Combined effect of piezo-viscous dependency and Couple Stresses on Squeeze-Film Characteristics of Rough Annular Plates

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Abstract: In this paper the combined effect of piezo viscous dependency and couple stresses on the performance of squeeze film characteristic of the Rough annular plates has been studied. The modified Reynolds equations are obtained for the two roughness structure in one-dimensional surface, namely radial roughness pattern and Azimuthal roughness pattern. The results has been presented numerically. It is observed that the effect of radial (azimuthal) roughness pattern on the bearing surface decrease (increase) the pressure, load carrying capacity, and approach of squeeze film time.

1. Introduction

In Modern studies surface roughness with couple stress have remarkable attribute in the research field and mechanical engineering. Lin et al.[1-2] studies the concept of stoke fluid model of couple stress and analyses that the couple stress variation effects the characteristic of squeeze film between two plates which are spherical and flat, also noticed that the encircle structure present in rough surface increases the mean of load carrying capacity and elongate the approach of squeeze film time compared to smooth surface. Syeda Tasneem Fathima el al.[3-5] discusses the derivation of Modified MHD Stochastic Reynolds equations with CCFS on Rough Porous Rectangular plates and Elliptical plates, In the non-Newtonian fluids the use of lubricant conducting electricity generates the transverse magnetic field due to which the squeeze film bearings, load capacity and approach of squeeze film time became better compared to NCNF. Sujatha el at.[6] and Ayyappa el at.[7] analyses on the same topic effect of variation in viscosity with Couple stress fluids in porous parallel, rectangular plates and short journal bearings. According to their observation the effect of viscosity variation reduces the velocity approach , load carrying capacity, pressure and squeeze film time compared with constant viscosity. Birdar Kashinath and Hanumagowda [8-10] studied the effect of MHD on porous wide composite slider bearing which is lubricated with couple stress fluids. It is observed that when the lubricated couple stress and magnetic field strength increases the parameter like load carrying capacity, coefficient of friction, fluid film pressure, frictional force increases in composite slider bearing where as in平面 slider bearing the steady state , dynamic stiffness and characteristics of damping improves. Naduvinamani [11] analysis that in circular stepped plates the effects of lubricated squeeze film and applied magnetic field with couple stress fluids increases the load carrying capacity and reduces approaching time when compared to non-magnetic case. Syeda Arishya Naseem fathima[12] observe that the effect of couple stresses and applied magnetic field externally increases
the values of carrying load capacity and decreases squeeze film time in convex curved plates. Syeda 
Tasneem Fathima el at.[13-14] investigates when magnetic field is applied in the rough surface it plays
a important role in improving the characteristics of bearings in circular plates and plane slider
bearings. Naduvinamani[15-16] studies the concept of parallel circular plates and stepped bearings on
the rough surface which is lubricated with couple stress and magnetic field, and its effects are shown
in two different operating parameter one is azimuthal(radial) is to increase (decrease) load carrying
capacity and approach time of squeeze film in circular stepped plates and lubricated couple stress and
viscosity pressure dependency increases the squeeze film time and load carrying capacity in parallel
circular plates. Work done by Naduvinamani el at.[17-18], it is analysed that magneto hydrodynamic
and lubricated couple stress roughness with squeeze film effects are same in both rectangular porous
plates and stepped circular plates. In both the cases the load carrying capacity and time of approach is
enhance in rectangular plates and squeeze film time reduces in circular stepped plates. Neminath
Bujappa Naduvinami el a. and Hanumagowda el at.[19-20] studies that in rough parallel plates and
circular stepped plates the effect of pressure dependent viscosity reduces the parameter values of load
carrying capacity and squeeze film time, where as in sphere and rough flat plate of circular stepped
plates the effect of lubricated couple stress and pressure dependent viscosity is applied due to which
the characteristic of squeeze film extents the load carrying capacity and approaching time is similar to
iso-viscous lubricants cases. Present work is discussed to study the effect of piezo-viscous dependency
and Couple Stresses on Squeeze-Film Characteristics of Rough Annular Plates.

2. Mathematical Formulation

The Geometric representation of the analysed problem is shown above in the Figure 1. When the two
Rough Annular plates are placed in series with one another having velocity \( V (=dh/dt). \)

Where \( p \) is the pressure between the two annular plates, \( H \) is the thickness of the film between the
plates, \( t \) is the squeeze film time, \( \mu \) is the viscosity of lubricant, \( u \) is the velocity components, \( \eta \) is the
responsible constant material for couple stresses. A magnetic field which is uniformly transverse is
introduced between the two annular plates bearing in the direction of Z-axes.

**Figure- 1:** Symmetric diagram of Annular Rough Plates

The basic equation for motion are given by

\[
\frac{1}{r} \frac{\partial}{\partial r} (ru) + \frac{\partial v}{\partial y} = 0
\]

(1)

\[
\mu \frac{\partial^2 u}{\partial y^2} - \eta \frac{\partial^4 u}{\partial y^4} = \frac{\partial p}{\partial r}
\]

(2)

\[
\frac{\partial p}{\partial y} = 0
\]

(3)

The relevant boundary conditions for velocity components are

At the upper surface \( y = h \):

\[
u = 0, \quad \frac{\partial^3 u}{\partial y^3} = 0,
\]

(4a)
\[ v = -\frac{\partial H}{\partial t} \]  
\[ \text{(4b)} \]

At the lower surface \( y=0; \)
\[ u = 0, \quad \frac{\partial^2 u}{\partial y^2} = 0, \]  
\[ \text{(5a)} \]
\[ v = 0 \]  
\[ \text{(5b)} \]

The Solution of the equation (1), using boundary conditions (4a) and (5a) the equation is
\[ u = -\frac{1}{2\mu} \frac{\partial p}{\partial r} \left[ y^2 - Hy + 2l^2 \left( 1 - \frac{\cosh\left( \frac{2y - H}{2l} \right)}{\cosh\left( \frac{H}{2l} \right)} \right) \right] \]  
\[ \text{(6)} \]

Where the film thickness is \( h, \) \( l = \left( \eta/\mu \right)^{2/3} \) is the parameter of couple stress, \( \mu \) the lubricant viscosity value remains constant, \( a \) is the co-efficient of pressure dependency of viscosity, the relationship between the viscosity-pressure dependency is analysed by Barus and co-workers is given by \( \mu = \mu_0 e^{a\rho} \)

The rate of flow of lubricant volume is given by
\[ Q = 2\pi \int_0^h u dy \]  
\[ \text{(7)} \]

Substituting the value of \( u \) in equation (7) from equation (6), the flux volume becomes
\[ Q = \frac{\pi r}{6\mu_0} \frac{\partial p}{\partial r} f(H,l,\alpha,\rho) \]  
\[ \text{(8)} \]

Substituting the expression of \( u \) from equation (6) in the continuity equation (1) and integrating the film thickness using the boundary conditions 4(b) and 5(b), the modified Reynolds equation is obtained
\[ \frac{1}{r} \frac{\partial}{\partial r} \left[ f(H,l,\alpha,\rho) r \frac{\partial p}{\partial r} \right] = 12\mu_0 \frac{dH}{dt} \]  
\[ \text{(9)} \]

Where
\[ f(H,l,\alpha,\rho) = H^3 e^{-\alpha\rho} - 12l^2 H e^{-2\alpha\rho} + 24l^3 e^{-2.5\alpha\rho} \tan\left( \frac{He^{\alpha\rho/2}}{2l} \right) \]  
\[ \text{(10)} \]

To model mathematically the roughness in the surface, the resistance to flow fluid film is reasoned in two parts
\[ H_s = h_s + h_s(r,\theta,\xi) \]  
\[ \text{(11)} \]

Let \( f(h_s) \) be the density probability function, \( h_s \) be the thickness of the stochastic film. Using the average thickness of stochastic film of modified Reynolds equations (8) and (9) compared to \( f(h_s) \).
\[ \frac{1}{r} \frac{\partial}{\partial r} \left\{ E \left\{ f(H,l,\alpha,p) \right\} r \frac{\partial E(p)}{\partial r} \right\} = 12\mu_0 \frac{\partial H}{\partial t} \]  

(12)

Where 
\[ E(*) = \int_{-\infty}^{\infty} f(h) \, dh \]  

(13)

For most of the lubricating surfaces, the Gaussian distribution for describing the roughness profile heights is valid up to at least three standard deviations, the roughness distribution function is assumed in the form

\[ f(h_r) = \begin{cases} \frac{35}{32\sigma^2} (c^2 - h_r^2)^3, & -c < h_r < c \\ 0 & \text{ elsewhere} \end{cases} \]  

(14)

Where the standard deviation \( c = \overline{\sigma} \) and \( c = 3\overline{\sigma} \).

In hydrodynamics lubrication of surface roughness is compare to the theory of Christensen’s Stochastic that there are two different pattern of roughness in one dimensional space namely

2.1 Radial Roughness Pattern

The one dimensional radial roughness pattern has the form of long, narrow ridges and valleys running in the radial direction (i.e. they are straight ridges and valley passing through \( z = 0, \ r = 0 \) to form star pattern), in this case the film thickness takes the form

\[ H = h + h_r(y, \xi) \]  

(15)

and the average modified Reynolds equation (9) takes the form

\[ \frac{1}{r} \frac{\partial}{\partial r} \left\{ E \left\{ f(H,l,\alpha,p) \right\} r \frac{\partial E(p)}{\partial r} \right\} = 12\mu_0 \frac{dH}{dt} \]  

(16)

2.2 Azimuthal Roughness Pattern

The Roughness pattern in one dimensional radial surface azimuthal roughness pattern on the bearing surface has the roughness structure in the form of long narrow ridges and valleys running in \( y \) - direction (i.e. they are circular ridges and valleys on the flat plate that are concentric on \( z = 0, \ r = 0 \)). In this case the film thickness assumes the form

\[ H = h + h_r(r, \xi) \]  

(17)

and the averaged modified Reynolds equation (9) takes the form

\[ \frac{1}{r} \frac{\partial}{\partial r} \left\{ E \left\{ f(H,l,\alpha,p) \right\} r \frac{\partial E(p)}{\partial r} \right\} = 12\mu_0 \frac{dH}{dt} \]  

(18)

Equations (16) and (18) together can be written as

\[ \frac{1}{r} \frac{\partial}{\partial r} \left\{ \frac{1}{g_r(H,l,\alpha,p)} g_r(H,l,\alpha,p) r \frac{\partial E(p)}{\partial r} \right\} = 12\mu_0 \frac{dH}{dt} \]  

(19)

Where

\[ g_r(H,l,\alpha,p) = \begin{cases} E \{ f(H,l,\alpha,p) \} & \text{ for radial roughness} \\ \left[ E \{ 1 / f(H,l,\alpha,p) \} \right]^{-1} & \text{ for azimuthal roughness} \end{cases} \]
\[ f_i(H_i, l, \alpha, p) = H_i^3 e^{-\alpha p} - 12l^2 H_i e^{-2\alpha p} + 24l^3 e^{-2.5\alpha p} \tanh(H_i e^{2p/2} / 2l) \]

Using non dimensional quantities

\[ l^* = \frac{2l}{h_0}, \quad r^* = \frac{r}{a}, \quad H_i^* = \frac{H_i}{h_0}, \quad C = \frac{c}{h_0}, \quad h^* = \frac{h}{h_0}, \quad h_i^* = \frac{h_i}{h_0}, \]

\[ p^* = \frac{E(p)h_0^3}{\mu_0 a^2 (-dH_i/dt)}, \quad G = \frac{\alpha_\mu R^2}{h_0^3}, \quad Q^* = \frac{Q \mu_0 R^2}{h_0^2} \]

the non dimensional Reynold equation and Volume flow rate is given by

\[ \frac{\partial}{\partial r^*} \left[ g_i^* \left( H_i^*, l^*, G, p^* \right) r^* \frac{\partial p^*}{\partial r^*} \right] = -12r^* \]

(20)

\[ Q^* = \frac{\pi r^*}{6} \frac{\partial P^*}{\partial r^*} g_i^* \left( H_i^*, l^*, G, p^* \right) \]

(21)

where

\[ g_i(H_i^*, l^*, C) = \begin{cases} E \left\{ f_i(H_i^*, l^*) \right\} & \text{for radial roughness} \\ \left[ E \left\{ 1 / f_i(H_i^*, l^*) \right\} \right]^{-1} & \text{for azimutal roughness} \end{cases} \]

and

\[ f_i(H_i^*, l^*) = -H_i^{*3} + 6l^2 H_i^{*1} \left( 4 + \sech^2(H_i^*/2l^*) \right) - 60l^3 \tanh(H_i^*/2l^*) \]

(22)

The boundary conditions for pressure are

\[ p^* = 0 \text{ at } r^* = a \]

\[ p^* = 0 \text{ at } r^* = 1 \]

(23)

The Non-dimensional pressure is given as

\[
 p^* = \frac{3(a^2 - 1)}{g_0(H^*, l^*, C)} \log r^* \left[ \log a^* \left( \frac{r^2 - 1}{(a^2 - 1)} \right) \right] \\
+ G \frac{9(a^2 - 1)g_i(H^*, l^*, C)}{2g_0(H^*, l^*, C)} \left[ \left( \frac{\log r^*}{\log a^*} \right) - \frac{\left( r^2 - 1 \right)}{(a^2 - 1)} \right] \]

(24)

The load capacity of the squeeze film is obtained by integrating the pressure under the boundary conditions for area of the plate

\[ W^* = 2\pi \int_a^1 Pr^* \, dr^* \]

(25)

The non-dimensional capacity of carrying load is given as:

\[
 W^* = \frac{3\pi(a^2 - 1)^2}{2g_i(H^*, l^*, C)} \left[ \frac{1}{\log a^*} \left( \frac{a^2 + 1}{a^2 - 1} \right) \right]^{+} \\
+ G \frac{9\pi(a^2 - 1)^2 g_i(H^*, l^*, C)}{g_0(H^*, l^*, C)} \left[ \frac{\left( 13a^2 - 7 \right) \log a^* + 4(a^2 - 1)}{16 \left( \log a^* \right)^2} \right. \\
- \frac{a^4 - 2a^2 - 2}{12(a^2 - 1)} \right] 
\]

(26)
When the non-dimensional capacity of carrying load ($W^*$) remains same, the thickness of the squeeze film decreases with respect to time, then the value of non-dimensional time ($t^*$) can be obtained by integrating

$$
t^* = \int \left[ \frac{3\pi(a^2-1)^2}{2g_0(H^*,l^*,C)} \left\{ \frac{1}{\log a^*} - \frac{(a^2+1)}{a^2-1} \right\} + \frac{9\pi(a^2-1)^2}{g_0(H^*,l^*,C)} \left( \frac{13a^2-7}{16(\log a^*)^2} - \frac{a^4-2a^2-2}{12(a^2-1)} \right) \right] \, dh^* \tag{27}$$

### 3. Results and Discussions

Analysing this paper is done by surface roughness of annular plates with combined effect of piezo-viscous dependency and non-Newtonian couple stress with respect to the different non-dimensional parameters like, parameter $l^*$ for couple stresses, parameter $C$ for roughness, where $l = \sqrt{\frac{\eta}{\mu}}$ is the parameter of the lubricant due to the pressure of polar additives. Hence geometry study of the mechanism of fluid interaction provided with the bearing is analysed. To calculate the non-dimensional quantities such as $P^*, W^*, t^*$ where $h^* = 0.6$, and $a^* = 0.02$.

#### 3.1 Non-dimensional pressure.

Variation in non-dimensional $P^*$ with $r^*$ for different values of $C$ when $h^* = 0.6, l^* = 0.3, G = 0.04, a^* = 0.3$ is seen in figure 2. It is determined that squeeze film pressure $P^*$ significantly increases with the increase value of $C$, the effect is significantly more in annular rough plates than smooth plates. Variation in non-dimensional $P^*$ with $r^*$ for different values of $l^*$ when $h^* = 0.6, C = 0.3, G = 0.04, a^* = 0.3$ is seen in figure 3. It is determined that squeeze film pressure $P^*$ significantly increases with the increase value of $l^*$, the effect is significantly more in annular rough plates than smooth plates. In Figure 4 it is viewed that variation in non-dimensional $P^*$ with $r^*$ for different values of $G$ when $h^* = 0.6, l^* = 0.3, C = 0.3, a^* = 0.2$. Overall the non-dimensional squeeze film pressure increases with the different values of $C$, $l^*$ and $G$ in the annular Rough plates is significantly more in Radial pattern compared to azimuthal pattern.

#### 3.2 Load supporting capacity.

From Figure 5, it is clearly understood that the variation of non-dimensional load capacity $W^*$ with $h^*$ changes when there is a variation in the value of $C$ and $h^* = 0.6, l^* = 0.3, G = 0.04, a^* = 0.2$ there is significantly increases in the non-dimensional load carrying capacity $W^*$ when $l^*$ is increased in Figure 6, the change in value of $l^*$ enhance the change of non-dimensional load carrying capacity $W^*$ for rough Annular Rough plates. Figure 7 shows the variation of $W^*$ change with respect to $G$. Overall the non-dimensional load carrying capacity increases with the different values of $C$, $l^*$ and $G$ in the annular Rough plates is significantly more in Radial pattern compared to azimuthal pattern the value of $h^*$ decreases simultaneously increases the value of $W^*$.

#### 3.3 Squeeze film time.

In Figure 8, Variation in non-dimensional squeeze time $t^*$ with $h^*$ for different values of $C$ when $h^* = 0.6, l^* = 0.3, G = 0.04, a^* = 0.2$ is the constant value. The squeeze time $t^*$ increases when the values
of $l^*$ are varies the non-dimensional $h^*$, $t^*$ increases when $l^*$ decreases in Figure 9. In Figure 10 the variation of non-dimensional squeeze film time $t^*$ changes when $G^*$ changes, overall the squeeze film $t^*$ increases when $h^*$ values increases in Annular rough plates in radial pattern compared to azimuthal pattern.

![Figure 2: Variation in non-dimensional pressure $p^*$ with $r^*$ for different values of $C$](image1)

$h^*=0.6, P^*=0.3, G=0.04, a^*=0.3$

![Figure 3: Variation in non-dimensional pressure $p^*$ with $r^*$ for different values of $P^*$](image2)

$h^*=0.6, C=0.3, G=0.04, a^*=0.3$

![Figure 4: Variation in non-dimensional pressure $p^*$ with $r^*$ for different values of $G$](image3)

$h^*=0.6, P^*=0.3, C=0.3, a^*=0.2$

![Figure 5: Variation in non-dimensional load carrying capacity $W^*$ with $h^*$ for different values of $C$](image4)

$h^*=0.6, P^*=0.3, C=0.3, a^*=0.3$

![Figure 6: Variation in non-dimensional load carrying capacity $W^*$ with $h^*$ for different values of $P^*$](image5)

$h^*=0.6, C=0.3, G=0.04, a^*=0.2$

![Figure 7: Variation in non-dimensional load carrying capacity $W^*$ with $h^*$ for different values of $G$](image6)

$h^*=0.6, C=0.3, P^*=0.3, a^*=0.2$
4. Concluding remarks.

The squeeze film characteristics between the piezo-viscous dependency and non-Newtonian couple stresses in Annular Rough Plates influence by surface roughness due to the presences of NNCS and PVD and magnetic field is observed.

- For the Rough Annular plates, the load carrying capacity $W^*$ increases when the values of $l^*$ couple stresses parameter increases, it is noticed that the non-dimensional squeeze film pressure enhanced significantly when C values are increased.

- Further significantly observed that the squeeze film time changes when the variation of $l^*$ values are changed.

- It is observed and analysed that this idea is used in the field of engineering to design the bearing with the squeeze film characteristics in rough annular surface bodies with combined effects of peizo-viscous dependency and non-Newtonian couple stresses with transverse magnetic field with Annular Rough Plates.
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Page 2:
In the Acknowledgements section, the following figure 1 appears:

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Figure- 1: Symmetric diagram of Annular Rough Plates”

This should read:

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