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# Magnetic fluids based lubrication performance on deformable rough finite journal bearings

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***Abstract:***
*This study investigates the combined effects of a non-Newtonian couple-stress lubricant and a magnetic fluid, incorporating velocity-slip and piezo-viscosity, on the lubrication performance of a finite journal bearing. Employing Stokes micro-continuum theory and the Barus viscosity-pressure relation with an artificial (homogeneous) slip surface, we analyze the load-carrying capacity, pressure distribution, and friction coefficient of the bearing. The results indicate that the piezo-viscosity parameter enhances both the maximum magnetic and hydrodynamic pressures within the bearing. Overall, the combined effects lead to a significant improvement in bearing performance.*

***Keywords:*** *Reynolds' type equation (RTE), Couple-stress lubricant (CSL), Finite journal bearing (FJB), Magnetic fluid (MF), Variable viscosity (VV), Velocity-slip (VS), Deformable roughness (DR), Pressure distribution (PD), Load profile (LP).*

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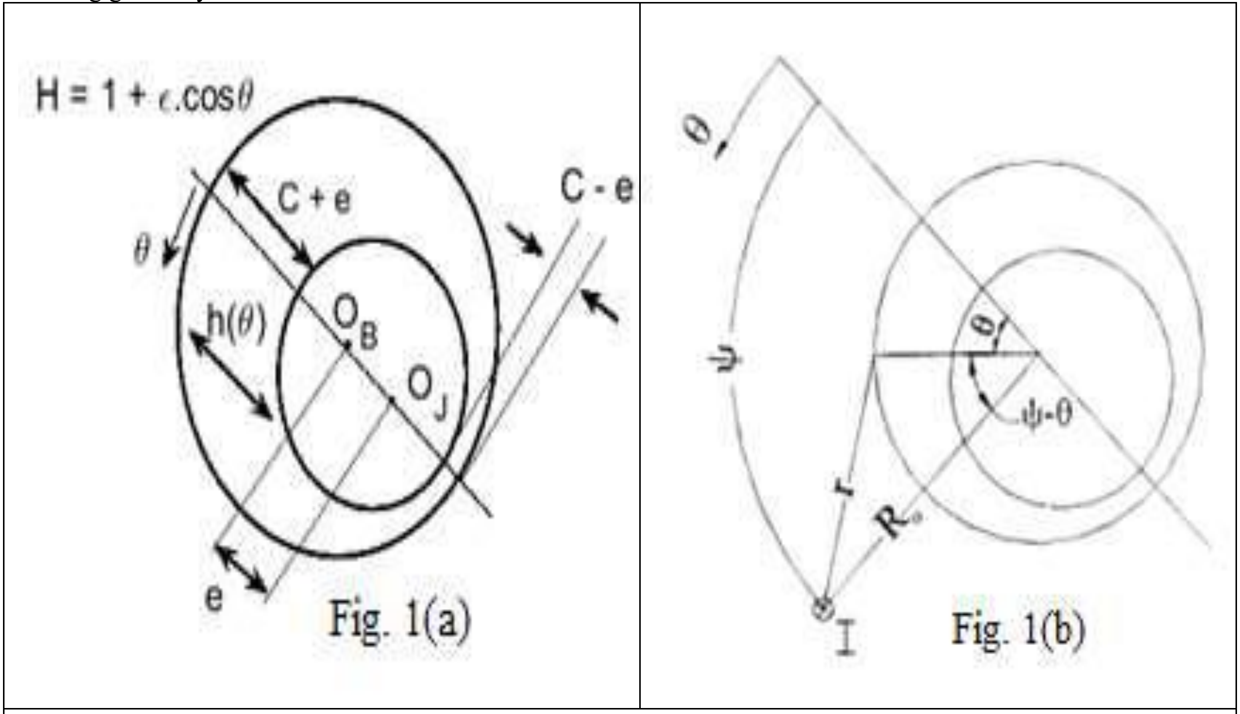
## I. Literature review and introduction:

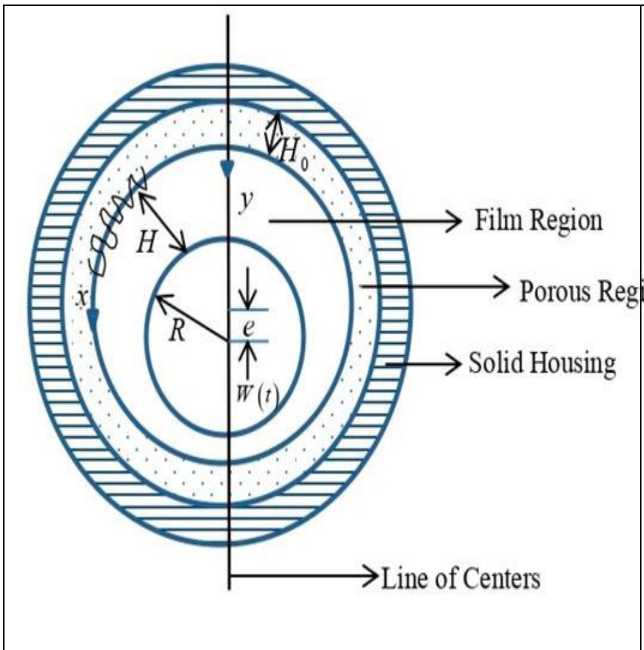
Ferrofluids (magnetic fluids) are composed of three main components: single-domain magnetic particles, a carrier (base) fluid, and a surfactant. Each magnetic particle is coated with long-chain or polymer molecules forming a surfactant layer, which maintains separation between particles and prevents aggregation due to magnetic forces. Recently, significant attention has been given to the application of ferrofluids in lubricated bearings ([1]–[6]), with studies showing that ferrofluid lubrication enhances bearing pressure and load-carrying capacity. Additionally, it improves bearing stability, reduces wear and friction, minimizes noise, and increases viscosity. Patel and Deheri [7] investigated the effects of slip velocity in journal bearings lubricated with magnetic fluids. Osman et al. [8] analyzed the influence of a current-carrying wire model on the design of hydrodynamic journal bearings using ferrofluids. J.R. Lin et al. [9] and Hanumagowda et al. [10] explored journal bearings lubricated with couple-stress fluids and pressure-dependent viscosity, applying the Barus formula to model the variable viscosity. Rao et al. [11] developed a theoretical model for partially textured, slip, slider, and journal bearings lubricated with couple-stress fluids, finding that slip and couple-stress parameters enhance load-carrying capacity and reduce friction. N.C. Das [12], along with Nada and Osman [13], studied the optimal load-bearing capacity of slider bearings lubricated with couple-stress fluids under a magnetic field, concluding that magnetic and couple-stress parameters both contribute to increased load capacity and improved pressure distribution. J.R. Lin et al. [14] further examined ferrofluid lubrication using the Shliomis model and Stokes' micro-continuum theory, considering the effects of rotating ferromagnetic particles and non-Newtonian properties. Rao and Prasad [15] studied the influence of velocity slip and viscosity variation in journal bearings, finding that while load-carrying capacity decreased with slip, friction was reduced with increasing viscosity. Oladeinde and Akpobi [16] analyzed the load capacity of slider bearings with slip surfaces and couple-stress fluids using the finite element method, demonstrating that the couple-stress parameter improves load capacity and that minimum slip velocity is required for maximum load. Dass et al. [17] investigated the combined effects of variable viscosity, couple-stress fluids, and three types of slip, concluding that engineered slip on the bearing surface combined with a no-slip journal surface yielded the highest load-carrying capacity and pressure distribution, while also reducing friction and significantly decreasing the attitude angle. In this study, we aim to investigate the combined effects of variable viscosity, couple-stress fluid, ferrofluid, and slip velocity on the performance of journal bearings. Specifically, we analyze how these parameters influence pressure distribution, load-carrying capacity, and friction, with the goal of optimizing the design of an ideal journal bearing. Accordingly, the effect of magnetic fluids, incorporating couple stresses, variable viscosity, and velocity slip on the lubrication performance of deformable, rough, finite journal bearings is studied and analyzed.

**Mathematical Formulation:**

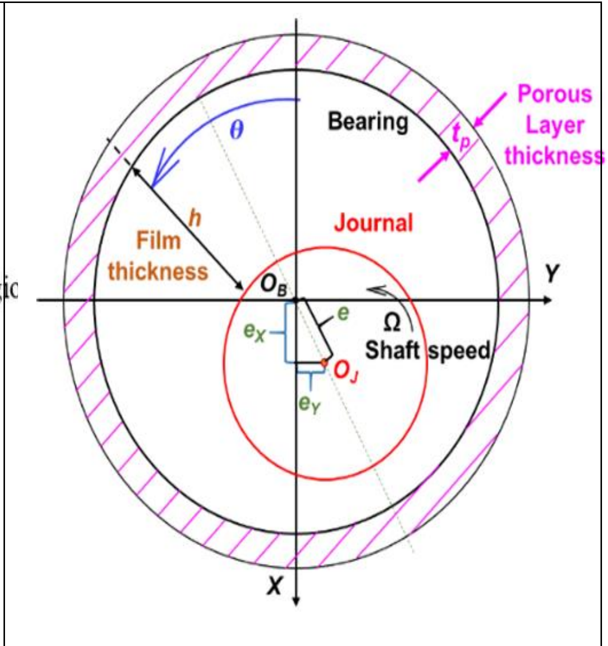
Bearing geometry:



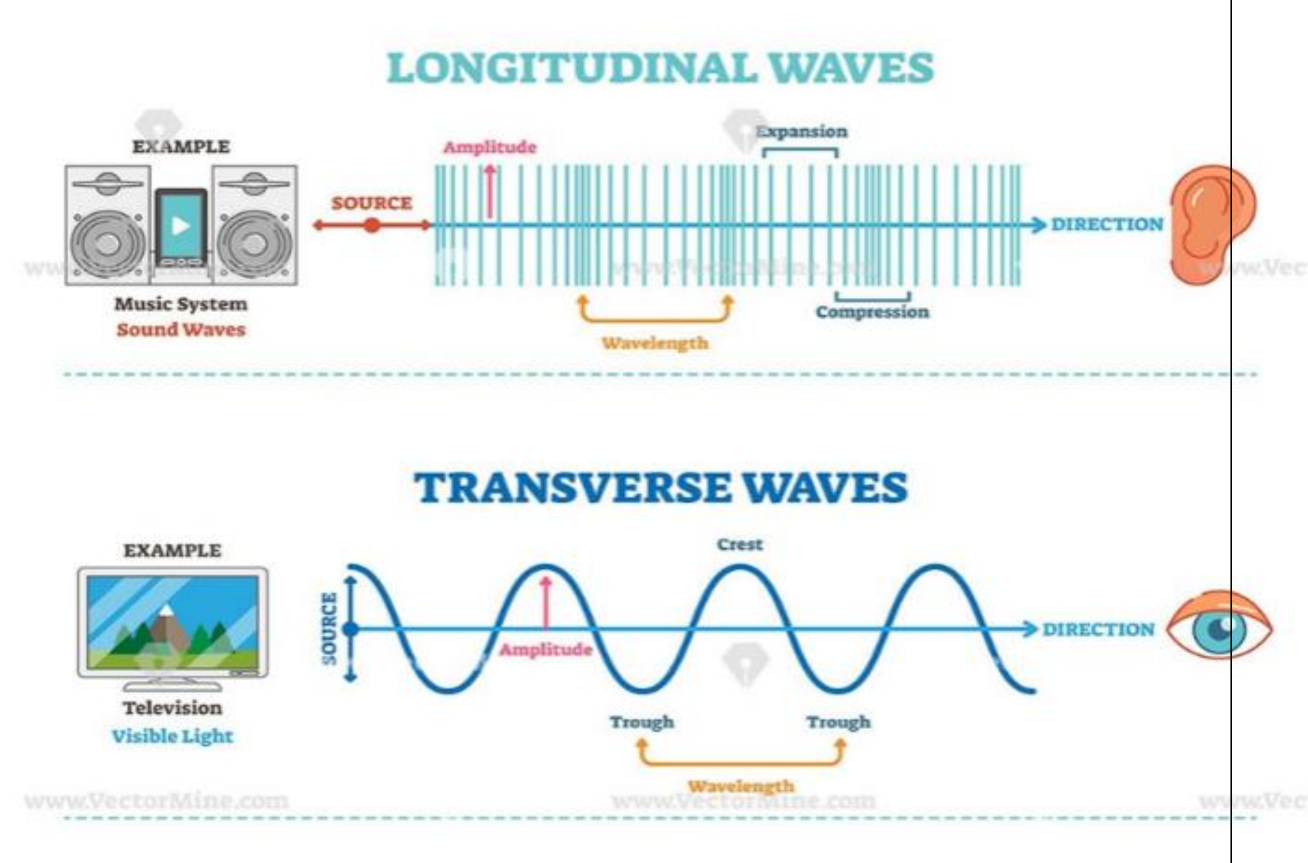

**Tyrone D. Dass et. al. 2020; ConETech-2020**




**Schematic diagram of porous journal bearing**




**Schematic diagram of a cylindrical journal and bearing with porous material layer; Luis San Andrés et. al. 2021**




**Longitudinal and Transverse surface roughness**

Figure 1(a) and (b) illustrate the magnetic field produced by a current passing through an infinitely long wire, which is displaced by a distance $R_0$ greater than the bearing radius. The wire is positioned at an angle $\psi$ relative to the line connecting the journal and bearing centers in order to achieve an enhanced pressure distribution, as proposed by Tarapov [18]. Meanwhile, Figure 1(b) shows the magnetic field model for the displaced infinitely long wire, where the field gradient exists only in the circumferential direction of the bearing. The Barus formula for the variation of the viscosity with pressure is given as:

$\mu = \mu_0 e^{\alpha p}$ (1)

The induced magnetic force for the ferro fluid in the presence of the magnetic field is given by Cowley and Rosensweig [19], Zelazo and Melchier [20]

$$F_m = \mu_0 X_m h_m (\nabla h_m) \tag{2}$$

$F_m$ will be used as an external body force in the field equations Zang [21]. From the Stoke's micro continuum theory the general momentum equation for an incompressible fluid with couple stress is given as:

$$\rho \frac{d\bar{u}}{dt} = -\nabla P + \rho \overline{F_m} + 0.5\rho \nabla \times \bar{C} + \mu \Delta \bar{u} - \eta \Delta^2 \bar{u} \tag{3}$$

In this context, u, $F_m$ and C denote the velocity, body force per unit mass, and body couple per unit mass, respectively. P represents the pressure, ρ is the fluid density, μ is the shear viscosity, and η is a material constant associated with the couple stress properties of the fluid. The couple stress parameter η characterizes the influence of couple stresses on the bearing performance of the system. Starting from the Navier–Stokes equations and incorporating the magnetic force as an external body force, the governing equations of motion for the fluid film are derived. It is assumed that body couples are absent, the fluid is incompressible, the lubricant is a non-Newtonian fluid with constant density, and slip occurs at the journal surface. Under these assumptions, the field equation in the X-direction is formulated as follows:

$$\eta \frac{\partial^4 u}{\partial z^4} - \frac{\partial}{\partial z}\left[\mu \frac{\partial u}{\partial z}\right] = -\frac{\partial p}{\partial x} + F_{mx} \tag{4}$$

**Feasible Boundary Conditions:**

The boundary conditions with slip on the journal are

$$u = U; \ \frac{\partial^2 u}{\partial z^2} = 0 \text{ at } z = 0 \tag{5a}$$

$$u = -\lambda \mu \frac{\partial u}{\partial z} \text{ and } \frac{\partial^2 u}{\partial z^2} = 0 \text{ at } z = h \tag{5b}$$

Using the boundary conditions in the field equation we obtain the velocity in the x-direction:

$$u = U - \frac{e^{-\alpha p}}{\mu_0}\frac{U}{\left[\lambda + \frac{e^{-\alpha p}}{\mu_0} g(h)\right]} z + \frac{e^{-\alpha p}}{\mu_0}\left(\frac{dp}{dx} - F_{mx}\right) \times \left\{ z^2 - \frac{\frac{e^{-\alpha p}}{\mu_0} g(h)^2 + 2\lambda g(h) - 2\lambda l e^{-\alpha p} \tanh\left[\frac{e^{-0.5\alpha p}}{2l} g(h)\right]}{\left(\lambda + \frac{e^{-\alpha p}}{\mu_0} g(h)\right)} z \right\}$$
$$+ \frac{e^{-\alpha p}}{\mu_0}\left[\frac{dp}{dx} - F_{mx}\right]\left[2l^2 e^{-\alpha p}\left(1 - \frac{\cosh\left\{e^{\alpha p}\left(\frac{(2z - g(h))}{2l}\right)\right\}}{\cosh\left\{e^{0.5\alpha p}\left(\frac{g(h)}{2l}\right)\right\}}\right)\right] \tag{6}$$

**Modified Averaged Reynolds' Equation:**

Flow into the permeable medium obeys Darcy's modified form of rule, whereas, hydro magnetic lubrication assumption holds in the film region. Tzeng and Saibel (1967) considered a method for irregular surface and accepted the one-dimensional film thickness to be of the shape

$h(x) = h^*(x) + h_s(x)$ (R1)

where $h^*(x)$ is the mean film thickness, $h_s(x)$ is the random deviation from the mean film thickness, h(x) is regarded as a random variable whose probability density function is either a Gaussian normal distribution function or Beta distribution function given by

$$f(h_s) = 1.09375C^{-7}\left[1 - h_s^2 C^{-2}\right], -c \le h_s \le c \tag{R2}$$

$f(h_s) = 0$, elsewhere (R3)

Distribution function above cited in the mathematical form is applied to obtain different parameters of roughness, like mean α, standard deviation σ and parameter ε (measure of symmetry) as

$\alpha = E(h_s)$ (R4)

$\sigma^2 = E[(h_s - \alpha)^2]$ (R5)

and

$\varepsilon = E[(h_s - \alpha)^3]$ (R6)

where

E is the expectancy operator

$$E(R) = \int_{-c}^{c} Rf(h_s)dh_s \quad \text{(R7)}$$

Under hydro-magnetic lubrication theory, stochastically averaging and adopting the properties of magnetic fluid lubrication (Neuringer and Rosensweig (1964), Bhat (2003), Christensen and Tonder (1970), Deheri et al. (2005), Rao and Prasad (2004)), the generalized Reynolds' equation is comes out as

$$\frac{\partial}{\partial x}\left[\frac{0.5g(h)e^{-\alpha p}}{\mu_0} f(h,l,\lambda,p)\frac{dp}{dx}\right] = \frac{\partial}{\partial x}\left[\frac{2\lambda\mu_0 + [g(h)]^{0.33}e^{-\alpha p}}{\lambda\mu_0 + [g(h)]^{0.33}e^{-\alpha p}} 0.5U[g(h)]^{0.33}\right] + \frac{\partial}{\partial x}\left[\frac{0.5g(h)e^{-\alpha p}}{\mu_0} f(h,l,\lambda,p)F_{mx}\right]$$

(7)

Using the Finite Difference Method, we solved the Modified Reynolds Equation that includes variable viscosity, magnetic effect, slip-parameter, and the couple-stress fluid with the help the following non dimensional parameters.

Here,

$g(h) = (h + pap'\delta)^3 + 3(\sigma^2 + \alpha^2)(h + pap'\delta) + 3(h + pap'\delta)^2\alpha + 3\sigma^2\alpha + \alpha^3 + \varepsilon + (12\phi/h^3)$

(R8)

Introducing the following dimensionless variables and parameters

$$H = \frac{h}{c} \qquad H_2 = \frac{h_2}{c} \qquad l^* = \frac{l}{c} \qquad x = R\theta$$

$$A = \frac{\lambda\mu_0}{c} \qquad P = \frac{\mu_0 URp^*}{2c^2} \qquad \overline{\alpha} = \alpha p_s \qquad h_m = h_{m0}H_m$$

$$\alpha_1 = \alpha^* = \frac{2\varsigma_0 X_m (h_{m0})^2 c^2}{R^2\mu_0 U} \qquad h^* = \frac{h}{h_0} \qquad \psi = \frac{\phi}{h_0^{\,3}}$$

$$\sigma^* = \frac{\sigma}{h_0} \qquad \alpha^* = \frac{\alpha}{h_0} \qquad \varepsilon^* = \frac{\varepsilon}{h_0^{\,3}} \qquad \delta^* = \frac{\delta}{h_0}$$

$g(h^*) = (1 + p^*\delta^*)^3 + 3(1 + p^*\delta^*)^2\alpha^* + 3(1 + p^*\delta^*)(\sigma^{*2} + \alpha^{*2}) + 3\sigma^{*2}\alpha^* + \alpha^{*3} + \varepsilon^* + 12\psi$ (R9)

By applying all the above non-dimensional terms to expression (7) and solving it with respect to the boundary conditions (5.a and 5.b), the load components (W) per unit length, along the direction perpendicular to the line of centers, can be obtained by integrating the pressure around the bearing from θ = 0 to θ = π. Similarly, the load components normal to the line of centers per unit length can also be determined. The dimensionless load is given by:

$$W^* = \left[\begin{array}{l}\left\{\int_0^{\pi}\left[\frac{e^{\overline{\alpha}}}{H^3F_2(H,l^*,A,\overline{\alpha})}\left(1+\frac{A}{A+He^{-\overline{\alpha}}}\right)H - \frac{e^{\overline{\alpha}}}{H^3F_2(H,l^*,A,\overline{\alpha})}\left(1+\frac{A}{A+H_2e^{-\overline{\alpha}}}\right)H_2\right]\cos\theta d\theta\right\}^2 + \\ \left\{\int_0^{\pi}\left[\frac{e^{\overline{\alpha}}}{H^3F_2(H,l^*,A,\overline{\alpha})}\left(1+\frac{A}{A+He^{-\overline{\alpha}}}\right)H - \frac{e^{\overline{\alpha}}}{H^3F_2(H,l^*,A,\overline{\alpha})}\left(1+\frac{A}{A+H_2e^{-\overline{\alpha}}}\right)H_2\right]\sin\theta d\theta\right\}^2\end{array}\right]^{0.5}$$

(8)

The attitude angle (ψ) is defined as the angular position of the line of action of the load-bearing capacity relative to the location of the minimum film thickness or the point of closest approach. The frictional parameter ($\mu_f$) and the attitude angle (ψ) are calculated accordingly.

$$\tan\psi = \frac{W_0^{\,*}}{W_{1.57}^{\,*}}$$

(9a)

$$\mu_f\left(\frac{R}{C}\right)=\frac{F^*}{W^*}$$

(9b)

where

$$F^*=\int_0^{\pi}\left(\frac{e^{\bar{\alpha}}}{H^3F_2\left(H,l^*,A,\bar{\alpha}\right)}\left\{1+\frac{A}{A+He^{-\bar{\alpha}}}\right\}H-\frac{e^{\bar{\alpha}}}{H^3F_2\left(H,l^*,A,\bar{\alpha}\right)}\left\{1+\frac{A}{A+H_2e^{-\bar{\alpha}}}\right\}H_2\right)\times$$

$$\left(\frac{e^{\bar{\alpha}}H\left(H-2l^*e^{\bar{\alpha}}\tanh\left(\frac{e^{-0.5\bar{\alpha}}}{2l^*}H\right)\right)}{2\left(A+e^{-\bar{\alpha}}H\right)}\right)d\theta-\int_{\pi}^{2\pi}\frac{1}{A+e^{-\bar{\alpha}}H}d\theta \tag{10}$$

## II. Fruitful Results and Discussions:

Expressions (8) and (10) clearly show that the load-carrying capacity and friction depend on various parameters, including standard deviation, variance, skewness, deformation, porosity, and aspect ratio. Furthermore, it is observed that negative variance and negative skewness in roughness increase the load profile, whereas standard deviation and porosity reduce the load-carrying capacity. In an attempt to characterize the ideal journal bearing, a combined effect of the couple-stress lubricant, magnetic fluid, velocity-slip, and the piezo–viscosity effect have been utilized. Previous results indicated that the magnetic parameter improves the load-carrying capacity and the friction of the journal bearing. With the incorporation of the couple-stress lubricant, variable viscosity, and the slip-velocity, there is further enhancement of these characteristics. The maximum magnetic and HD pressures increase with increasing piezo-viscosity parameter in the presence of the slip parameter. Higher load values occur in the presence of the magnetic parameter, high eccentricity ratio, and higher piezo-viscosity parameter. For an increase in the magnetic parameter the friction decreases. The combined effect of high couple-stress parameter and low piezo-viscosity parameter yields the lowest friction. From the above results, we can enhance the journal bearing parameters by combining the slip, piezo-viscosity, and couple-stress parameters. Thereby, the production of an ideal journal bearing with minimum friction and high load carrying capacity can be achieved. The study of magnetic fluids, which incorporates couple stresses, variable viscosity, and velocity slip, on the lubrication performance of finite journal bearings is a highly specialized area of tribology (the study of friction, wear, and lubrication). In this context, finite journal bearings are typically used in rotating machinery, such as engines or turbines, where the bearing supports the rotating shaft and the performance of the lubrication system is crucial to reducing friction and wear. Load-Carrying Capacity and Friction Dependence: The load-carrying capacity and frictional behavior of a surface (especially in contact mechanics, tribology, or porous materials) are influenced by various surface and material characteristics.

**Key parameters include:**

Standard Deviation of Roughness (σ)

Represents how much the surface heights deviate from the mean.

Higher standard deviation typically leads to more irregular surface profiles, causing reduced real contact area and thus lower load-carrying capacity.

Variance of Roughness ($\sigma^2$)

Directly related to the standard deviation [Variance = (Standard Deviation)$^2$].

Negative variance (in a statistical modeling sense, meaning surfaces with valleys instead of peaks) increases the load-carrying capacity by enhancing contact area (due to valleys being filled or deformed under load).

Skewness of Roughness (Sk)

Measures the asymmetry of the surface profile. Negative skewness (more valleys than peaks) improves load support because the valleys help accommodate deformation without causing sharp stress concentrations.

Surfaces with positive skewness (sharp peaks) tend to wear faster and carry less load effectively.

Deformation (Elastic or Plastic)

As surfaces deform, contact areas increase, often improving load-carrying capacity. Deformation behavior also strongly influences friction — elastic deformation often leads to lower friction, while plastic deformation can cause higher friction due to material flow and adhesion.

Porosity (P). Refers to the presence of voids within a material or surface. Higher porosity leads to lower load-carrying capacity because pores act like stress concentrators and reduce the effective solid area supporting the load. Porosity can also alter friction depending on the pore size distribution and material structure.

Aspect Ratio (AR)

Defined often as height-to-width ratio of surface features or pores. High aspect ratio structures (tall and narrow) are typically less stable under load and can collapse, reducing load-carrying ability. Low aspect ratio features (short and wide) can better sustain loads due to broader load distribution.

**Observations**
Negative variance and negative skewness → Increase load-carrying capacity (More valleys than peaks, helping distribute the load better).
Higher standard deviation and higher porosity → Decrease load-carrying capacity (More irregularity and less solid material to carry the load).
Friction is a more complex function, often increasing with deformation (especially plastic), and varying with roughness details.

**Load profile**

$L = f(\sigma, \sigma^2, Sk, P, AR, D)$

Here,
$\sigma$ = roughness (standard deviation)
$\sigma^2$ = roughness variance
Sk = surface skewness
P = applied pressure
AR = aspect ratio (maybe of roughness features)
D = deformation (elastic)

**And the dependencies are:**

$\frac{\partial L}{\partial \sigma} < 0 \Rightarrow$ increasing roughness decreases load capacity

$\frac{\partial L}{\partial P} < 0 \Rightarrow$ increasing pressure decreases load capacity

$\frac{\partial L}{\partial S_k} < 0 \Rightarrow$ (for negative skewness) → more negative skewness decreases load capacity.

$\frac{\partial L}{\partial(-\mathrm{var}iance)} > 0 \Rightarrow$ larger variance (i.e., rougher features?) helps increase load capacity.

$\frac{\partial L}{\partial AR} \Rightarrow$ depends — usually, higher AR (sharper or taller features?) reduces.

$\frac{\partial L}{\partial D} > 0 \Rightarrow$ more elastic deformation increases real contact area and thus increases L.

These are fluids whose viscosity can be controlled by applying a magnetic field. This means that the lubrication characteristics of the fluid can be altered dynamically in response to external magnetic fields, providing the ability to fine-tune the bearing's performance. When a magnetic field is applied, the suspended particles (usually ferrous) in the fluid align with the field lines, changing the rheological properties of the fluid and making it more resistant to flow, i.e., the fluid becomes "thicker." Couple stresses refer to the torque or rotational effects that are exerted on the fluid. In the case of magnetic fluids, these couple stresses may be influenced by the alignment and interaction of the magnetic particles, which can contribute to the complex flow behavior in the bearing. In the context of journal bearings, couple stresses can affect the pressure distribution and load-carrying capacity of the lubricant film, leading to changes in the friction and wear characteristics. The viscosity of the lubricant in the bearing is typically not constant and can vary with factors such as temperature, pressure, and, in the case of magnetic fluids, the intensity of the applied magnetic field. Variable viscosity means that the lubricant film in the bearing can change its thickness and load-bearing capacity dynamically. Higher viscosity can lead to a thicker lubricant film, potentially improving the bearing's performance, while lower viscosity can reduce the film thickness and increase the risk of metal-to-metal contact. Velocity slip refers to the difference in velocity between the surface of the bearing (the shaft) and the lubricant. Normally, a no-slip condition is assumed, where the velocity of the fluid at the bearing surface matches that of the surface. However, in certain situations, particularly under high shear rates or with certain fluids like magnetic fluids, the velocity of the fluid near the surface may be different from that of the bearing surface. Velocity slip can affect the shear stresses and pressure distribution within the lubricant film, influencing the lubrication performance.

When incorporating these factors into the study of lubrication in finite journal bearings, the following effects are typically observed: Magnetic fluids offer the potential for adaptive control of viscosity. When subjected to a magnetic field, the viscosity can be increased, improving the load-carrying capacity of the lubricant. This could help prevent metal-to-metal contact and reduce wear. The influence of couple stresses also means that the lubricant can develop better film support, improving the bearing's performance under high-load conditions. Variable viscosity and couple stresses can lead to an increase in the effective film thickness. This helps maintain the separation between the bearing surfaces, which reduces friction and wear. In conjunction with velocity slip, the dynamic adjustment of viscosity due to magnetic fields can allow for more precise control over the film thickness, ensuring optimal lubrication in varying operational conditions. The ability to control the viscosity of the lubricant dynamically can significantly reduce friction and wear, especially under varying load conditions. The magnetic fluid’s ability to alter its properties in response to external magnetic fields allows it to adapt to changing operational conditions, minimizing the risks of dry friction and reducing wear. Moreover, the couple stresses generated by the fluid can reduce the tendency for the lubricant film to collapse under high shear stress, further improving the bearing's durability. Magnetic fluids exhibit non-Newtonian behavior (their viscosity changes with the rate of shear). This complicates the lubrication model but can also offer performance benefits, such as maintaining an appropriate lubricant film even under varying loads and speeds. The presence of velocity slip can destabilize the lubrication system, especially in bearings with high speeds or under extreme conditions. However, the use of magnetic fluids, particularly with external magnetic control, can provide a mechanism to stabilize the system by modifying the viscosity and fluid behavior in real-time. To quantify the impact of these factors on lubrication performance, mathematical models are used that account for the behavior of magnetic fluids in journal bearings. These models typically involve solving the Reynolds equation with additional terms that account for the magnetic field effects (magneto hydrodynamic forces), couple stresses, and variable viscosity. Velocity slip is incorporated using slip boundary conditions at the bearing surface. The Reynolds equation for lubrication is modified to account for the non-Newtonian, magneto rheological fluid's behavior. This includes viscosity changes with the magnetic field and pressure dependence. The equation will also incorporate terms representing couple stresses that arise from the magnetic interactions between the particles in the fluid and the field, affecting the shear forces and torque transmission. To model velocity slip, the classical no-slip boundary condition is replaced with a slip condition, which might depend on the applied magnetic field, fluid properties, and bearing geometry. The inclusion of magnetic fluids in finite journal bearings, considering couple stresses, variable viscosity, and velocity slip, has the potential to enhance the performance of the lubrication system significantly. These effects can lead to improved load-carrying capacity, reduced friction and wear, and better stability of the lubrication film under varying conditions. However, to fully optimize these effects, advanced modeling and experimental work are needed to understand the interplay of these factors in different operational regimes.

**Future Scope:**

In the pursuit of designing an ideal journal bearing, the combined effects of a couple-stress lubricant, magnetic fluid, velocity-slip, and the piezo-viscosity effect have been explored. Previous studies have shown that the magnetic parameter enhances both the load-carrying capacity and the frictional performance of the journal bearing. The addition of a couple-stress lubricant, variable viscosity, and velocity-slip further improves these characteristics. Notably, the maximum magnetic and hydrodynamic pressures increase with a higher piezo-viscosity parameter, particularly in the presence of slip effects. Higher load capacities are observed with increased magnetic parameters, higher eccentricity ratios, and elevated piezo-viscosity parameters. Additionally, an increase in the magnetic parameter results in reduced friction. The combination of a high couple-stress parameter and a low piezo-viscosity parameter leads to the lowest friction values. These findings suggest that journal bearing performance can be significantly enhanced by optimizing the slip, piezo-viscosity, and couple-stress parameters, thereby paving the way for the development of an ideal journal bearing with minimal friction and superior load-carrying capacity.

**Data Availability Declaration of Competing Interest:**
There is up to certain extent data has been used for this research and cited at the proper place/s. Author/s declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this article.

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## Citations and References:


[1]. Tipei, Nicolae. 1983. Overall Characteristics of Bearings Lubricated With Ferrofluids. Journal of Lubrication Technology, Vol. 105. Transactions of the ASME.

[2]. Agrawal, V.K. 1986. Magnetic-fluid based porous inclined slider bearing. Wear, Vol. 107, (pp. 133-139).

[3]. S. Verma and K. Budheeja. 2012. Study of Piezo-Viscous Effect on Multirecessed Capillary Compensated Hydrostatic Journal Bearing. ResearchGate, National Conference on Advancements in Mechanical Engineering and Energy environment, SLIET, Longowal, (Pb) India, 6-7.

[4]. N. S. Patel, D. P. Vakharia, and G. M. Deheri. 2012. A Study on the Performance of a Magnetic-Fluid-Based Hydrodynamic Short Journal Bearing. International Scholarly Research Network ISRN Mechanical Engineering .Article ID 603460, doi:10.5402/2012/603460.

[5]. Jaw-Ren Lin, Po-Jui Li, and Tzu-Chen Hung. 2013. Effects of Non-Newtonian Ferrofluids on the Performance Characteristics of Long Journal Bearings. FDMP, vol. 9, no.4 (pp.419-434).

[6]. G.S. Nada, G.T. Abdel-Jaber , Abdo, H.S.M. 2012. Thermal effects on hydrodynamic journal bearing lubricated by magnetic fluids with couple-stress. International Journal of Mechanical and Mechatronics Engineering IJMME-IJENS Vol:12 No. 05.

[7]. Patel, R. U, Deheri, G.M. 2013. Effect of Slip Velocity on the Performance of a Short Bearing Lubricated with a Magnetic Fluid. Acta Polytechnica 53 (6): 890-894, doi: 10.14311//AP. 2013.53.0890.

[8]. Osman, T.A., Nada, G.S., and Safar, Z.S. 2001.Effect of using current-carrying-wire models in the design of hydrodynamic journal bearings lubricated with ferrofluid. Tribology letters Vol. 11, No. 1.

[9]. Jaw-Ren Lin, Li-Ming Chu, Chi-Ren Hung, and Rong-Fang Lu. 2011. Squeeze Film Problems of Long Partial Journal Bearings for Non-Newtonian Couple Stress Fluids with Pressure-Dependent Viscosity. Z. Naturforsch. 66a, 512 – 518 (2011) / DOI: 10.5560/ZNA.2011-0009.

[10]. Hanumagowda B N, Savitramma G, Salma A. and Noorjahan. 2018. Combined effect of Piezo-viscous dependency and nonNewtonian couple stresses in Annular Plates Squeeze-Film characteristics. National Conference on Mathematical Techniques and its Applications (NCMTA 18) IOP Publishing, IOP Conf. Series: Journal of Physics: Conf. Series 1000 (2018) 012083 doi :10.1088/1742-6596/1000/1/012083

[11]. T. V. V. L. N. Rao, A. M. A. Rani, T. Nagarajan, F. M. Hashim. 2013. Analysis of Couple Stress Fluid Lubricated Journal Bearing with Partial Slip Surface. IEEE Business Engineering and Industrial Applications Colloquium (BEIAC).

[12]. N. C. Das. 1998. A study of optimum load-bearing capacity for slider bearings lubricated with couple stress fluids in magnetic field. Tribology International Vol. 31, No. 7, (pp. 393–400).

[13]. G.S. Nada and T.A. Osman, 2007. Static Performance of Finite Hydrodynamic Journal Bearings Lubricated by Magnetic Fluids with Couple Stresses. Tribology Letters DOI 10.1007/s11249-007-9222-0.

[14]. Jaw-Ren Lin, Po-Jui Li, and Tzu-Chen Hung. 2013. Lubrication Performances of Short Journal Bearings Operating with Non-Newtonian Ferrofluids. Z. Naturforsch. 68a, 249 – 254 / DOI: 10.5560/ZNA.2012-0114

[15]. Rao R.R., Prasad K.R. 2004. Effects of Velocity-Slip and Viscosity Variation on Journal Bearings. ANZIAM J. 46 (2004), 143-155.

[16]. M.H.Oladeinde and J.A.Akpobi. 2010. A Study of Load Capacity of Finite Slider Bearings with Slip Surfaces and Stokesian Couple Stress Fluids. International Journal of Engineering Research in Africa Vol. 1 (pp 57-66).

[17]. Tyrone Dass, S. Rao Gunkala, D.M.G Comissiong. 2019. The combined effect of velocity-slip, variable viscosity, couple-stress fluid in the lubrication of a finite journal bearing. Communicated to Ain Shams Engineering Journal, Elsevier.

[18]. I.E. Tarapov, 1972. Magnetohydrodynamic, 8 (1972) 444

[19]. Cowley, M.D., Rosensweig, R.E.1967. The interfacial stability of feromagnetic fluid.. Fluid Mech. 30, 671 688.

[20]. Zelazo, R.E., Melcher, J.R..1969. Dynamics and stability of ferrofluids surface interaction. J. Fluid Mech., 39, 1-24.

[21]. Zhang Y. 1991. Static characteristics of magnetized journal bearings lubricated with ferrofluid. ASME J. Tribol. 113, 533-538.

[22]. Tyrone D. Dass, Sreedhara Rao Gunakala and Donna M.G. Comissiong, THE COMBINED EFFECT OF MAGNETIC FLUIDS WITH COUPLE STRESSES, VARIABLE VISCOSITY AND VELOCITY-SLIP ON THE LUBRICATION OF FINITE JOURNAL BEARINGS, The International Conference on Emerging Trends in Engineering and Technology (IConETech-2020) Faculty of Engineering, The UWI, St. Augustine, June 1st – 5th, 2020, pp. 888-898.

[23]. Christensen H. (1970): Stochastic model for hydrodynamic lubrication of rough surfaces. – Proc. Inst. Mech. Engrs, vol.184, pt 1, pp.1013-1022.

[24]. Christensen H. and Tønder K. (1971): The hydrodynamic lubrication of rough bearing surfaces of finite width. – ASME, J. Lubric. Technol., vol.93, No.2, pp.324-330.

[25]. Christensen H. and Tønder K. (1973): The hydrodynamic lubrication of rough journal bearings. – ASME, J. Lubric. Technol., vol.95, No.1, pp.166-172.

[26]. Prin. Dr. Ms. Pragna A. Vadher, Dr. Dhinesha Ruwanthi Perera, Dr. Sanjeev Kumar, Dr. Gunamani B. Deheri, Rakesh M. Patel, A study of deformable rough surfaces on finite slider bearing, Technische Sicherheit(Technical Security) Technische Sicherheit ISSN:2191-0073, ISSN:1434-9728/E-ISSN:1436-4948 || Impact Factor 6.1, 2025, 25(4), pp. 12-23.

[27]. Rakesh M. Patel, Prin. Dr. Pragna A. Vadher, Dr. Gunamani B. Deheri, Dr. Bharatkumar N. Valani, Ferrofluid-Based Tilted Deformable Rough Porous Pad Bearing, GIS JOURNAL OF SCIENCE, 2025, 12(2), pp. 348-359.

[28]. Dr. G. M. Deheri, Dr. S. G. Sorathiya, Dr. J. V. Adeshara, Dr. H. P. Patel and Rakesh M. Patel, Effect of Slip Velocity on Longitudinal Rough Hydro magnetic Squeeze Film Conducting Rotating Circular Plates, Journal of Materials and Engineering, 2025, 3(2), pp. 137-143.

[29]. Prin. Dr. Pragna A. Vadher, Dr. Gunamani B. Deheri, Dr. Bharatkumar N. Valani and Rakesh M. Patel, Behaviour of Deformable Elliptical Plates with Micropolar Fluids, Journal of Technology, 2025, 12(12), p.p. 693-703.

[30]. Tzeng, S. T. and Saibel, E., Surface roughness effect on slider bearing lubrication, Trans. ASME, J. Lub. Tech., 10, p.p. 334-338, 1967.

[31]. Bhat, M. V., Lubrication with a magnetic fluid. Team Spirit (India) Pvt. Ltd., New Delhi, 2003.

[32]. Patel R.M. and Deheri G.M., Magnetic fluid based squeeze film behaviour between rotating porous circular plates with a concentric circular pocket and surface roughness, effects, Int. J. of Applied Mechanics and Engineering, 8(2), 271-277, 2003.

[33]. Vadher, P., Daheri, G. M., and Patel, R. M., Hydrpmagnetic squeeze film Between conducting porous transversely rough triangular plates, Journal of Engineering Annals of Faculty of engineering Hunedora, 6(1), 155-168, 2008.