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Abstract: *The rough circular stepped plate, which is influenced by the effect of MHD and for this study, micro-polar fluid is used as a lubricant. The governing equations are that taken up for analysis are based on Eringen's micropolar theory. To estimate the effects of surface roughness, Christens theory is applied for the present analysis. The fluid film pressure is derived using the modified Reynolds equation. The expression of the load sustaining capacity of the circular stepped plate is estimated based on the pressure that is developed in the film region. The roughness of the plate decreases the pressure and load sustaining capacity, but the influence of MHD increases the pressure and load sustaining capacity. The relative percentage of the load sustaining capacity is analyzed based on roughness and magnetic field.*

Keywords: *Micropolar fluid, Circular stepped plate, Surface roughness, MHD*

Introduction

Squeeze film lubrication finds application across various sectors, including clutches, brakes, gears, and machine tools. Squeeze film bearings offer significant utility by facilitating high-pressure generation and increased load sustaining capacity through surface compression. Many investigations have predominantly focused on squeeze film bearings lubricated with Newtonian fluids. Cameron [1] conducted an analysis of squeeze film lubrication between two parallel plates extending infinitely. Jackson [2] explored the characteristics of squeeze film lubrication utilizing Newtonian lubricants. The dynamics of squeezing flow between parallel plates were investigated by Gupta AS and Gupta PS [3]. Burbidge and Servais [4] examined the behavior of apparently lubricated thin films under squeeze flow conditions. Bujurke et al. [5] developed the study of incompressible fluid flow between parallel plates induced by the normal motion of a plate. Naduvinamani et al. [7] worked on the paper, this makes use of the Rabinowitsch fluid model to estimate the lubricant's impact on squeezing film lubrication between circular stepped plates by utilizing its non-Newtonian characteristics. Naduvinamani and Rajashekar [8] analyzed the concepts of squeeze velocity and sliding velocity in inclined stepped composite bearings. S. Rao and Rahul [21] worked on a wide, parallel, rectangular-porous plate with squeeze-film characteristics.

Many researchers consider Eringen's micropolar fluid theory as a subset of microfluids. Micropolar fluid models serve as valuable tools for simulating the behavior of lubricating oils containing various contaminants and additives. According to Roopa et al. [13], the presence of micropolar fluid has a substantial impact on the squeeze film properties of elliptical plates. Sangeetha et al.[9] investigated the impact of magnetohydrodynamics (MHD) on a rough conical bearing in the presence of a micropolar fluid. Hanumagowda et al. [6] worked on a rough porous

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circular stepped plate lubricated with micropolar fluid, where they discovered that the bearing properties, such as dimensionless pressure, load sustaining capacity and non-dimensional squeezing time, show a progressive increase with increasing coupling number and micropolar parameter.

Magnetohydrodynamics (MHD) explores the intricate interplay between electromagnetic forces and conducting fluids, constituting a scientific domain of interest. The field of magnetohydrodynamic lubrication theory has undergone significant growth in recent years. Various academic investigations have focused on electrically conducting fluids, drawing from Cowling's [10] hydromagnetic flow theory. Sasada et al. pivotal research [11] extensively explored multiple (MHD) bearing models and their practical and experimental applications.The impact of MHD on the curved circular plate in the flat, rough porous plate was examined by Jayaprakash et al. [14]. In a recent study, Asha and colleagues [12] investigated the influence of a magnetic field and micro-polar fluid on wide, parallel rectangular plates. Their findings revealed a notable impact of the magnetic field on these plates, resulting in enhanced lubricating properties compared to non-magnetic conditions. Vinutha et al. [15] examined the properties of a long cylinder on an infinitely rough plate using MHD, and numerous other researchers applied MHD while working with various plate and fluid combinations.

With the assumption of perfectly smooth bearing surfaces, the studies predominantly emphasize the performance of squeeze films. However, in engineering practice, the forming and finishing stages of the process often yield surfaces with roughness. In lubricated contacts, it's common for the roughness height to align with the mean asperity. Patir and Cheng [16],[17] developed a model to analyze the impact of surface roughness, utilizing statistically defined properties of randomly generated rough surfaces. This approach contrasts the averaged flow characteristics between seemingly smooth surfaces with the actual flow dynamics between rough surfaces. Christensen [18],[19] introduced an alternate film averaging technique specifically tailored for surfaces with striated roughness, employing stochastic principles. Additionally, Christensen and Tonder [20] developed random models for hydrodynamic lubrication on rough surfaces and expanded Reynold's equation to incorporate rough bearings. Udaya [22] explored the performance of hydrostatic rough thrust bearings. They observed that longitudinal roughness patterns lead to a decrease in load sustaining capacity with increasing roughness, while circumferential roughness patterns result in an increase in load capacity as roughness increases. Naduvinamni and Angad [23] conducted an analysis of the static and dynamic characteristics of rough porous Rayleigh step bearings. Their findings indicated that the presence of a negatively skewed surface roughness structure resulted in a reduction of the volume flow rate. In their study, Halambi et al. [24] examined the effects of magnetic fields and surface roughness on the micropolar squeeze film lubrication between rough porous elliptical plates. Faizan Ahmed and Sujatha [25] worked together to look into how surface roughness, magnetohydrodynamics, and micropolar fluid affect each other when they are put between porous triangular plates. The effect of magnetohydrodynamics with micropolar fluid as a lubricant between stepped porous parallel plates was studied by Naduvinamani and Angadi [26].

The current study analyzes a rough circular stepped plate under the influence of MHD and a microplar fluid as a lubricant, a topic not previously explored in previous studies. The modified Reynolds equation for surface roughness is derive and used to estimate the pressure. The main motive of this work is to estimate the fluid film pressure and load sustaining capacity for the considered model. The fluid film pressure and load sustaining capacity are analysed by varying the various parameters like fluid film thickness h_1^*, Hartman number M, coupling number N, surface roughness parameter c, characteristic length of the micropolar fluid L and the plate's step height K. The results of the analysis are represented graphically, and the comparison between the radial roughness and azimuthal roughness is tabulated.

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Geometry of the Bearing

The rough circular stepped plates is physically illustrated in Figure (1). The upper plate is moving toward the lower plate with squeezing velocity $V = \frac{dh}{dt}$. The micropolar fluid in the film region of the circular stepped plates is used as a lubricant. There is a step in the plate with a step height K and the film regions are divided into two parts, h_2 is the minimum film thickness and h_1 is the maximum. In this case, it is considered that the lower plate is rough. The applied magnetic field B_0 is applied perpendicular to the fluid flow direction of the circular stepped plate. Assuming the basic assumptions of thin film lubrication, to hold true the governing equations pertaining to a micropolar fluid, are given by:

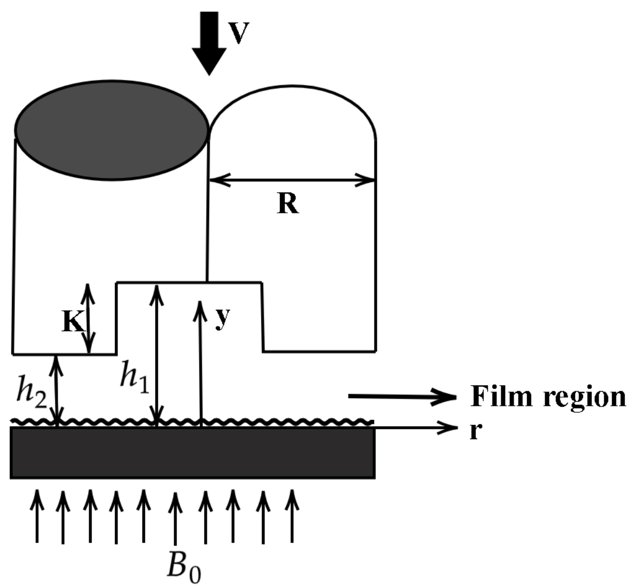

Figure 1: Geometry of rough circular stepped plate

$$\left(\frac{2\mu+\chi}{2}\right)\frac{\partial^2 u}{\partial y^2} + \chi\frac{\partial v_1}{\partial y} - \sigma B_0^2 u = \frac{\partial p}{\partial r} \tag{1}$$

$$\gamma\frac{\partial^2 v_1}{\partial y^2} - 2\chi v_1 - \chi\frac{\partial u}{\partial y} = 0 \tag{2}$$

$$\frac{1}{r}\frac{\partial(ru)}{\partial r} + \frac{\partial v}{\partial y} = 0 \tag{3}$$

$$\frac{\partial p}{\partial y} = 0 \tag{4}$$

In the equations mentioned the velocity components in the directions of r and y are represented by the variables u and v respectively, v_1 is micro-rotational velocity, B_0 denotes strength of the magnetic field and σ represents the electrical conductivity of the fluid, μ represent the viscosity of the micro-polar fluid, χ its spin viscosity and γ its viscosity coefficient. The film thickness H between the plates is of the following form

$$H = \begin{cases} h_1 for \ \ 0 \le r \le KR \\ h_2 for \ \ \mathrm{KR} \le r \le R \end{cases} \tag{5}$$

For the velocity components, the necessary boundary conditions are:

(i) On the lower surface $(y = 0)$

$$u = 0, v = 0, v_1 = 0 \tag{6}$$

(ii) On the upper surface $(y = H)$

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$$u = 0, v = V, v_1 = 0 \tag{7}$$

On solving the equations (1) and (2) by using boundary conditions (6) and (7), the velocity component u is obtained as

$$u = \frac{a-b}{\sigma B_0^2[c-d]} \frac{\partial p}{\partial r} \tag{8}$$

where

$$a = g_1 sinh(k_1 H)[cosh(k_2 H) - cosh(k_2 y)] b = g_2 sinh(k_2 H)[cosh(k_1 H) - cosh(k_1 y)] c = g_2 sinh(k_2 H) cosh(k_1 H) d = g_1 sinh(k_1 H) cosh(k_2 H)$$

By using the expression of u and integrating the equation (3) w.r.to y over the film thickness, the modified Reynold's equation is obtained as

$$\frac{\partial}{\partial r}[f(H, N, L, M) r \frac{\partial p}{\partial r}] = -Vr \tag{9}$$

The non-dimensional form the modified Reynold's equation is

$$\frac{\partial}{\partial r^*}[F(H, N, L, M) r^* \frac{\partial \overline{p}}{\partial r^*}] = -r^* \tag{10}$$

where $F(H, N, L, M) = \frac{a_1 - b_1}{M^2 \overline{k_1 k_2}(c_1 - d_1)}$,

$a_1 = \overline{g_1}\,\overline{k_1} sinh(\frac{\overline{k_1}H}{2})(\overline{k_2} H cosh(\frac{\overline{k_2}H}{2}) - 2 sinh(\frac{\overline{k_2}H}{2}))$,

$b_1 = \overline{g_2}\,\overline{k_2} sinh(\frac{\overline{k_2}H}{2})(\overline{k_1} H cosh(\frac{\overline{k_1}H}{2}) - 2 sinh(\frac{\overline{k_1}H}{2}))$,

$c_1 = \overline{g_1} cosh(\frac{\overline{k_2}H}{2})(sinh(\frac{\overline{k_1}H}{2}))$,

$d_1 = \overline{g_2} cosh(\frac{\overline{k_1}H}{2})(sinh(\frac{\overline{k_2}H}{2}))$,

$\overline{g_1} = \frac{M^2(1-N^2) - \overline{k_1}^2}{2N^2 \overline{k_1}}$, $\overline{g_2} = \frac{M^2(1-N^2) - \overline{k_2}^2}{2N^2 \overline{k_2}}$,

$\overline{k_1} = \sqrt{\frac{\overline{\alpha} + \sqrt{\overline{\alpha}^2 - 4\overline{\beta}}}{2}}$, $\overline{k_2} = \sqrt{\frac{\overline{\alpha} - \sqrt{\overline{\alpha}^2 - 4\overline{\beta}}}{2}}$,

$\overline{\alpha} = \frac{N^2 + M^2(1-N^2)L^2}{L^2}$, $\overline{\beta} = \frac{N^2 M^2}{L^2}$,

$M = B_0 h_2 \sqrt{\frac{\sigma}{\mu}}$, $L = \frac{\sqrt{\frac{\gamma}{4\mu}}}{h_2}$, $N = \sqrt{\frac{\chi}{2\mu + \chi}}$.

Here the non-dimensional quantities are as follows

$$r^* = \frac{r}{R}, \overline{\alpha} = h_2^2 \alpha, \overline{\beta} = h_2^4 \beta, \overline{g_1} = h_2 g_1, \overline{g_2} = h_2 g_2, P = \frac{E(\overline{p}) h_2^3}{\mu R^2 V}$$

where M is Hartmann number, N is dimensionless coupling number, L is characteristic material length. The roughness aspect of the circular plates are brought out by describing the film region H as a sum of two parts that is $H = h(x) + h_s$, where h_s represents the randomly varying thickness quantity that is determined from the mean level and thus describes the surface roughness, and $h(x)$ is the mean film thickness.

The stochastic Reynolds equation is taken up in the form by taking expectation on both sides of equation (10) and applying the Christensen stochastic approach for the surface roughness.

$$\frac{\partial}{\partial r^*}\{(E[F(H, N, L, M)]) r^* \frac{\partial E(\overline{p})}{\partial r^*}\} = -r^* \tag{11}$$

$$E(*) = \int_{-\infty}^{\infty} (*) f(h_s) dh_s$$

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where the selected random variable h_s's PDF $f(h_s)$ is defined as

$$f(h_s) = \begin{cases} \frac{35}{32}\frac{(c^2-h_s)^3}{c^7} & if \quad -c \le h_s \le c \\ 0 & elsewhere \end{cases}$$

where the highest deviation from the average film thickness is denoted by c. Two roughness striations radial roughness pattern and azimuthal roughness pattern are of interest in the current study from a theoretical perspective of the Christensen stochastic approach to rough surfaces.

Radial and Azimuthal Roughness

The geometry of the bearing surface has long and narrow ridges and valleys that extend in a radial direction for radial one-dimensional roughness. Thus, there are two parts to the film thickness $H = h + h_s(r,\zeta)$ and equation (11) takes the form

$$\frac{\partial}{\partial r^*}\{(E[F(H,N,L,M)])r^*\frac{\partial E(\overline{p})}{\partial r^*}\} = -r^* \tag{12}$$

The geometry of the bearing surface consists of long, narrow ridges and valleys that extend in the azimuthal direction for the azimuthal one-dimensional roughness. As a result, there are two parts to the film thickness. $H = h + h_s(y,\zeta)$, and equation (11) takes the form

$$\frac{\partial}{\partial r^*}\{(E[1/F(H,N,L,M)])^{-1}r^*\frac{\partial E(\overline{p})}{\partial r^*}\} = -r^* \tag{13}$$

Combining equations (12) and (13) the modified Reynold's equation is got as

$$\frac{\partial}{\partial r^*}\{(G(H,N,L,M,c))r^*\frac{\partial P}{\partial r^*}\} = -r^* \tag{14}$$

where,

$$G(H,N,L,M,c) = \begin{cases} E[F(H,N,L,M)] & Radialroughness \\ (E[1/F(H,N,L,M)])^{-1} & Azimuthalroughness \end{cases}$$

The related boundary conditions for pressure are given below,

$$P_1 = P_2 at r^* = K \tag{15}$$

$$P_2 = 0\ at r^* = 1 \tag{16}$$

On integrating equation(14) using suitable boundary conditions (15) and (16) the dimensionless expression of pressure is obtained as

$$P_1 = \frac{K^2 - r^{*2}}{4G_1(h_1^*,N,L,M,c)} + \frac{1-K^2}{4G_2(1,N,L,M,c)} \tag{17}$$

$$P_2 = \frac{1-r^{*2}}{4G_2(1,N,L,M,c)} \tag{18}$$

The load sustaining capacity w is obtained by using equation (17) and (18).

$$w = 2\pi\int_0^{KR} r(p_1)dr + 2\pi\int_{KR}^{R} r(p_2)dr \tag{19}$$

The dimensionless load sustaining capacity W is of the form

$$W = \frac{K^3}{G_1(h_1^*,N,L,M,c)} + \frac{1-K^3}{G_2(1,N,L,M,c)} \tag{20}$$

Results and Discussions

The article analyzes the effects of surface roughness and MHD on a circular stepped plate,

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lubricated with micropolar fluid. The radial and azimuthal roughness are estimated using the Christens methodology. The variations of the parameters like bearing film thickness h_1^*, Hartmanns number M, coupling number N, bearing step height K and roughness parameters c helps to estimate the circular stepped plate's fluid film pressure and load sustaining capacity. The analysis on the relative percentages for load sustaining capacity, magnetic field strength, and surface roughness is also carried out. The outcomes are represented graphically and by tables.

Dimensionless Pressure

Figures (2) to (6) depict the variations of dimensionless pressure P for the radial roughness case. Figures (2), (5), and (6) show the variation between the dimensionless pressure P and axial distance r^* for different values of fluid film thickness h_1^*, characteristic length of micropolar fluid L and surface roughness parameter c respectively. It is observed that the dimensionless pressure decreases while increasing the parameters h_1^*, L and c. Figure 3 and 4 show the variation between the dimensionless pressure P and axial distance r^* for different values of Hartman number M and coupling number N respectively. There is an increase in the dimensionless pressure as the parameters M and N increases.

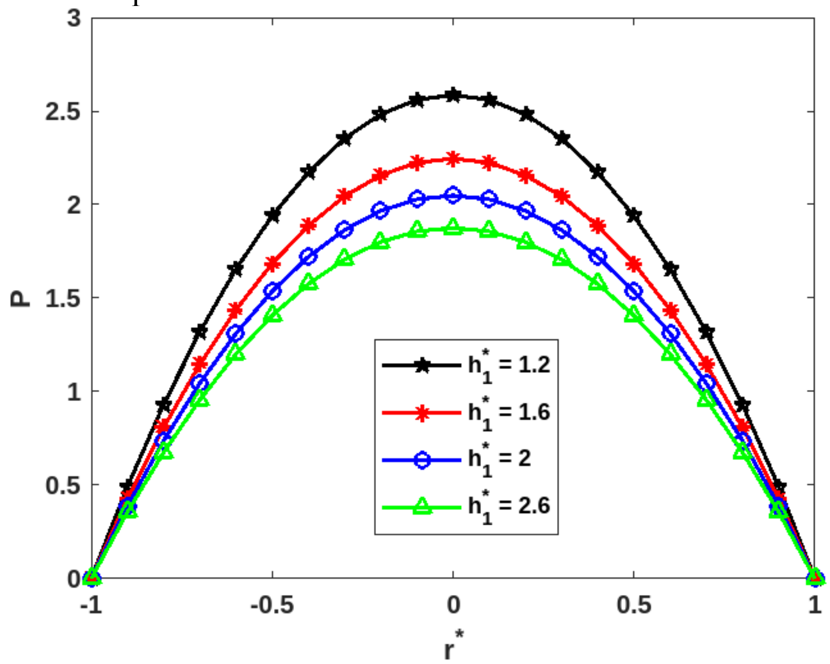

Figure 2: Plot of P with r^* for different values of h_1^*

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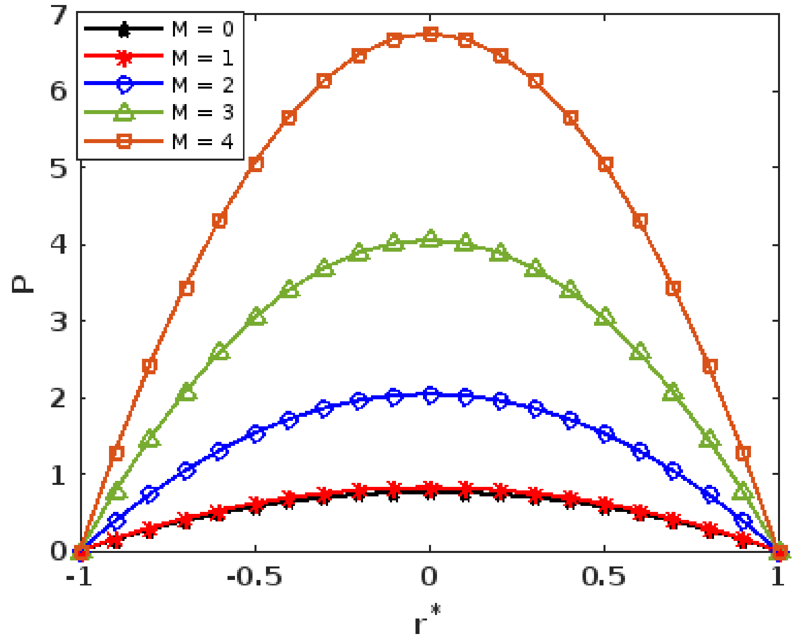

Figure 3: Plot of P with r^* for different values of M

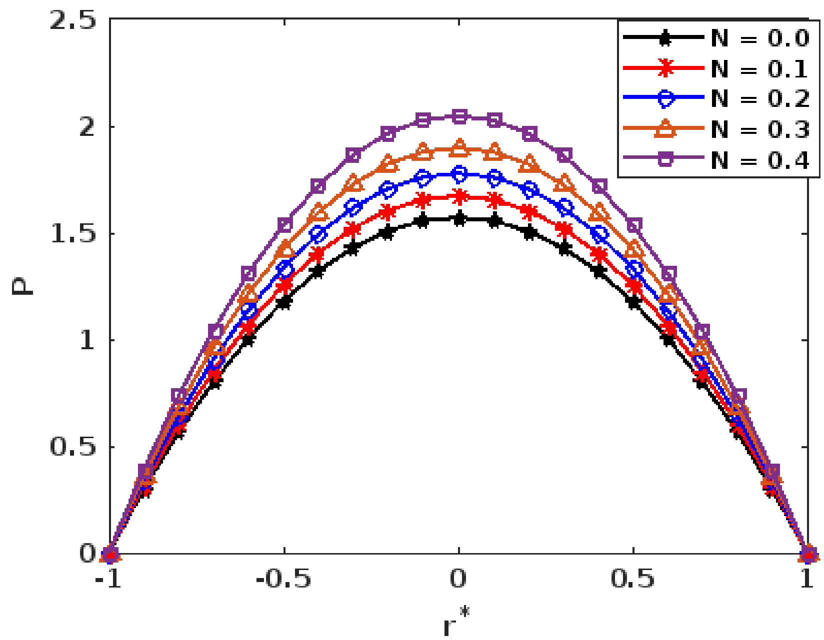

Figure 4: Plot of P with r^* for different values of N

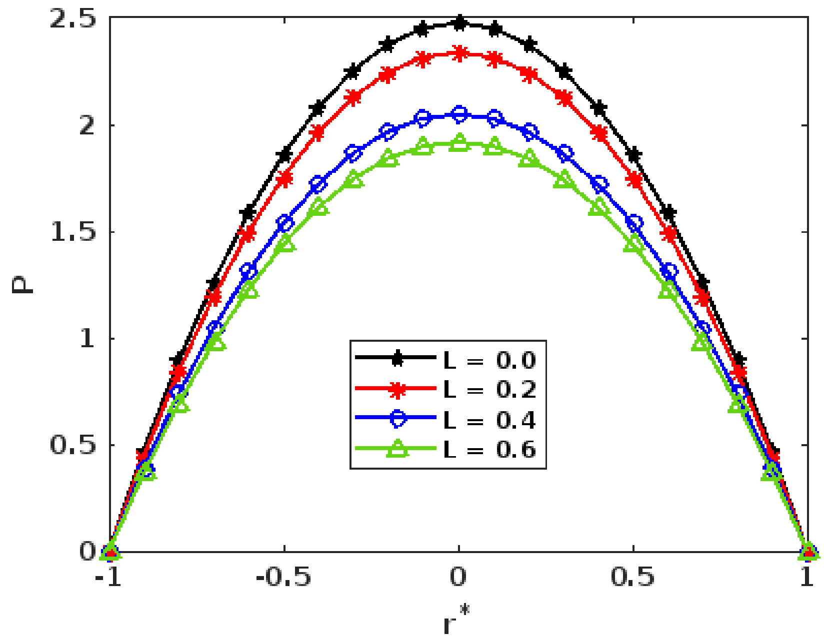

Figure 5: Plot of P with r^* for different values of L

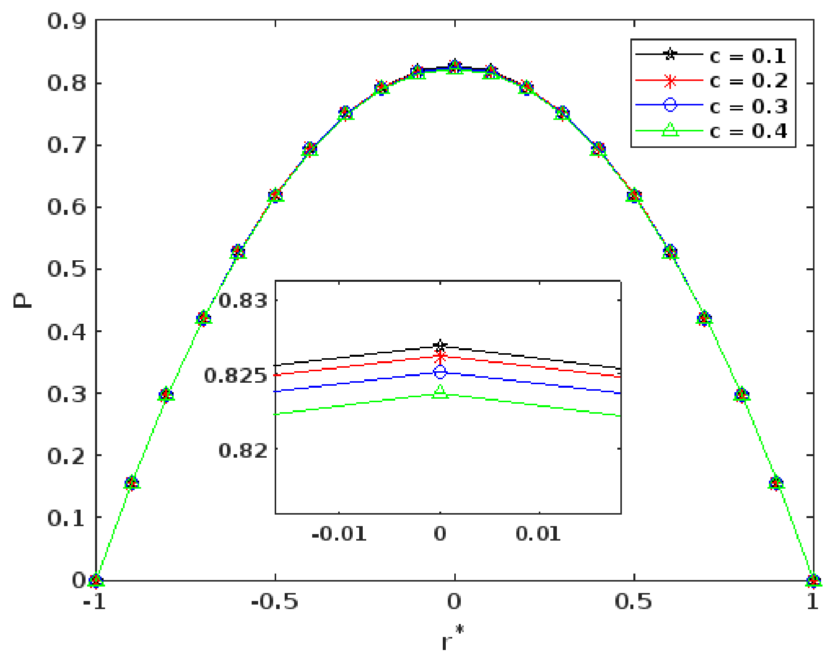

Figure 6: Plot of P with r^* for different values of c

load sustaining capacity

Figure (7) to figure (10) depict the variations of dimensionless load sustaining capacity W for the radial roughness case. Figure (7) and (8) show the variation between the dimensionless load sustaining capacity W and fluid film thickness h_1^* for various values of the surface roughness parameter c and the plate's step height K respectively. It is noted that the dimensionless load sustaining capacity decreases while increasing the parameters c and K. Figure

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(9) and (10) show the variation between the dimensionless load sustaining capacity W and axial distance h_1^* for different values of Hartman number M and coupling number N respectively. It is observed that the dimensionless load sustaining capacity increases while increasing the parameters M and N. The increasing and decreasing of the plate's load sustaining capacity depends on the pressure that develops in the film region. If the film pressure is high, the load sustaining capacity is also high. If the fluid film pressure decreases in the film region, the load sustaining capacity also decreases.

Tables 1, 2, 3 and 4 depict the comparison between the surface roughness ceases which are radial roughness and azimuthal roughness of the load sustaining capacity W for different values of N, M, c and K. It is observed from the tables that azimuthal roughness has a greater impact on the load sustaining capacity than radial roughness.

Estimation of relative percentage of workload

The relative percentage of the dimensionless load sustaining capacity W is analysed based on roughness case which is represented as R_{Wc} and for the case of MHD it represented as R_{WM}, which are defined as

$$R_{Wc} = \left(\frac{W_{roughness} - W_{non-roughness}}{W_{non-roughness}}\right) \times 100$$

$$R_{WM} = \left(\frac{W_{magnetic} - W_{non-magnetic}}{W_{non-magnetic}}\right) \times 100$$

The out comes of R_{Wc} and R_{WM} are given in the table 5 for different values of coupling number $N = 0.1, 0.2, 0.3$ and film thickness $h_1^* = 1.2, 1.4, 1.6, 1.8, 2, 2.2, 2.4$. It is observed from that the R_{Wc} increases when $N = 0.1$ other then it decreases. It noted from R_{Wc} the highest relative percentage is 806.903, when $N = 0.1$ and $h_1^* = 2$.

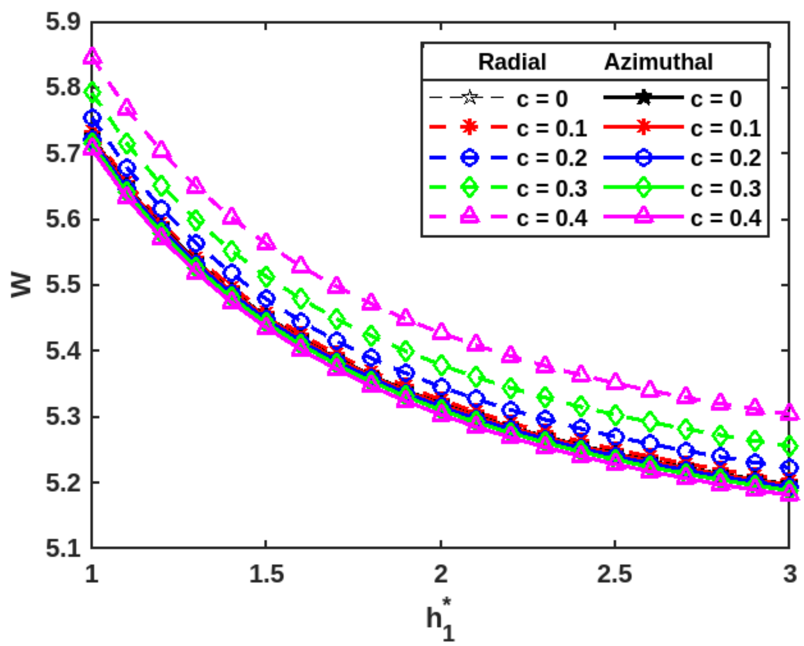

Figure 7: Plot of W with h_1^* for different values of c

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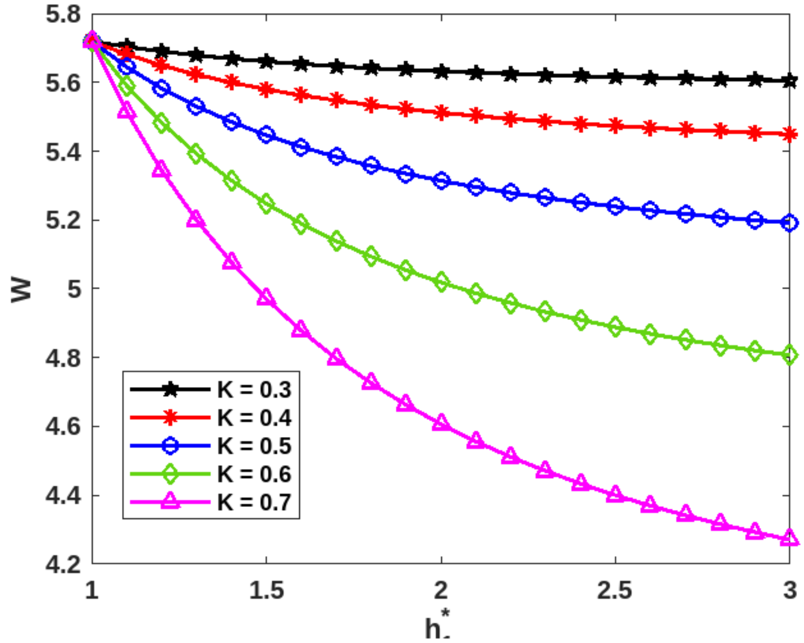

Figure 8: Plot of W with h_1^* for different values of K

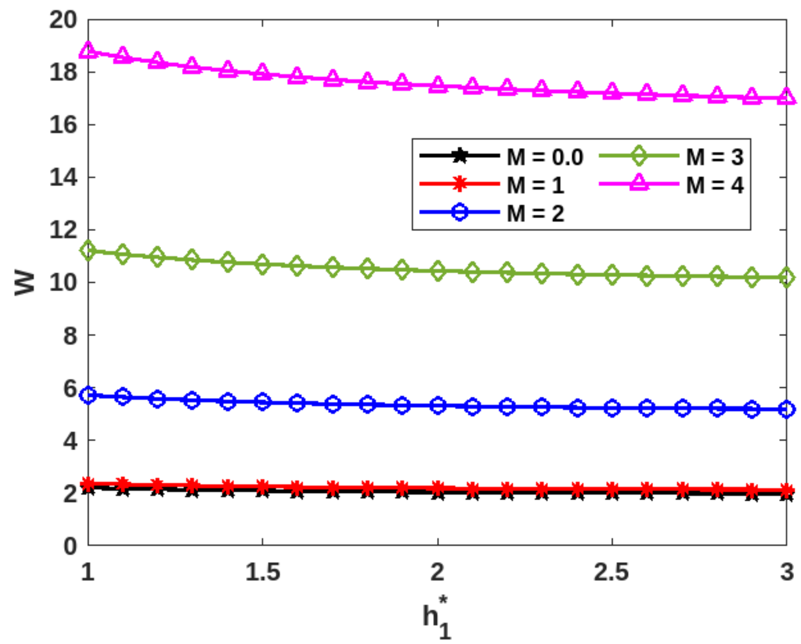

Figure 9: Plot of W with h_1^* for different values of M

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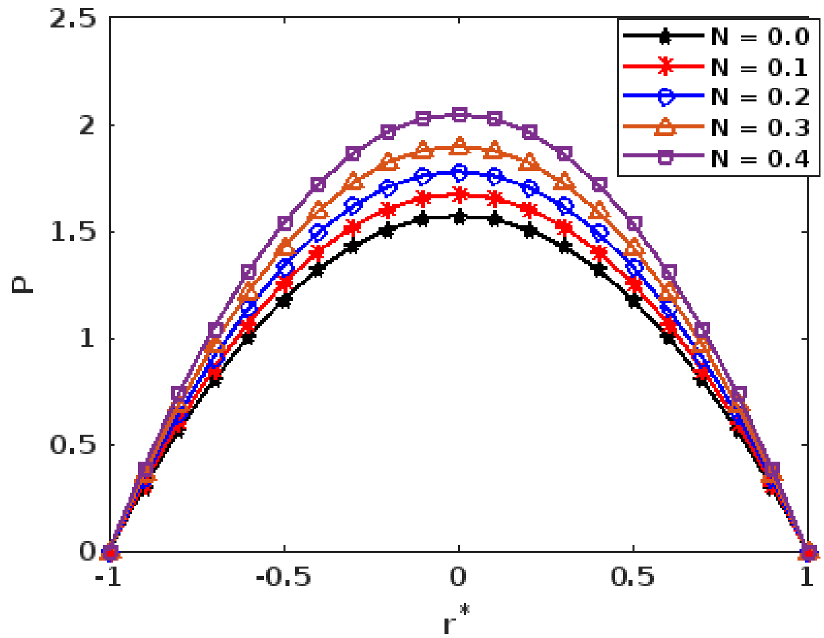

Figure 10: Plot of W with h_1^* for different values of N

Table 1: Comparison between radial and azimuthal roughness for load caring capacity with different values of N

h_1^*	Radial roughness			Azimuthal roughness		
	***N*=0.0**	***N*=0.2**	***N*=0.4**	***N*=0.0**	***N*=0.2**	***N*=0.4**
1.2	4.0544	4.6963	5.5829	4.0702	4.7177	5.6151
1.4	3.9958	4.6226	5.4854	4.0112	4.6434	5.5167
1.6	3.9515	4.567	5.4131	3.9666	4.5875	5.4438
1.8	3.9165	4.5235	5.3577	3.9315	4.5439	5.3881
2	3.888	4.4885	5.3142	3.9029	4.5088	5.3444
2.2	3.864	4.4597	5.2794	3.8789	4.4798	5.3095
2.4	3.8435	4.4355	5.2511	3.8583	4.4555	5.281

Table 2: Comparison between radial and azimuthal roughness for load caring capacity with different values of M

h_1^*	Radial roughness			Azimuthal roughness		
	***M*=0**	***M*=2**	***M*=4**	***M*=0**	***M*=2**	***M*=4**
1.2	2.1461	5.5829	18.3509	2.1597	5.6151	18.4477

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1.4	2.1062	5.4854	18.0397	2.1193	5.5167	18.134
1.6	2.0761	5.4131	17.8037	2.0891	5.4438	17.8965
1.8	2.0528	5.3577	17.6176	2.0656	5.3881	17.7095
2	2.0344	5.3142	17.4663	2.0471	5.3444	17.5576
2.2	2.0196	5.2794	17.3402	2.0322	5.3095	17.4311
2.4	2.0075	5.2511	17.2331	2.0201	5.281	17.3237

Table 3: Comparison between radial and azimuthal roughness for load caring capacity with different values of c

h_1^*	**Radial roughness**			**Azimuthal roughness**		
	$c = 0.0$	$c = 0.2$	$c = 0.4$	$c = 0.0$	$c = 0.2$	$c = 0.4$
1.2	5.5872	5.5829	5.5699	5.5872	5.6151	5.7021
1.4	5.4897	5.4854	5.4728	5.4897	5.5167	5.6013
1.6	5.4172	5.4131	5.4007	5.4172	5.4438	5.5271
1.8	5.3618	5.3577	5.3455	5.3618	5.3881	5.4705
2	5.3182	5.3142	5.3021	5.3182	5.3444	5.4263
2.2	5.2834	5.2794	5.2673	5.2834	5.3095	5.391
2.4	5.2551	5.2511	5.2391	5.2551	5.281	5.3623

Table 4: Comparison between radial and azimuthal roughness for load caring capacity with different values of K

h_1^*	**Radial roughness**			**Azimuthal roughness**		
	K=0.5	K=0.6	K=0.7	K=0.5	K=0.6	K=0.7
1.2	5.5829	5.4832	5.3441	5.6151	5.5141	5.3733
1.4	5.4854	5.3148	5.0767	5.5167	5.3442	5.1034
1.6	5.4131	5.1898	4.8781	5.4438	5.2182	4.9034
1.8	5.3577	5.094	4.7261	5.3881	5.1219	4.7504
2	5.3142	5.0189	4.6068	5.3444	5.0464	4.6306
2.2	5.2794	4.9588	4.5113	5.3095	4.986	4.5347
2.4	5.2511	4.9099	4.4336	5.281	4.9369	4.4567

Table 5: Change in R_{WC} and R_{WM} for different values of N and h_1^*

N	h_1^*	R_{Wc}	R_{WM}
0.1	1.2	0.00458	806.146
	1.4	0.00465	806.644
	1.6	0.00471	806.701
	1.8	0.00475	806.894
	2	0.00479	806.903
	2.2	0.00482	806.881
	2.4	0.00484	806.542
0.2	1.2	-0.0149	590.937
	1.4	-0.0151	591.178
	1.6	-0.0153	591.446

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	1.8	-0.0177	591.667
	2	-0.0178	591.816
	2.2	-0.0157	591.963
	2.4	-0.0158	592.074
0.3	1.2	-0.0413	373.682
	1.4	-0.042	374.135
	1.6	-0.0425	374.524
	1.8	-0.0409	374.91
	2	-0.0412	375.306
	2.2	-0.0415	375.612
	2.4	-0.0417	375.901

Conclusions

The present analysis is on the effect of surface roughness on a circular stepped plate that is lubricated with micro polar fluid and is influenced by MHD. The expected modified Reynolds equation is used to derive the expression for fluid film pressure P and load sustaining capacity W. The efficient pressure and load sustaining capacity of the rough circular stepped plate are obtained by varying the parameters like Hartman number$M$, coupling number N, characteristic material length L, film thickness h_1^*, roughness parameter c and step height K. The comparisons between the radial and azimuthal roughness is tabulated. The relative percentage of load sustaining capacity is analyzed, and the results are tabulated.

• The Hartman number M and coupling number N are directly proportional to the pressure P and load sustaining capacity W.

• The surface roughness c, step height K and characteristic material length L are indirectly proportional to the pressure P and load sustaining capacity W

• The load sustaining capacity W increases with increasing radial roughness and decreases with increasing azimuthal roughness when the roughness parameter c is increased.

• The azimuthal roughness has greater load sustaining capacity than the radial roughness.

• The relative percentages R_{Wc} and R_{WM} are high when $N = 0.1, h_1^* = 2.4$ and $N = 0.1, h_1^* = 2$

Nomenclature

V=squeezing velocity
μ=Newtonian viscosity coefficient
γ=viscosity coefficient of micropolar fluids
δ=thickness of porous layer
χ=spin viscosity
h_1=greatest film thickness
h_2=least possible film thickness
K=step size
L=characteristic length
M=Hartmann number
N=coupling number
P=dimensionless pressure

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P_1=dimensionless pressure in the region $0 \leqslant r \leqslant K$
P_2=dimensionless pressure in the region $K \leqslant r \leqslant 1$
W=Non-dimensional load sustaining capacity
w=load sustaining capacity

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