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The Effect of Magnetic Field and Fluid Inertia on the Film Pressure between Two Axially Oscillating Parallel Circular Plates with a Second Order Fluid as Lubricant

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Abstract

The present theoretical study investigates the effects of magnetic field and fluid inertia on the squeeze film between two parallel plates of which the top plate is axially oscillating. The lubricant considered in the present study is a conducting visco-elastic second order liquid, representing polymeric additives. The soul aim is to study the efficiency of the non-Newtonian fluid lubricant in the presence of magnetic field in enhancing the load bearing capacity and building up positive pressure distribution. Flow phenomena are characterized by non-dimensional parameters such as Reynolds number (R_e), elastic parameter (S_2^*), cross-viscosity parameter (S_3^*) and magnetic Parameter i.e. Hartmann number (M).

It is remarked that the visco-elastic lubricants in the presence of magnetic field enhance the efficiency of axially oscillating parallel circular plate type bearing but the fluid possessing higher cross viscosity under the dominance of inertia force ($R_e > 1$) is unsuitable for the purpose.

Keywords: Circular plates, visco-elastic fluid, fluid inertia, pressure, magnetic field.

Glossary

Ferrofluid: A ferrofluid is a stable colloidal suspension of sub-domain magnetic particles in a liquid carrier. The particles, which have an average size of about $100A^0$ (10nm), are coated with stabilizing dispersing agent (surfactant) which prevents particle agglomeration even a when a strong magnetic field gradient is applied to the ferrofluid.

Bearing: A bearing is a system of machine element whose function is to support an applied load by reducing friction between the relatively moving surfaces.

Squeeze film bearing: It is one type of fluid-film bearing which can support a load because of oscillatory relative normal motion.

1. Introduction

The effects of fluid inertia between axially oscillating parallel circular plates were studied both experimentally and theoretically by Kuhn and Yates [1]. Tichy and Winer [2] studied the inertial consideration in parallel circular squeeze film bearings using regular perturbation technique. Elkough [3] accounted for all the inertia terms in the equation of motion for a laminar non-Newtonian squeeze film. Ramanaiah [4] analyzed the problem of squeeze film considering the inertia effects with power law fluid as lubricant. Naduvinamani et al. [5] theoretically analyzed the problem of magneto-hydrodynamic couple stressed squeeze film lubrication between rough circular stepped plates and they found that azimuthal roughness pattern increased the mean load carrying capacity and squeeze film time. Naduvinamani et al. [6] have undertaken a detailed study of magnetic effects on rectangular plates serving as bearing surfaces and reported that bearing characteristics such as pressure distribution, load capacity and squeezing time enhance for increasing the Hartmann number. Patel et al. [7] considered magnetic fluid based squeeze film between porous rotating rough circular plates.

Karadere [8] obtained a pressure distribution in a thrust bearing by using Reynolds equation in case of stable lubricant viscosity and isothermal conditions. Singh and Gupta [9] presented a theoretical investigation concerning the effect of ferrofluid lubrication on the dynamic characteristic of curved slider bearing based on Shliomis model for magnetic fluid flow and observed that the effect of rotation of magnetic particles improved the stiffness and damping capacities of the bearing. Kudenatti et al. [10] have presented numerical solution of the MHD Reynolds equation for squeeze film lubrication between porous and rough rectangular plates. They have applied finite difference based multigrid method for the solution of modified Reynolds equation to investigate the combined effects of surface roughness, magnetic field, couple stress fluid and permeability.

The effects of electric and magnetic fields on the flow of electrically conducting lubricants have been studied extensively and the studies reveal that MHD bearings have several theoretical advantages over conventional bearings. The most common type MHD bearing is the slider bearing and the two general configurations of the slider are of interest. One configuration uses a transverse magnetic field with a tangential electric field, while the other, uses a tangential magnetic field with a tangential electric field. Das [11] discussed the optimum load bearing capacity for slider bearings lubricated with coupled stress fluids in magnetic field. Wei et al. [12] studied the ferrofluid lubrication with an external magnetic field. Zahn and Rosenweigh[13] described the motion of magnetic fluids through porous media under the influence of obliquely applied magnetic field and established that the magnetization induced a positive effect on the performance of the bearing system. The load carrying capacity was found to be increased.

Bhat and Deheri [14] investigated the squeeze film behaviour between porous annular disks using a magnetic fluid lubricant with the external magnetic field oblique to the lower disk and concluded that the application of magnetic fluid as a lubricant enhanced the performance of the squeeze film bearing system. Shan and Bhat [15] considered the squeeze film based on magnetic fluid in the curved porous circular plates taking a magnetic fluid lubricant in the presence of an external magnetic field oblique to the lower plate. Hamza [16] studied the motion of an electrically conducting fluid film squeezed between two parallel disks in the presence of a magnetic field applied perpendicular to the disks. Here a regular perturbation scheme was used for the analytic solutions and it was shown that the electromagnetic forces increase the load carrying capacity. Bujurke and Kudenatti [17] studied the effect of surface roughness on the squeeze film behaviour between two rectangular plates with an electrically conducting fluid in the presence of transverse magnetic field and observed that the roughness and magnetic field provided a significant load carrying capacity and ensured a delayed squeezing time compared to classical case. Hsu et al. [18] studied the squeeze film characteristics between rotating circular disc with an electrically conducting lubricant in the presence of a transverse magnetic field and established that the squeeze film characteristics of a rotating circular disc were improved. Lin et al. [19] studied the effect of convective fluid inertia forces on magnetic fluid in conical squeeze film in the presence of external magnetic field considering Shliomis model. In this case the squeeze film performance improved with larger value of the inertial parameter of fluid inertia forces, volume concentration of ferrite particles and the strength of the applied magnetic field. Wannatong et al. [20] have presented a review on simulation of piston secondary motion and pressure distribution in lubrication film. Further, Asbik et al. [21] worked on coupled boundary layer for laminar film condensation of downward flowingsteam-air mixture onto a single horizontal tube. Dash and Kamila [22] have studied the effect of fluid inertia on the film pressure between two axially oscillating parallel circular plates with a second order fluid as lubricant but they have restricted their discussion to non-conducting lubricant without the presence of magnetic field.

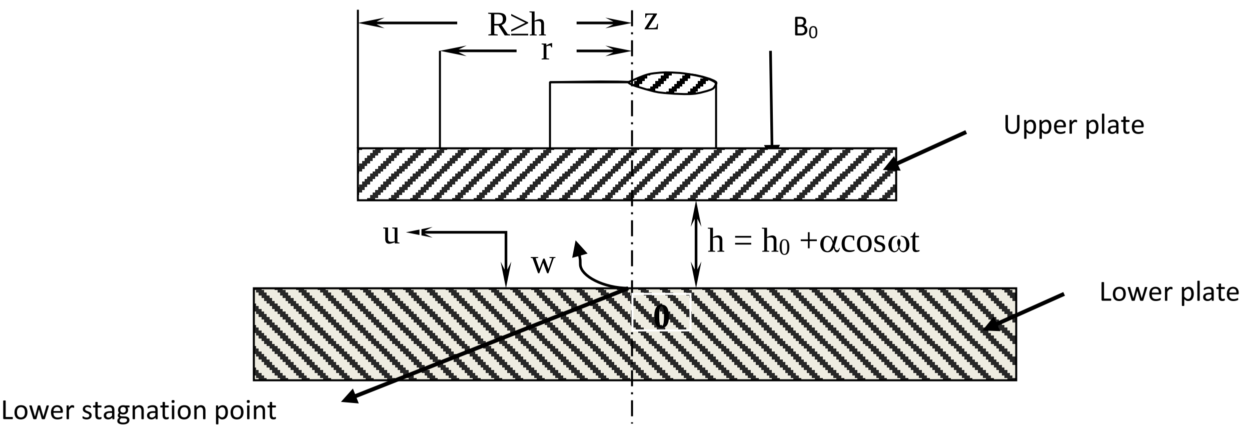

Figure1. The geometry of bearing

However, most of the above referred studies are related to Newtonian lubricants. But in the present study we have considered the non-Newtonian second order fluid as lubricant.

Moreover, we have applied an external trabsverse magnetic field which interacts with electrically conducting property of the lubricant producing a pondermotive force of electromagnetic origin. Further, the fluid model considered here represents a real fluid experimentally found by Oldroyd et al. [23]. The mixture of polymethyl Methacrylate in pyridine at 25^{o}C containing 30.5 gm of polymer per litre and having density 0.98 gm/ml fits well in the above model.

2. Basic equations

The equation of continuity for an incompressible fluid is given by

$$V_{i,i} = 0 \tag{1}$$

The equation of motion is

$$\rho\left(\frac{\partial v^i}{\partial t} + v^j v^i{}_{,j}\right) = \rho X_i - P_{,i} + P^{ij}{}_{,j} + \bar{J} \times \bar{B} \tag{2}$$

Where v^i is the velocity vector, X_i is the external body force acting on the fluid element per unit mass in the ith direction, P is the mean pressure, P^{ij} is the stress in the fluid, $\bar{J}$ is the current density and $\bar{B}$ is the magnetic induction vector.

The constitutive equation for second order fluid of Coleman and Noll [24] can be derived as

$$P^{ij} = -P^I + \mu_1 A_1 + \mu_2 A_2 + \mu_3 A_1^2 \tag{3}$$

Where μ_1, μ_2 and μ_3 are material constants. It is customary to call μ_1 the coefficient of ordinary viscosity, μ_2 the coefficient of visco-elasticity, μ_3 the coefficient of cross-viscosity. It is important to point out that such a fluid exhibits normal stress effects in shear flows and equation (3) is valid for low shear rates, μ_2 being negative from thermodynamic considerations, A_1 and A_2 are defined by

$$A_1 = A_{(1)ij} = v_{i,j} + v_{j,i},$$

$$A_2 = A_{(2)ij} = a_{j,i} + a_{i,j} + 2v_{m,i} \; v_{m,j},$$

Where v_i is the velocity vector and a_i, the acceleration vector given by $a_i = \frac{\partial v_i}{\partial t} + v_m \, v_{i,m}$.

3. Analysis of the Problem

We consider a second order incompressible fluid between two parallel circular plates initially separated by a small distance h_0. There is a sinusoidal axial oscillation of the top plate with amplitude α. Cylindrical polar co-ordinates are used to describe the flow phenomena. The velocity components in the radial r and in the axial z directions are denoted as u and w respectively. The geometry of the problem is shown in Fig.1 and the velocity field defined as

$$u = u(r,z,t), \;\; w = w(z,t) \tag{4}$$

The surviving stress components from equation (3) in cylindrical polar co-ordinates are

$$P^{rr} = 2\mu_1 \frac{\partial u}{\partial r} + 2\mu_2\left[\frac{\partial}{\partial r}\left(u\frac{\partial u}{\partial r} + w\frac{\partial u}{\partial z}\right) + \left(\frac{\partial u}{\partial r}\right)^2 + \frac{\partial^2 u}{\partial r \partial t}\right] + \mu_3\left[4\left(\frac{\partial u}{\partial r}\right)^2 + \left(\frac{\partial u}{\partial z}\right)^2\right] \tag{5}$$

$$P^{\theta\theta} = 2\mu_1 \frac{u}{r} + 2\mu_2\left[\frac{\partial}{\partial r}\left(u\frac{\partial u}{\partial r} + w\frac{\partial u}{\partial z}\right) + \frac{u^2}{r^2} + \frac{1}{r}\frac{\partial u}{\partial t}\right] + 4\mu_3\frac{u^2}{r^2} \tag{6}$$

$$P^{zz} = 2\mu_1 \frac{\partial w}{\partial z} + 2\mu_2\left[\frac{\partial}{\partial z}\left(w\frac{\partial w}{\partial z}\right) + \left(\frac{\partial u}{\partial z}\right)^2 + \left(\frac{\partial w}{\partial z}\right)^2 + \frac{\partial^2 w}{\partial z \partial t}\right] + \mu_3\left[\left(\frac{\partial u}{\partial z}\right)^2 + 4\left(\frac{\partial w}{\partial z}\right)^2\right] \tag{7}$$

$$P^{zr} = \mu_1\left(\frac{\partial u}{\partial z}\right) + \mu_2\left[\frac{\partial}{\partial z}\left(u\frac{\partial u}{\partial r} + w\frac{\partial u}{\partial z}\right) + \frac{\partial}{\partial r}\left(w\frac{\partial w}{\partial z}\right) + 2\frac{\partial u}{\partial r}\frac{\partial u}{\partial z} + \frac{\partial^2 u}{\partial z \partial t}\right] + \mu_3\left[2\frac{\partial u}{\partial z}\left(\frac{\partial u}{\partial r} + \frac{\partial w}{\partial z}\right)\right] \tag{8}$$

The equations of momentum and continuity take the forms

$$\rho\left(\frac{\partial u}{\partial t} + u\frac{\partial u}{\partial r} + w\frac{\partial u}{\partial z}\right) = -\frac{\partial P}{\partial r} + \frac{\partial P^{rr}}{\partial r} + \frac{\partial P^{rz}}{\partial z} + \frac{P^{rr} - P^{\theta\theta}}{r} \tag{9}$$

$$\rho\left(\frac{\partial w}{\partial t} + w\frac{\partial w}{\partial z}\right) = -\frac{\partial P}{\partial z} + \frac{\partial P^{zz}}{\partial z} + \frac{\partial P^{rz}}{\partial r} + \frac{P^{rz}}{r} \tag{10}$$

and $$\frac{\partial u}{\partial r} + \frac{\partial w}{\partial z} + \frac{u}{r} = 0 \tag{11}$$

The momentum and continuity equations under the trasverse magnetic field, with relatively thin film, neglecting the variation of axial velocity in the radial direction and the pressure gradient along Z-axis are given by

$$\frac{\partial u}{\partial t} + u\frac{\partial u}{\partial r} + w\frac{\partial u}{\partial z} = -\frac{1}{\rho}\frac{\partial P}{\partial r} + \nu_1\frac{\partial^2 u}{\partial z^2} + \frac{\mu_2}{\rho}\left[w\frac{\partial^3 u}{\partial z^3} + \frac{2}{r}\left(\frac{\partial u}{\partial z}\right)^2 + \frac{\partial^3 u}{\partial z^2 \partial t}\right]$$

$$+\frac{\mu_3}{\rho}\left[\frac{1}{r}\left(\frac{\partial u}{\partial z}\right)^2-\frac{2u}{r}\frac{\partial^2 u}{\partial z^2}\right]-\frac{\sigma B_0^2 u}{\rho} \tag{12}$$

$$\frac{\partial w}{\partial t}+w\frac{\partial w}{\partial z}=\nu_1\frac{\partial^2 w}{\partial z^2}+\frac{\mu_2}{\rho}\left[11\frac{\partial w}{\partial z}\frac{\partial^2 w}{\partial z^2}+w\frac{\partial^3 w}{\partial z^3}+r^2\frac{\partial^2 w}{\partial z^2}\frac{\partial^3 w}{\partial z^3}+\frac{\partial^3 w}{\partial z^2\partial t}\right]$$

$$+\frac{\mu_3}{\rho}\left[\frac{r^2}{2}\frac{\partial^2 w}{\partial z^2}\frac{\partial^3 w}{\partial z^3}+7\frac{\partial w}{\partial z}\frac{\partial^2 w}{\partial z^2}\right]-\sigma\frac{B_0^2 w}{\rho} \tag{13}$$

and $u=-\frac{r}{2}\frac{\partial w}{\partial z}$ (14)

where $\nu_1=\frac{\mu_1}{\rho}$, the kinematic coefficient of viscosity.

It is assumed that the magnetic Reynold's number is so small that the induced magnetic field can be neglected in comparison with the applied one. It is also assumed that no applied and polarization voltage exists. This then corresponds to the case when no energy is added or extracted from the fluid by the electric field.

4. Solution of the equations

An iteration technique has been used to solve equation (12). Following Kahlert [25], equation (12) can be written as

$$\frac{\partial^2 u}{\partial z^2}=\frac{1}{\mu_1}\frac{\partial P}{\partial r}+\frac{1}{\nu_1}G\left(r,z,t\right) \tag{15}$$

where $$G\left(r,z,t\right)=\frac{\partial u}{\partial t}+u\frac{\partial u}{\partial r}+w\frac{\partial u}{\partial z}-\frac{\mu_2}{\rho}\left[w\frac{\partial^3 u}{\partial z^3}+\frac{2}{r}\left(\frac{\partial u}{\partial z}\right)^2+\frac{\partial^3 u}{\partial z^2\partial t}\right]$$

$$-\frac{\mu_3}{\rho}\left[\frac{1}{r}\left(\frac{\partial u}{\partial z}\right)^2-\frac{2u}{r}\frac{\partial^2 u}{\partial z^2}\right]+\frac{\sigma B_0^2 u}{\rho} \tag{16}$$

The boundary conditions are:

$$\left.\begin{array}{l} u=0, w=0 \text{ at } z=0 \\ u=0, w=V \text{ at } z=h \end{array}\right\} \tag{17}$$

where V is the velocity of the top.

The first iterate solution u_1 and w_1 are obtained by putting G(r, z, t) = 0 in (15) and using the boundary conditions (17), we get

$$u_1=\frac{3Vr}{h^3}\left(z^2-hz\right) \tag{18}$$

and $w_1 = -\frac{V}{h^3}\left(2z^3 - 3hz^2\right)$ (19)

With these values of velocities, equation (16) becomes

$$G(r,z,t) = 3r\frac{\partial V}{\partial t}\left[\frac{z^2}{h^3} - \frac{z}{h^2} - \frac{2\mu_2}{\rho h^3}\right]$$

$$+3rV^2\left[\frac{2z^3}{h^5} - \frac{z^4}{h^6} - \frac{6\mu_2}{\rho}\left(\frac{4z^2}{h^6} + \frac{1}{h^4} - \frac{4z}{h^5}\right) - \frac{3\mu_3}{\rho h^4}\right] + \frac{3rV\sigma B_0^2}{\rho h^3}\left(z^2 - hz\right) \quad (20)$$

Substituting equation (20) into equation (15) and using the boundary conditions (17), the second iterate solution u_2 of the radial velocity is obtained as

$$u_2 = \frac{1}{2\mu_1}\frac{\partial P}{\partial r}\left(z^2 - hz\right) + \frac{3r}{\nu_1}\frac{\partial V}{\partial t}\left[\frac{z^4}{12h^3} - \frac{z^3}{6h^2} + \frac{z}{12} + \frac{\mu_2}{\rho}\left(\frac{z}{h^2} - \frac{z^2}{h^3}\right)\right]$$
$$+\frac{3rV^2}{\nu_1}\left[\frac{z^5}{10h^5} - \frac{z^6}{30h^6} - \frac{z}{15h} - \frac{6\mu_2}{\rho}\left(\frac{z^4}{3h^6} - \frac{2}{3}\frac{z^3}{h^5} + \frac{z^2}{2h^4} - \frac{z}{6h^3}\right) + \frac{3\mu_3}{\rho}\left(\frac{z}{2h^3} - \frac{z^2}{2h^4}\right)\right]$$
$$+\frac{rV\sigma B_0^2}{4\rho\nu_1}\left[\frac{z^4}{h^3} - \frac{2z^3}{h^2} + z\right] \quad (21)$$

If V_0 is the maximum velocity of the top plate then from equations (14) and (21), we get

$$\frac{\partial w_2}{\partial z} = -\frac{1}{\mu_1 r}\frac{\partial P}{\partial r}\left(z^2 - hz\right)$$
$$-\frac{6V_0^2}{\nu_1}\left[\frac{z^5}{10h^5} - \frac{z^6}{30h^6} - \frac{z}{15h} - \frac{6\mu_2}{\rho}\left(\frac{z^4}{3h^6} - \frac{2}{3}\frac{z^3}{h^5} + \frac{z^2}{2h^4} - \frac{z}{6h^3}\right) + \frac{3\mu_3}{\rho}\left(\frac{z}{2h^3} - \frac{z^2}{2h^4}\right)\right]$$
$$-\frac{V_0\sigma B_0^2}{2\rho\nu_1}\left[\frac{z^4}{h^3} - \frac{2z^3}{h^2} + z\right] \quad (22)$$

Integration of equation (22) with boundary conditions

w = 0 at z = 0 and $w = -V_0$ at z = h = h_0 gives an expression for the radial pressure gradient at any r at the time of maximum velocity of the top plate.

$$\frac{\partial P}{\partial r} = -\frac{6r\mu_1 V_0}{h_0^3} - \frac{36r\mu_1 V_0^2}{\nu_1 h_0^3}\left[\frac{3h_0}{140} - \frac{\mu_2}{10\rho h_0} - \frac{\mu_3}{4\rho h_0}\right] + \frac{3}{5}\frac{rV_0\sigma B_0^2}{h_0} \quad (23)$$

Integrating equation (23) by using the boundary condition p = 0 at r = R , where R is the radius of the upper plate, the pressure at the maximum velocity of the top plate is

$$P^* = \frac{3\mu_1 V_0}{h_0^3}\left(R^2 - r^2\right)\left[1 + \frac{9}{70}\frac{h_0 V_0}{\nu_1} - \frac{3}{5}\frac{\mu_2 V_0}{\mu_1 h_0} - \frac{3}{2}\frac{\mu_3 V_0}{\mu_1 h_0}\right] - \frac{3}{10}\frac{V_0\sigma B_0^2}{h_0}\left(R^2 - r^2\right) \quad (24)$$

The non-dimensional form of pressure is

$$P = 3R_e\left(1-\frac{r^2}{R^2}\right)\left[1+R_e\left(\frac{9}{70}+\frac{3}{5}S_2^*-\frac{3}{2}S_3^*\right)\right]-\frac{3}{10}R_eM^2\left(1-\frac{r^2}{R^2}\right) \tag{25}$$

Where $R_e = \dfrac{h_0V_0}{\nu_1}$, Reynolds number

$S_2^* = \dfrac{-\mu_2}{\rho h_0^2}$, Elastic parameter

$S_3^* = \dfrac{\mu_3}{\rho h_0^2}$, Cross-viscosity parameter

$M^2 = \dfrac{h_0^2\sigma B_0^2}{\mu_1}$, Magnetic field

and $P = \dfrac{\rho h_0^4 P^*}{\mu_1^2 R^2}$, non-dimensional pressure.

5. Results and discussion

The objective of the following discussion is to bring out the effects of fluid inertia on the film pressure between two axially oscillating parallel circular plates with a second order fluid as lubricant under the influence of magnetic field. The effects of Reynolds number (R_e), elastic parameter (S_2^*), Cross-viscosity parameter (S_3^*) and magnetic parameter (M) on the fluid pressure are discussed.

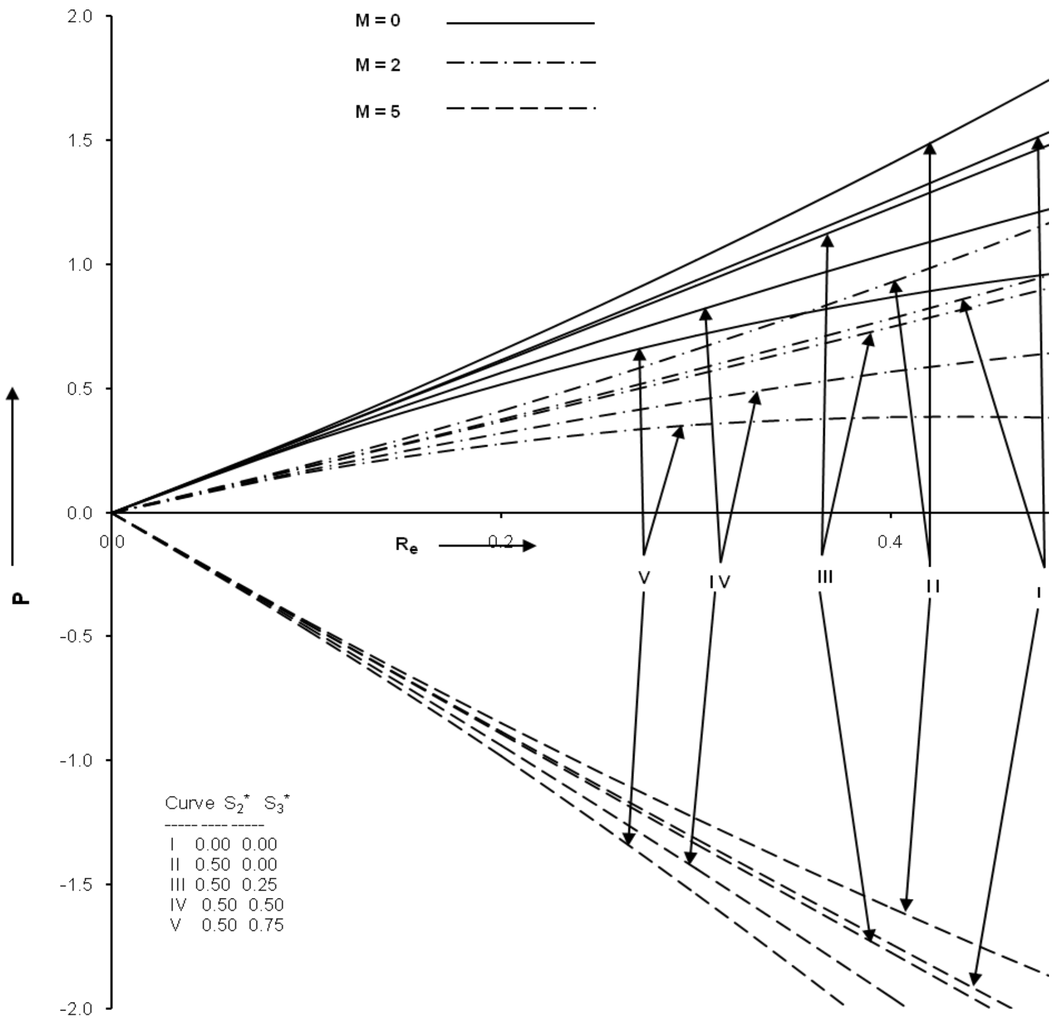

Fig-2.1 Pressure variation at the central point (r = 0) of bottom plate for different values of R_e, M and S_3^*

Fig.2.1 presents the graphical representation of pressure variation versus Reynolds number for different values of S_2^* and S_3^*. Reynolds number measures the ratio of inertia force and viscous force. In this figure, variation of pressure is studied under the dominance of viscous force since $R_e < 1$. An increase in R_e representing the case of greater inertia effect for variation of pressure in the bearing. It is evident that the positive pressure increases all most linearly as Reynolds number increases under the moderate magnetic field intensity but a negative pressure is built up as R_e increases which is a noteworthy observation in case of moderately high value of magnetic parameter. Thus, it is concluded that for building up positive pressure gradient which is desirable for enhancing load bearing capacity, magnetic field with moderately low intensity is to be applied to the non-Newtonian lubricant under study.

On careful observation it is seen that the most favourable condition for developing higher positive pressure is achieved in the absence of magnetic field (M =0) and cross-viscosity ($S_3^* = 0$) with non zero elasticity (Curve II, $S_2^* \neq 0$). Thus, it is concluded that elasticity property of the non-Newtonian lubricant is favourable for building up higher pressure and hence load bearing capacity. It is further pointed out that the observation relating to the case of without magnetic field, coincides with Dash and Kamila [22]. It is remarked that the pressure variation at the center increases with an increase in the values of Reynolds number (R_e<1) i.e. under the dominance of viscous force. This remark also coincides with the observation of SinhaRoy and Biswal [26].

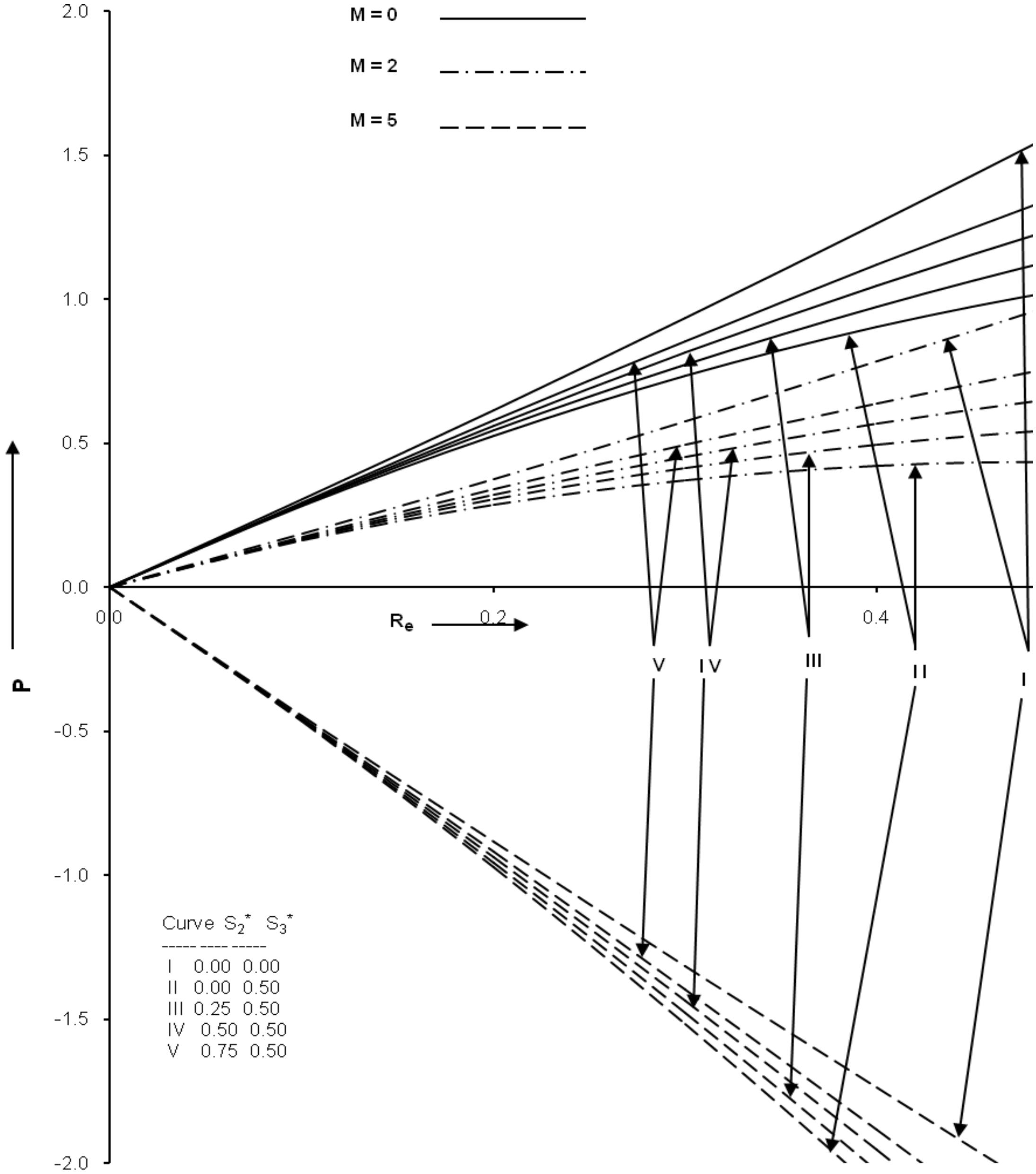

Fig-2.2 Pressure variation at the central point (r = 0) of bottom plate for different values of R_e , M and S_2^*

Fig. 2.2 exhibits the variation of pressure at the centre of the bearing. It is to note that Newtonian lubricant (Curve I, $S_2^* = 0, S_3^* = 0$), generates maximum pressure at the centre in the absence of magnetic field. But presence of cross-viscosity decreases the pressure distribution. The elastic property in conjunction with cross-viscosity increases the pressure at the centre. Further, it is observed that moderately high value of M generates negative pressure.

The presence of magnetic field decreases the pressure with an increase in the values of Reynolds number (R_e). It is quite interesting to note that for higher values of R_e, the pressure becomes negative even this holds for moderate values of magnetic number. But in case of slightly higher values of M i.e. for (M =5), the pressure remains negative for all values of R_e.

Table-1 presents the pressure variation for various values of S_2^*, S_3^*, R_e and M. One of the striking feature of the table-1 is that, it presents the case when R_e >1 i.e. the case of inertia force dominates over the viscous force which was not shown in the figures. It is evident that Reynolds number (R_e >1) is not favourable for building up higher pressure at the centre. It is also evident that elasticity of the lubricant (S_2^*=0.25, S_3^*=0.5, R_e =1, M= 0 and M = 2) builds up positive pressure which is desirable in designing the bearing system but cross viscosity associated with inertia dominance and higher magnetic intensity imbibes negative pressure which is not desirable.

Table-1: Pressure variation for different values of S_2^* and S_3^*

S_2^*	S_3^*	R_e	M		
			0	2	5
0	0	1.0	3.38572	2.18571	-4.11429
0.5	0	1.0	4.28572	3.08572	-3.21429
0.5	0.25	1.0	3.16072	1.96072	-4.33929
0.5	0.5	1.0	2.03571	0.83571	-5.46429
0.5	0.75	1.0	0.91071	-0.28929	-6,58929
0	0.5	2.0	1.13571	-0.06429	-6.36429
0.25	0.5	1.0	1.58571	0.38571	-5.91429
0.75	0.5	1.0	2.48572	1.28571	-5.01429

6. Conclusion

It is concluded that for moderately high values of magnetic field, the pressure reduces significantly and even becomes negative.

Therefore, the magnetic field with high intensity with moderate Reynolds number is not favourable for generating film pressure. Since higher pressure generation is a desirable quantity, it can be said that bearing with axially oscillating parallel circular plate type should not operate with a second order fluid as lubricant under the influence of moderately high magnetic field intensity.

Further it is observed that an increase in elastic parameter (S_2^*) leads to an increase in the pressure between the film which is desirable i.e., the presence of the elastic element favours the load bearing capacity where as an increase in cross-viscosity parameter (S_3^*) the pressure decreases. Thus, fluid possessing higher cross-viscosity is not suitable for the present bearing set up. Further, dominance of viscous force (R_e <1) favours in generating higher pressure.

Therefore, the visco-elastic lubricants enhance the efficiency of the bearing with axially oscillating parallel circular plate type. This should be a vital point for selection of the lubricant.

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