relies on typical oil lubrication. Because magnetization increases the viscosity of a substance, this is probably the cause. As seen in the accompanying picture, the expression in equation (5.13) is linear with respect to the magnetization parameter. [19-20]

CONCLUSION

According to this research, the Shliomis model is more suitable for this kind of bearing system than the Neuringer-Rosensweig model. According to the graphic results, roughness should be given more consideration while designing the bearing system. [21] There is one more thing to mention: unlike standard lubricants, this system can operate with some load even when there is no flow. [22]

This article proves that roughness and sliding have a negligible effect on magnetism. Slip must be kept to a minimum, even while variation (-ve) and negatively skewed roughness (-ve) are involved. As a result of this study, it is necessary to include roughness into the design of the bearing system. The curvature parameters provide for further design freedom. When there is no flow, the bearing system can retain a certain level of load, unlike typical lubricants-based systems.

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