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MHD Micro Polar Fluid Flow on Porous Triangular Plates

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Abstract: This study examines the combined effects of deformable surface roughness (DSR) and magneto hydrodynamics (MHD) on the squeeze film behavior between triangular plates (TP). Utilizing Christensen's stochastic theory, a modified Reynolds equation is developed for a one-dimensional structure featuring both azimuthal and radial roughness patterns. A micro polar fluid is used as the lubricant. Analytical solutions are obtained for the mean squeeze film pressure and workload. Comparisons between MHD and non-MHD scenarios reveal that the inclusion of MHD significantly enhances both pressure and workload. Moreover, as the roughness parameter increases, the pressure and workload also increase with radial distance and film thickness, respectively. Additionally, higher values of the coupling number, Hartmann number, and roughness parameter are found to extend the squeeze time of the lubricant.
Keywords: Reynolds' type equation (RE), Micro polar fluid flow (MPFF), MHD, Porous surface (PS), Deformable surface roughness (DSR), Triangular plates (TP).
Subject Classification: 74A55 Theories of friction (tribology); 74Dxx Materials of strain-rate type and history type, other materials with memory (including elastic materials with viscous damping, various viscoelastic materials); 00A69 General applied mathematics {For physics, see 00A79 and Sections 70 through 86} 00A71 Theory of mathematical modeling 00A72 General methods of simulation.

I. INTRODUCTION

Background on micro polar fluids, MHD, and surface roughness: Squeeze film technology is widely utilized across a broad range of industries, from small mechanical components like disc clutches to large-scale turbomachinery in power plants. In these systems, lubricants are introduced to reduce friction and protect rotating parts. With advancements in modern machine tools, the use of non-Newtonian fluids has become increasingly prevalent. Recent experiments have demonstrated that blending base oils with long-chain additives significantly enhances lubrication performance, leading to reduced friction and less surface wear. The rheological behavior of these non-Newtonian fluids has been studied using various micro-continuum theory models. Among them, micro polar fluids (MPFs) are particularly notable due to their ability to exhibit micro-rotational motion and account for the inertial effects of rotating micro particles.

II. LITERATURE REVIEW

Eringen [5] laid the groundwork for the micro-continuum theory of micro polar fluids (MPF). Building on this foundation, several studies have explored the application of MPF in analyzing squeeze film flow across various geometries. For instance, Prakash and Sinha [15] investigated the effects of MPF between parallel plates, while Sinha and Singh [21] focused on hemispherical bearings. Siddangouda [20] examined the behavior of MPF in parallel stepped plates. These studies consistently demonstrate that using MPF as a lubricant enhances the load-carrying capacity of the system. In mechanics, the movement of an object under the influence of external forces is described as dynamics. When such motion involves fluid particles subjected to a magnetic field, the field of study is termed magneto hydrodynamics (MHD).

Hartmann's pioneering work in MHD flow examined the impact of a magnetic field (B_0) applied perpendicular to fluid motion. His theoretical and experimental investigations of an incompressible fluid confined between parallel planes led to the formulation of the now widely known Hartmann flow. Since then, numerous theoretical and analytical studies have expanded upon this concept, including contributions by Anncy et al. [1], Salah et al. [19], Elniel et al. [4], Nayak [12], Cowling [3], M. Hamza [7], Kuzma [11], Sujatha and Sundarammal [22], Toloian et al. [23], Fathima et al. [6], Patel et al. [16], and others. These studies have shown that the electromagnetic forces present in MHD flows significantly enhance load-carrying capacity compared to non-magnetic cases. Over time, lubricants used in machines undergo chemical and physical transformations due to interactions with additives and operating surfaces. These interactions can convert a Newtonian fluid into a non-Newtonian one and influence the condition of the contact surfaces. Ignoring these changes can result in severe wear and damage to machine components. As a result, researchers have increasingly focused on the role of lubricants in modifying surface texture and behavior. Christensen [2] introduced a stochastic model to assess the effect of surface roughness on bearing performance. Further investigations by Prakash and Tonder [17], Naduvinamani and Siddangouda [13], Vadher [24], Rao et al. [18], and Hanumagowda et al. [8] explored the influence of lubricant-surface interactions across different geometric configurations. Their findings confirm that both MHD effects and surface roughness contribute significantly to improved load capacity. For example, Halambi et al. [9] analyzed MHD lubrication in rough, porous elliptic plates; Hanumagowda [10] studied rough flat and curved annular plates; and Naduvinamani [14] explored porous circular stepped plates. The present study is motivated by the lubrication behavior in wet clutches, which use lubricants to cool clutch packs featuring triangular surface grooves. Similar squeeze film actions between triangular plates are found in many mechanical systems. This inspired an investigation into squeeze film lubrication between triangular plates, incorporating an external magnetic field and using MPF as the lubricant. Furthermore, the effect of surface roughness on this configuration provides additional motivation for exploring this model.

A. *Geometrical Formulation*

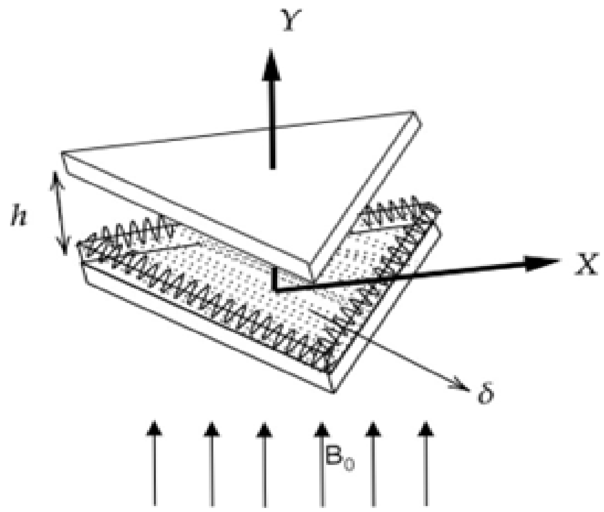

Figure 1: Bearing geometry of triangular plates

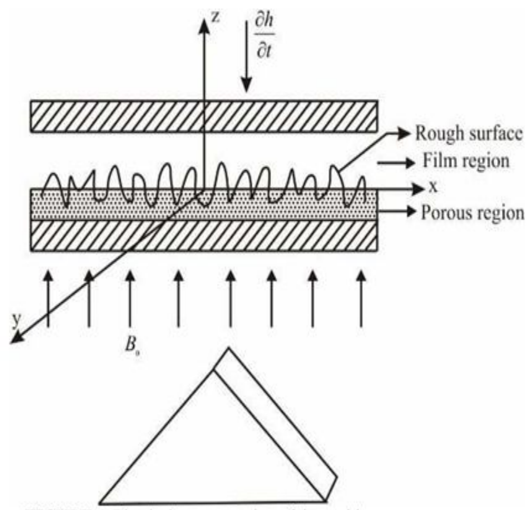

Figure 2: Geometry of roughness

Figure (1) represents the geometry of the triangular plates that are lubricated with micro polar fluid, while the roughness configuration is mentioned in figure (2). The gap between the upper and the lower plate is filled with the micro polar fluid. The thickness of this fluid is considered as h. The bottom plate is oriented towards the x axis and remains fixed. The upper plate moves towards the lower plate with a normal velocity $v - \frac{d(2h)}{dt}$.

The lower plate is porous with thickness δ and it is supported below by a solid backing. Also, the lower surface is assumed to be rough, which adds to the surface roughness asperities, which can be expressed mathematically by the form,

$$H = h\,(t) + h_s(x, y, \zeta) \quad \text{(R1)}$$

The first part of the above expression represents the deterministic part of the film thickness and the second part represents the random part of the film thickness. ζ is the index parameter that determines the exact roughness pattern. The expectancy operator E(⋆) is given by

$$E(*) = \int_{-\infty}^{\infty} (*) f(h_s) ds \quad \text{(R2)}$$

where $f(h_s)$ is the probability density function of the stochastic variable hs and is expressed as,

$$f(h_s) = \frac{1.093}{c^7}\left(c^2 - {h_s}^2\right)^3; -c < h_s < c \tag{R3}$$

$f(h_s) = 0$ otherwise (R4)

Film thickness h(x) of the lubricant film is considered as $h(x) = \bar{h}(x) + h_s$; Where $\bar{h}(x)$ mean film thickness and h_s is the deviation from the mean film thickness characterizing the random roughness of the bearing surfaces. h_s is considered to be stochastic in nature and administered by the probability density function $f(h_s)$; $-c \le h_s \le c$, c is maximum deviation from the mean film thickness. Mean α, standard deviation σ and skewness ε is resolute by the relationships:

$$\alpha = E(h_s) \tag{R5}$$

$$\sigma^2 = E[(h_s - \alpha)^2] \tag{R6}$$

and

$$\varepsilon = E[(h_s - \alpha)^3] \tag{R7}$$

Bearing surfaces are hypothetical to be deformable rough and averaged using the stochastic model of Christensen and Tonder (1969a, 1969b, 1970).

B. Solution Methodology

The basic assumptions of thin film lubrication for a micro polar fluid as described by Eringen [1] is assumed to hold true for the case discussed here. The following equations are considered for deriving the expression for velocity

$$\left(\mu + \frac{\chi}{2}\right)\frac{\partial^2 u}{\partial y^2} + \chi\frac{\partial v_2}{\partial y} - \sigma {B_0}^2 = -\frac{\partial p}{\partial x} \tag{1}$$

$$\gamma\frac{\partial^2 v_2}{\partial y^2} - 2\chi v_2 - \chi\frac{\partial u}{\partial y} = 0 \tag{2}$$

$$\frac{\partial p}{\partial y} = 0 \tag{3}$$

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \tag{4}$$

Equations (1), (2) and (4) represent conservation of linear momentum, conservation of angular momentum and conservation of mass, respectively. (u, v) represents velocity components in the directions (x, y). v_2 represents the velocity component of micro-rotation, p represents film pressure, σ represents electrical conductivity of the fluid, γ denotes the viscosity co-efficient of micro polar fluid, χ denotes the spin viscosity, μ indicates the Newtonian viscosity and B_0 denotes the magnetic field strength.

The boundary conditions (B.C) applicable for the above situation are;

For the upper plate: (y = h)

$$u = v_2 = 0,\ v - \frac{d(2h)}{dt} \tag{5}$$

For the lower plate: (y = −h)

$$u = v_2 = 0,\ v = v^* \tag{6}$$

By solving equations (1) and (2) and applying B. C's (5) and (6), the expression for the velocity component u is given by

$$u = -\frac{\frac{\partial p}{\partial x}[\omega_2 \sinh(r_2 h)\{\cosh(r_1 h) - \cosh(r_1 y)\} - \omega_1 \sinh(r_1 h)\{\cosh(r_2 h) - \cosh(r_2 y)\}]}{\sigma {B_0}^2[\omega_2 \sinh(r_2 h)\cosh(r_1 h) - \omega_1 \sinh(r_1 h)\cosh(r_2 h)]} \tag{7}$$

The nature of the flow of fluid through the porous medium was first described by Darcy who gave the expression for the velocity component (q^*) in the porous medium as

$$q^* = -\frac{k}{(\mu+\chi)}\nabla p^* \tag{8}$$

Here p^* denotes the pressure in porous region and k represents permeability. The continuity equation in the porous region is given by

$$\frac{\partial^2 p^*}{\partial x^2} + \frac{\partial^2 p^*}{\partial y^2} = 0 \tag{9}$$

The component of velocity v^* at y= − h (lower surface) is obtained as,

$$(v^*)_{y=-h} = \left(\frac{k\delta}{\mu+\chi}\right)\left\{\frac{\partial^2 p}{\partial x^2}\right\} \tag{10}$$

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The continuity equation (4) along with B. C's (5) and (6) leads to the modified Reynolds' equation which is given by

$$\frac{\partial^2 p}{\partial x^2}\left[DSR(g(h), M, L, N) + \frac{k\delta}{(\mu+\chi)}\right] = -12h^{\bullet} \quad (11)$$

Here

$g(h) = (h + p_a p'\delta)^3 + 3(\sigma^2 + \alpha^2)(h + p_a p'\delta) + 3(h + p_a p'\delta)^2\alpha + 3\sigma^2\alpha + \alpha^3 + \varepsilon$

is the deformable roughness function, depends on the standard deviation, variance, skewness, deformable parameter and ambient pressure

$$DSR(g(h), M, L, N) = \frac{\Gamma_1 - \Gamma_2}{\sigma B_0{}^2 r_1 r_2 \Gamma_3} \quad (12)$$

$$\Gamma_1 = r_2\omega_2 \sinh(r_2h)\{\cosh(r_1h)\, r_1 - \sinh(r_1h)\}$$

$$\Gamma_2 = r_1\omega_1 \sinh(r_1h)\{\cosh(r_2h)\, r_2 - \sinh(r_2h)\}$$

$$\Gamma_3 = \omega_2 \sinh(r_2h)\cosh(r_1h) - \omega_1 \sinh(r_1h)\cosh(r_2h)$$

$$r_1 = \sqrt{\left\{\frac{\lambda_1 + \sqrt{\lambda_1{}^2 - 4\lambda_2}}{2}\right\}} \qquad r_2 = \sqrt{\left\{\frac{\lambda_1 - \sqrt{\lambda_1{}^2 - 4\lambda_2}}{2}\right\}}$$

$$\omega_1 = \frac{\sigma B_0{}^2 - (2\mu+\chi)r_1{}^2}{2\chi r_1} \qquad \omega_2 = \frac{\sigma B_0{}^2 - (2\mu+\chi)r_2{}^2}{2\chi r_2}$$

$$\lambda_1 = \frac{4\mu\chi + 2\gamma\sigma B_0{}^2}{\gamma(2\mu+\chi)} \qquad \lambda_2 = \frac{4\chi\sigma B_0{}^2}{\gamma(2\mu+\chi)}$$

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

$$\psi = \frac{\varphi}{h_0{}^3} \qquad X = \frac{x}{a} \qquad p^* = p_a p' \qquad h^* = \frac{h}{h_0}$$

$$P = -\frac{h_0{}^3 p}{\mu \frac{dh_0}{dt} a^2 \cos ec\omega} \qquad W = -\frac{h_0{}^3 w}{\mu\pi \frac{dh_0}{dt} a^4 \cos ec^2\omega}$$

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

Dimensionless form of the modified Reynolds' equation is given by

$$\frac{\partial^2 p}{\partial x^2}[DSR(g(h^*), M, L, N, \psi)] = 12 \quad (13)$$

The stochastic average of equation (13) is obtained by applying expectation operation on both sides of (13) with respect to $f(h_s)$, which takes the form

$$\frac{\partial^2 E(p)}{\partial x^2}\left[E\left(DSR\{g(h^*), M, L, N, \psi\}\right)\right] = -12 \quad (14)$$

Equation (14) represents the average modified Reynolds' equation for radial roughness pattern,

$$\frac{\partial^2 E(p)}{\partial x^2}\left[\frac{1}{E\left(DSR\{g(h^*), M, L, N, \psi\}\right)}\right] = -12 \quad (15)$$

Equation (15) represents the average modified Reynolds equation for azimuthal roughness pattern. Both the roughness patterns, namely radial and azimuthal, are generally oriented in the direction of x and y, respectively. In this article, radial roughness is considered to be a one-dimensional roughness structure. The values for the azimuthal roughness can be obtained similar to the radial roughness by rotating the coordinate axes suitably,

Here

$$DSR\left(g\left(h^*\right), M, L, N, \psi\right) = \frac{24[G_1 - G_2]}{\sigma B_o{}^2 r_1{}^* r_2{}^* \{G_3 - G_4\} + [12\psi\left(\frac{1-N^2}{1+N^2}\right)}$$

$$G_1 = r_2^* \Phi_2 \sinh\left(\frac{r_2^* H^*}{2}\right)\left[\cosh\left(\frac{r_1^* H^*}{2}\right)\frac{r_1^* H^*}{2} - \sinh\left(\frac{r_1^* H^*}{2}\right)\right]$$

$$G_2 = r_1^* \Phi_1 \sinh\left(\frac{r_1^* H^*}{2}\right)\left[\cosh\left(\frac{r_1^* H^*}{2}\right)\frac{r_2^* H^*}{2} - \sinh\left(\frac{r_2^* H^*}{2}\right)\right]$$

$$G_3 = \Phi_2 \sinh\left(\frac{r_2^* H^*}{2}\right)\cosh\left(\frac{r_1^* H^*}{2}\right) \qquad G_4 = \Phi_1 \sinh\left(\frac{r_1^* H^*}{2}\right)\cosh\left(\frac{r_2^* H^*}{2}\right)$$

$$r_1^* = r_1 H_0 = \sqrt{\frac{\lambda_1^* + \sqrt{\lambda_1^{*2} - 4\lambda_2^*}}{2}} \qquad r_2^* = r_2 H_0 = \sqrt{\frac{\lambda_1^* - \sqrt{\lambda_1^{*2} - 4\lambda_2^*}}{2}}$$

$$\lambda_1^* = \lambda_1 H_0^2 = \frac{N^2 + M^2(1-N^2)L^2}{L^2} \qquad \lambda_2^* = \lambda_2 H_0^2 = \frac{N^2 M^2}{L^2}$$

$$\omega_1^* = \omega_1 H_0 = \frac{M^2(1-N^2) - r_1^{*2}}{2N^2 r_1^{*2}} \qquad \omega_2^* = \omega_2 H_0 = \frac{M^2(1-N^2) - r_2^{*2}}{2N^2 r_2^{*2}}$$

$$N = \sqrt{\frac{\chi}{2\mu + \chi}} \qquad L = \frac{\sqrt{\frac{\chi}{4\mu}}}{H_0} \qquad M = B_0 H_0 \sqrt{\frac{\sigma}{\mu}}$$

Where the dimensionless variables are used in the above expressions are

$$x^* = \frac{x}{A} \qquad H^* = h^* + hs \qquad h^* = \frac{h}{h_0} = \frac{2h}{H_0}$$

$$r_1^* = r_1 H_0 \qquad r_2^* = r_{21} H_0 \qquad \omega_1^* = \omega_1 H_0$$

$$\omega_2^* = \omega_2 H_0 \qquad \Psi = \frac{k\delta}{H_0^3}$$

Combining equation (14) and equation (15) gives

$$\frac{\partial^2 E(p)}{\partial x^2}\left[S\left(DSR\{g(h^*), M, L, N, \psi, c\}\right)\right] = -12 \qquad (16)$$

Here,

$$S\left(DSR\{g(h^*), M, L, N, \psi, c\}\right) = \left\{\begin{matrix} E\left[DRSR^*\{g(h^*), M, L, N, \psi\}\right] \\ E\left[\left(\frac{1}{DRSR^*\{g(h^*), M, L, N, \psi\}}\right)^{-1}\right] \end{matrix}\right\}$$

The pressure boundary conditions for a triangular plate are as follows p(x′, y′) = 0 where

$$(x'-a)\left(x'-\sqrt{3y'}+2a\right)\left(x'+\sqrt{3y'}+2a\right) = 0$$

a is the length of the equilateral triangle whose equation is

$$(x-a)\left(x-\sqrt{3y}+2a\right)\left(x+\sqrt{3y}+2a\right) = 0$$

The pressure in the dimensionless form is obtained as

$$P^* = 0.77\left[\frac{(1-x)\left(1-\frac{\sqrt{3}y}{2}+\frac{x}{2}\right)\left(1+\frac{\sqrt{3}y}{2}+\frac{x}{2}\right)}{S\left(DSR\left\{g(h^*),M,L,N,\psi,c\right\}\right)}\right] \tag{17}$$

The expected workload is obtained by

$$E(w)=\int_{-2a}^{a}\int_{\frac{-2a+x}{\sqrt{3}}}^{\frac{2a+x}{\sqrt{3}}} E(p)\,dy\,dx$$

The load profile in the dimensionless form is given by,

$$W^* = \frac{E(w)h_0^{\ 3}}{27\mu h_0 a^2} = \frac{\sqrt{3}}{5}\left[\frac{1}{S\left(DSR\left\{g(h^*),M,L,N,\psi,c\right\}\right)}\right] \tag{18}$$

The expression for the time-height relation in the dimensionless form is given by,

$$T^* = -\frac{E(w)h_0^{\ 2}}{27\mu a^2}dt = \frac{\sqrt{3}}{5}\int_{H_1^*}^{1}\left[\frac{dH^*}{S\left(DSR\left\{g(h^*),M,L,N,\psi,c\right\}\right)}\right] \tag{19}$$

III. RESULTS AND DISCUSSION

This article studies about a pair of triangular plates that squeezes out the micro polar fluid present between them as they approach each other in the presence of MHD. The effects of porosity and surface roughness are given due interest while analyzing the effects of MHD. The coupling number N and length L, which characterize the contact between fluid clearance, are two dimensionless characteristics that describes the nature of the micro polar fluid. Comparison is made between the effect created by the presence and the absence of magnetic field M. The effect of porosity Ψ and the surface roughness parameter c on the triangular plates are also analyzed and discussed below. The effect produced by the presence and absence of MHD and roughness has been brought out both using graphs and tables. Instead of carrying out the research work for analyzing the MHD effect separately and roughness effect separately, as usually done by researches, this paper aims to consider both these effects together. The interdependence of these two effects in pressure, workload and time height has been brought out by the graphs and tables. Hence an ideal situation of working of the bearing is considered for the analysis and the results are etched.

A. *Fruitful Conclusions*

The influence of squeezing action on micro polar fluid in the presence of an externally applied magnetic field is considered in this article. The basic for this fluid model was developed by Eringen, which is applied to the considered geometry of triangular plates, which are rough in nature and are backed by a porous facing at the lower surface. The pressure distribution is obtained by solving the Reynolds equation and the load-carrying capacity is obtained by integrating the pressure over the film region. The workload is intensified by the application of MPF as a lubricant, which is characterized by the coupling number N and the fluid clearance gap L. The induced magnetic field enhances the workload in contrast to the non-magnetic scenario of a similar situation. The presence of porosity reduces the workload, while the effect of surface roughness increases the load compared to the smooth case by 9.05%, as illustrated in Table V. Additionally, it is observed that the squeeze time increases with higher values of the coupling number N, clearance gap L, Hartmann number M, in comparison to the non-magnetic case, and with the roughness parameter c, in comparison to the smooth case, by 20.45%. By considering the above observations an ideal bearing lubricant can maximize the performance of the plate. The research and technological advancements discussed suggest that the integration of non-Newtonian fluids like MPF and the application of MHD principles have significant potential to enhance lubrication systems. These innovations improve load-carrying capacity and reduce wear on machine parts, especially when accounting for surface roughness and interactions between the lubricant and the surfaces involved. The study of squeeze film lubrication, particularly in complex geometries like triangular plates, is an exciting area with practical applications in various machinery and power plant systems.

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This passage presents a comprehensive overview of various studies and advancements in the field of squeeze film technology, particularly in the context of lubrication using non-Newtonian fluids such as micro polar fluids (MPF) and magnetohydrodynamics (MHD). Below is a more organized summary and analysis of the key ideas:

A. *Squeeze Film Technology and Lubrication*

Squeeze film lubrication plays a critical role in various industrial systems, from small machinery parts like disc clutches to large systems in power plants, where lubricants are squeezed into rotating parts to reduce friction and wear.
The development of modern machine tools has led to increased application of non-Newtonian fluids in these systems. Non-Newtonian fluids, unlike traditional fluids, have variable viscosity that depends on factors such as shear rate.

B. *Non-Newtonian Fluids*

Micro polar Fluids (MPF): MPF are a type of non-Newtonian fluid with unique characteristics due to the presence of micro-rotations in the fluid's particles. This behavior leads to enhanced lubrication properties, making MPF ideal for increasing the load-carrying capacity in various lubrication systems.
The application of MPF has been studied extensively in different geometrical structures, such as parallel plates and hemispherical bearings. These studies have demonstrated that MPF increases load capacity due to its unique rheological properties.

C. *Magnetohydrodynamics (MHD)*

The study of MHD concerns the behavior of conducting fluids under the influence of a magnetic field. This theory, initiated by Hartmann, explores the effects of a magnetic field on the flow characteristics of an incompressible fluid between parallel planes, known as Hartmann flow.
Recent studies (such as those by Anncy et al., Salah et al., and M. Hamza) have shown that applying an external magnetic field improves the load-carrying capacity of lubricating systems, particularly in magnetic fluids.

D. *Surface Roughness and Lubricant Interaction*

As lubricants interact with machine parts, their properties change due to the additives within them, which can convert a Newtonian fluid into a non-Newtonian fluid. These interactions can lead to surface changes that contribute to wear and tear on the machinery if not properly managed.
Researchers have studied the impact of surface roughness on bearing performance, with stochastic models developed by Christensen and others to examine how roughness affects the effectiveness of lubricants.
The interaction between lubricants and surface textures has been explored for both smooth and rough bearing surfaces, with findings indicating that surface roughness, along with the application of MHD, significantly enhances load-carrying capacity.

E. *Wet Clutches and Squeeze Film Lubrication*

The motivation behind this work was inspired by wet clutches, which are used to cool the clutch pack by circulating lubricant through grooves on the clutch surfaces. These grooves are typically triangular in shape, a configuration found in various other mechanical systems as well. The study presented in the passage focuses on investigating squeeze film lubrication between triangular plates, particularly under the influence of an external magnetic field and when lubricated with MPF. The effect of surface roughness on these systems was also of particular interest, given its potential impact on the overall performance of the lubrication system.