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# Inclined Slider Bearing with Magnetic Rheological fluid under the Effects of Second Order Rotation

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**Abstract**

*A slider bearing consisting of connected surfaces with magneto hydro dynamic fluid as lubricant is analyzed in the present study. A MR-model has been used as a non-Newtonian fluid in a slider bearing. A mathematical model of magneto hydro dynamic fluid flow in a slider bearing is conferred. An extended generalized Reynolds equation of motion for second order Rotatory theory of hydrodynamic lubrication is used for this study. Under the assumptions of the order of magnitudes of the variables, it can be seen only the viscous and non-Newtonian terms have effects, whereas the inertia terms are negligible. The pressure distribution in the bearing is calculated by neglecting higher order terms. The pressure is employed to analyze the bearing load carrying capacity. The results are conferred through table and graphs.*

*A generalized two dimensional Reynolds-type equation is derived using the equations of motion. The equation is solved to get the dimensionless pressure. The pressure is used to evaluate the bearing characteristics such as load carrying capacity. The results are presented graphically. The film pressure varies with density and viscosity hence it increases with the intensity of magnetic field. Thus the load carrying capacity of the bearing is enhanced by the application of the magnetic field. In presence of magnetic field there is increase in ratio $\alpha, \beta$ that leads to increased pressure as well as load carrying capacity in the bearing.*

*The differential equation is solved to induce the dimensionless pressure. The film pressure varies with Hartmann number, density and viscosity thus it will increase with the intensity of magnetic force. Therefore the load carrying capacity of the bearing is increased by the appliance of the magnetic force. In presence of magnetic force there's increase in aspects ratio α, β that results in increasing the pressure in addition as load carrying capacity within the bearing.*

**Keywords:** Continuity, Density, Film thickness, Reynolds equation, Rotation number, Taylor's number, Viscosity.

## 1. Introduction

In most mechanical systems wherever relative motion happens between two components, lubricants are introduced to minimize friction and wear. The geometry of the contacting parts determines the structure of the lubricating film. The researchers have thought of totally different configurations of the lubricating film within the clearance zone in their analysis. The contacting surfaces are narrowing geometrically in linear vogue as thought of by Ozalp. He utilized the repetitive transfer matrix approach to recommend optimum film profile parameters for reduced coefficient of friction (Ozalp et al. 2006). Das has given a comparative study of optimum load-carrying capacity for finite and infinite slider bearings has additionally been created. It's discovered that each the values of most load capacity and therefore the corresponding I/O film quantitative relation depend upon couple stress and magnetic parameters and therefore the form of bearings jointly (Das, 1998). Bayrakpeken et al. dispensed a comparative study of inclined and parabolic slider bearings employing a non-Newtonian fluid within the clearance zone of the slider bearings. He derived an expression for the performance characteristics of the bearings (Bayrakceken et al. 2006). Shah et al. studied a slider bearing with exponential film thickness profile and analyzed the expressions for variation of dimensionless pressure and coefficient of friction (Shah et al. 2003), (Siddangouda et al. 2014). Yurusoy obtained a perturbation answer for pressure distribution during a slider bearing with a Powel-Eyring fluid as lubricating substance (Yürüsoy, 2003). Bujurke et al. used a second grade fluid during a taper flat slider bearing almost like that employed by Ozalp and made a Von-korman momentum integral solution (Bujurke et al. 2007). Shah computed values for the bearing characteristics of a secant film slider bearing employing a magnetic fluid lubricating substance. Various kinds of fluids are utilized in the clearance zone of slider bearings and their performance investigated (Shah et al. 2003).

However, so as to boost lubricating performance, the increasing use of Newtonian stuff that has been homogenized with long chain polymers has been discovered. Since the standard micro-continuum theory cannot accurately describe the flow of those varieties of fluids, varied micro-continuum theories are projected (Lin et al. 2004). Stokes projected the best micro-continuum theory which allows the presence of couple stresses, body couples and non-symmetric tenors (Stokes, 1966), (Lin, 1997). Various researchers have investigated the impact of the couple stress fluid model on the steady state performance of slider bearing configurations victimization using different numerical methods. In recent times, most numerical hydrodynamic lubrication has concerned the utilization of the Reynolds equation and therefore the finite distinction technique. A finite distinction multigrid approach was wont to investigate the squeeze film behavior of poroelastic bearing with couple stress fluid as lubricant by Bujurke et al. They have investigated that poroelastic bearings with couple stress fluid as lubricant increased pressure distribution and ensured important load carrying capacity (Mitidierri, 2005).

Sarangi et al. solved the modified Reynolds equation extended to incorporate couple stress effects in lubricants blended with polar additives by using the finite distinction methodology with a consecutive over relaxation theme. They analyzed the increase in load carrying capacity and reduction in friction coefficient as compared to Newtonian lubricants. Lin used the conjugate methodology of iteration to make up the pressure generated in a finite journal bearing lubricated with a couple stress fluids. The results obtained together with increase within the load carrying capacity accept as true with those obtained by Sarangi and Bujurke et al (Sarangi et al. 2005). Elsharkawy provided a numerical solution for a mathematical model describing the hydrodynamic lubrication of misaligned journal bearings with couple stress fluids as lubricants by using the finite distinction methodology (Elsharkawy, 2004). Lin calculated the steady and discomposed pressure of a two dimensional plane inclined slider bearing incorporating a couple stress fluid using the conjugate gradient methodology and reportable improved steady and dynamic performance as compared to the Newtonian case particularly for higher aspect ratios (Lin, 2003).

Nada and Osman investigated the matter of finite hydrodynamic bearing lubricated by magnetic fluids with couple stresses using the finite distinction methodology. For various couple stress parameters and magnetic coefficients, they obtained the pressure distribution. They analyzed that fluids with couple stresses are higher as compared with the Newtonian case once comparison of the bearing static characteristics. The open literature is replete with slider bearing structure with couple stress fluids as lubricants using the finite distinction methodology because the numerical tool for analysis as are often deduced from the literature cited (Nada, 2007). Previous researchers appear to not have exploited the pertinence of finite part strategies in slider bearing structure. The finite part methodology is perhaps the foremost correct and versatile, however tends to be terribly time overwhelming and needs high data, not assessable to the common designer (Flores, 2006), therefore it's obvious absence within the study.

The MHD lubrication of the finite slider bearings was also analyzed by Lin (Lin, 2002). He has shown that the application of magnetic field signifies an influence in load carrying capacity, power loss and the friction parameters of slider bearing. Fathima et al. have observed the MHD Slider bearing lubricated with conducting couple stress fluid between two electrical conducting plates under the influence of magnetic field in the free space. They obtained a closed form of solution for the film pressure and the load carrying capacity analytically in the terms of the inlet-outlet ratio of slider bearings (Fathima et al. 2018).

The influence of the oscillatory shear on the magnetic flux dependent electrical conductivity of the magnetorheological fluid (MRF) was reported. On applying a 0.96 T magnetic flux, the electrical conductivity may increase regarding 1500 times larger than the one while not magnetic flux. By increasing the amount fraction of carbonyl iron particles within the MRF from five percent to thirty percent, the electrical conductivity inflated regarding 565 times (Ruan et al. 2017).

The rotation of fluid film about an axis that lies across the film gives some new results in lubrication problems that were derived by Banerjee et al. (Banerjee et al. 1982) in hydrodynamics. The origin of rotation can be traced by certain general theorems related to vorticity in the rotating hydrodynamics. The rotation induces a component of vorticity in the direction of rotation of fluid film and the effects arising from it are predominant, for large Taylor's Number, it results in the streamlines becoming confined to plane transverse to the direction of rotation of the film.

The "Extended Generalized Reynolds Equation", which takes into account of the effects of the uniform rotation about an axis that lies across the fluid film and depends on the rotation number $M$ i.e., the square root of the conventional Taylor's Number, given by Banerjee et al.

The most common form of lubricated slider bearing is Plane Inclined Pad. The geometry of plane-inclined pad is given by figure (1). Which shows that the gap $h$ decreases with increasing $y$, hence the runner has to move towards origin in $y$-direction. Its velocity is $U$. the minimum film thickness is $h_o$ and the minimum film thickness is $h_i$ .The position of $h_o$ is at a distance $H$ from the origin and $h_i$ is at a distance $L$ from the origin. At $h=h_m$, we get maximum pressure at which $\frac{dP}{dy} = 0$.

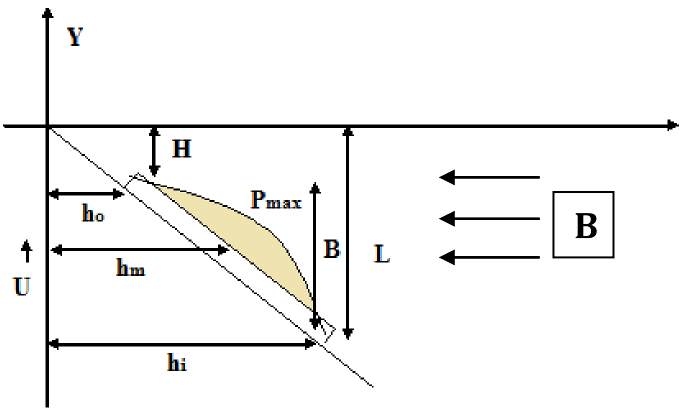


Figure-1 (Geometry of plane inclined slider)

Taking $n$ as:

$$n = \frac{h_i - h_o}{h_o} \tag{1.1}$$

The film thickness $h$ can be expressed at any point as:

$$h = h_o\left(1 + \frac{ny}{L}\right) \tag{1.2}$$

$$h = y \cot \alpha, \tag{1.3}$$

Where $\alpha$ is the angle of inclination of the pad.

$$cot\ \alpha = \frac{h_o}{H} = \frac{h_i}{L} = \frac{dh}{dy} \tag{1.4}$$

$$\frac{dh}{dy} = \frac{nh_o}{L} = \frac{h_i - h_o}{L} \tag{1.5}$$

$$\frac{h_o}{H} = \frac{h_i}{L} = \frac{h_i - h_o}{L-H} = \frac{L\frac{dh}{dy}}{L-H} \tag{1.6}$$

## 2. Governing equations and boundary conditions

The "Extended Generalized Reynolds Equation", which takes into account of the effects of the uniform rotation about an axis that lies across the fluid film and depends on the rotation number $M$ i.e., the square root of the conventional Taylor's Number, given by Banerjee et al. and Miyan (Miyan, 2018) is as follows:

$$\begin{aligned}&\frac{\partial}{\partial x}\left[-\frac{h^3}{12\mu}\left(1-\frac{17M^2\rho^2h^4}{1680\mu^2}\right)\rho\frac{\partial P}{\partial x}\right]+\frac{\partial}{\partial y}\left[-\frac{h^3}{12\mu}\left(1-\frac{17M^2\rho^2h^4}{1680\mu^2}\right)\rho\frac{\partial P}{\partial y}\right]\\&+\frac{\partial}{\partial x}\left[-\frac{M\rho^2h^5}{120\mu^2}\left(1-\frac{31M^2\rho^2h^4}{3024\mu^2}\right)\frac{\partial P}{\partial y}\right]-\frac{\partial}{\partial y}\left[-\frac{M\rho^2h^5}{120\mu^2}\left(1-\frac{31M^2\rho^2h^4}{3024\mu^2}\right)\frac{\partial P}{\partial x}\right]\\&=-\frac{\partial}{\partial x}\left[\frac{\rho U}{2}\left\{h-\frac{M^2\rho^2h^5}{120\mu^2}\left(1-\frac{31M^2\rho^2h^4}{3024\mu^2}\right)\right\}\right]\\&-\frac{\partial}{\partial y}\left[\frac{M\rho^2U}{2}\left\{-\frac{h^3}{12\mu}\left(1-\frac{17M^2\rho^2h^4}{1680\mu^2}\right)\right\}\right]\end{aligned} \tag{2.1}$$

Where; $\rho$ = fluid density, $\mu$ =viscosity, $h$ =film thickness of fluid film, $U$ = sliding velocity, $x, y, z$ =coordinates, $P$ = pressure, $M$=Rotation number (Square root of conventional Taylor's number).

If the bearing is infinitely short, then the pressure gradient in $x$-direction is much smaller than the pressure gradient in $y$-direction. In $y$-direction the gradient $\partial p/\partial y$ is of the order of $(P/L)$ and in the $x$-direction, and is of order of $(P/B)$. If $L << B$, Where; $L$= Length of bearing, $B$= Breadth of the bearing, then

$$\frac{P}{L} \gg \frac{P}{B} \Rightarrow \frac{\partial P}{\partial x} \ll \frac{\partial P}{\partial y} \tag{2.2}$$

We have

$$M^2 = T_a = \frac{4\Omega^2 L^2}{\mu^2} \tag{2.3}$$

$$H_a = L\mathbf{B}\sqrt{\frac{\sigma}{\mu}} \tag{2.4}$$

Where,

$T_a$ =Taylor's number

$H_a$=Hartmann number

$\Omega$ =Characteristic angular velocity

$L$ =Characteristic length scale perpendicular to the direction of rotation

**B**=Magnetic field intensity

$\sigma$ =Electric cunductivity

For the determination of pressure the boundary conditions are:

$\frac{dP}{dy} = 0$; at $h=h_m$ i.e., the pressure gradient vanishes at maximum pressure.

$P=0$ at $h=h_o$ and $P=0$ at $h=h_o\,(1+n)$

**3. Result and Discussion**

Taking the pressure distribution as the function of the coordinate along the length of the slider only, we have $P = P(y)$, we have

$$\frac{d}{dy}\left[-\frac{h^3}{12\mu}\left(1-\frac{17\,\Omega^2 H_a^4 \rho^2 h^4}{420\mu^2 \mathbf{B}^4\sigma^2}\right)\rho\frac{\partial P}{\partial y}\right] = -\frac{d}{dy}\left[\frac{\Omega \mathbf{B}^2\rho^2 U}{\sigma H_a^2}L^3\left\{-\frac{h^3}{12\mu}\left(1-\frac{17\,\Omega^2 H_a^4\rho^2 h^4}{420\mu^2\mathbf{B}^4\sigma^2}\right)\right\}\right] \tag{3.1}$$

So we have the differential equation for the pressure will be

$$\frac{dP}{dy} = \frac{H_a^2\,\Omega}{\mathbf{B}^2\sigma}\rho U\left[\frac{K_m}{K(h)}-1\right];\alpha = \frac{h_i}{h_o} \tag{3.2}$$

$$K(h) = -\frac{h^3}{12\mu}\left(1-\frac{17\,\Omega^2 H_a^4\rho^2 h^4}{420\mu^2\mathbf{B}^4\sigma^2}\right) \tag{3.3}$$

$$K_m = -\frac{{h_m}^3}{12\mu}\left(1-\frac{17\,\Omega^2 H_a^4\rho^2 {h_m}^4}{420\mu^2\mathbf{B}^4\sigma^2}\right) \tag{3.4}$$

If the extent of rotation is small, so that it is reasonable to ignore second and higher powers of ***M*** compared with the first power of ***M*,** the expressions for pressure and load capacity in terms of Hartmann's number is given by

$$P = -\frac{\rho U\,\Omega H_a^2}{n h_0 \sigma \mathbf{B}^2}\,L\left[(h-h_0)+\frac{h_0}{(n+2)\,G(h_0)}\left\{F(h)\frac{{h_0}^2}{h^2}-F(h_0)\right\}\right] \tag{3.5}$$

Where, By taking the values of different mathematical terms in C.G.S. system as: $\mu = 0.112, U = 100, \rho = 3, L = 10, n = 1, y = 7.5, h = 0.015, h_i = 0.02, h_o = 0.01$.

$$F(h) = \left(1-\frac{17\,\Omega^2 H_a^4\rho^2 h^4}{420\mu^2\mathbf{B}^4\sigma^2}\right)$$

$$G(h_0) = \left(1+\frac{17\,\Omega^2 H_a^4\rho^2 {h_0}^4}{420\mu^2\mathbf{B}^4\sigma^2}\right)$$

The dimensionless pressure can be taken as:

$$P^* = H_a^2\,\beta\left[\frac{(\alpha-1)}{n}+\frac{(n+1)^2}{n(n+2)}\left(\frac{1}{\alpha^2}-1\right)\right]$$

Where $\alpha, \beta$ are dimensionless aspect ratios.

$$W = \int_0^L P\,dy = \frac{L}{nh_0}\int_{h_0}^{h_0(1+n)} P\,dh \qquad (3.6)$$

$$W = \frac{\rho U H_a^2\,\Omega L^2 n}{2(n+2)\mathbf{B}^2\sigma} \qquad (3.7)$$

The dimensionless load capacity can be taken as:

$$W^* = H_a^2\,L\frac{n}{2(n+2)}\beta \qquad (3.8)$$

Table 1

| B | $H_a$ | $F(h_0)$ | $G(h_0)$ | $F(h)$ | P |
|---|---|---|---|---|---|
| 2 | 2 | 0.97965 | 1.00402 | 0.97965 | 391.84267 |
| 2 | 4 | 0.93569 | 1.06431 | 0.67442 | 4735.77866 |
| 2 | 6 | 0.67442 | 1.32558 | -0.64823 | 14432.82369 |
| 2 | 8 | -0.02899 | 2.02898 | -4.20293 | 32858.92700 |
| 2 | 10 | -1.51217 | 3.51217 | -11.71784 | 60082.44257 |
| 4 | 2 | 0.99974 | 1.00025 | 0.99873 | 529.91112 |
| 4 | 4 | 0.99598 | 1.00401 | 0.97965 | 5459.25934 |
| 4 | 6 | 0.97965 | 1.02035 | 0.89699 | 13954.45537 |
| 4 | 8 | 0.93569 | 1.06431 | 0.67442 | 26987.71682 |
| 4 | 10 | 0.84300 | 1.15699 | 0.20514 | 46629.67376 |
| 6 | 2 | 0.99995 | 1.00009 | 0.99975 | 235.31443 |
| 6 | 4 | 0.99920 | 1.00158 | 0.99598 | 2412.35568 |
| 6 | 6 | 0.99598 | 1.00798 | 0.97965 | 6046.15875 |
| 6 | 8 | 0.98729 | 1.02523 | 0.93569 | 11155.54791 |
| 6 | 10 | 0.96898 | 1.06161 | 0.84299 | 17786.61911 |
| 8 | 2 | 0.99998 | 1.00098 | 0.99992 | 131.96808 |
| 8 | 4 | 0.99975 | 1.01575 | 0.99873 | 1332.22916 |
| 8 | 6 | 0.99873 | 1.07976 | 0.99356 | 3135.53159 |
| 8 | 8 | 0.99598 | 1.25208 | 0.97965 | 4986.32059 |
| 8 | 10 | 0.99018 | 1.61545 | 0.95032 | 6179.07651 |
| 10 | 2 | 0.99999 | 1.00000 | 0.99997 | 84.70954 |
| 10 | 4 | 0.99989 | 1.00000 | 0.99948 | 868.47139 |
| 10 | 6 | 0.99948 | 1.00004 | 0.99736 | 2176.80656 |
| 10 | 8 | 0.99835 | 1.00015 | 0.99167 | 4017.19362 |
| 10 | 10 | 0.99018 | 1.00037 | 0.97965 | 6409.6075 |

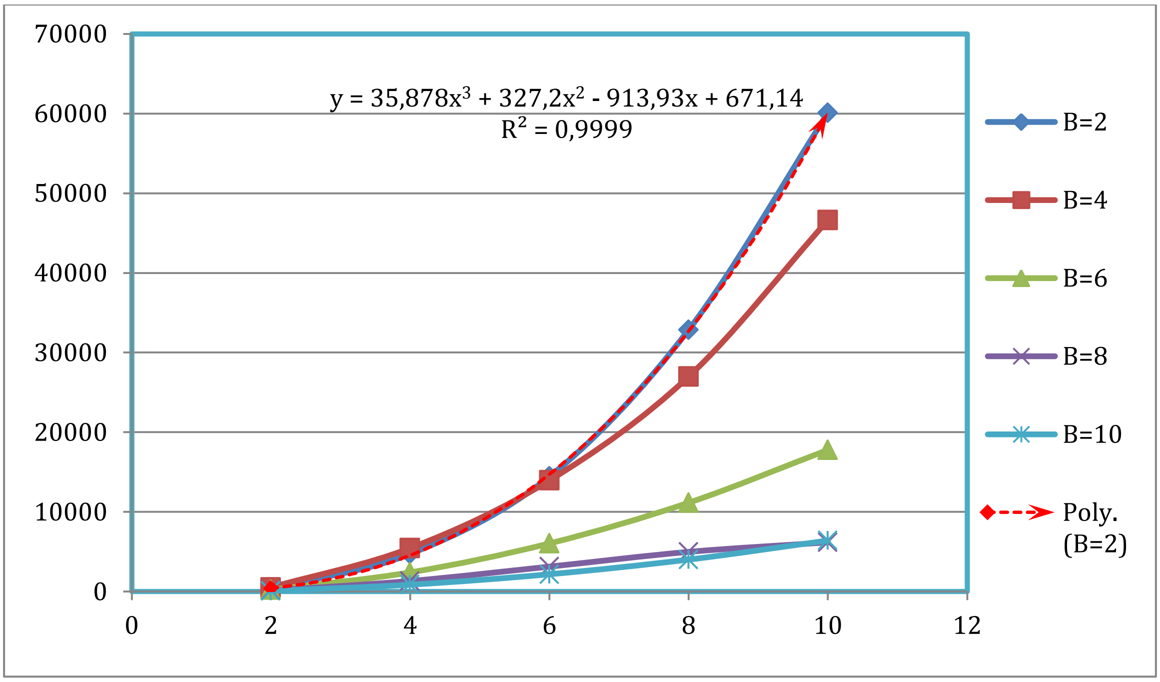


Figure-1 Variation of dimensionless pressure $P^*$ with Hartmann number $Ha$ for different values of Magnetic field

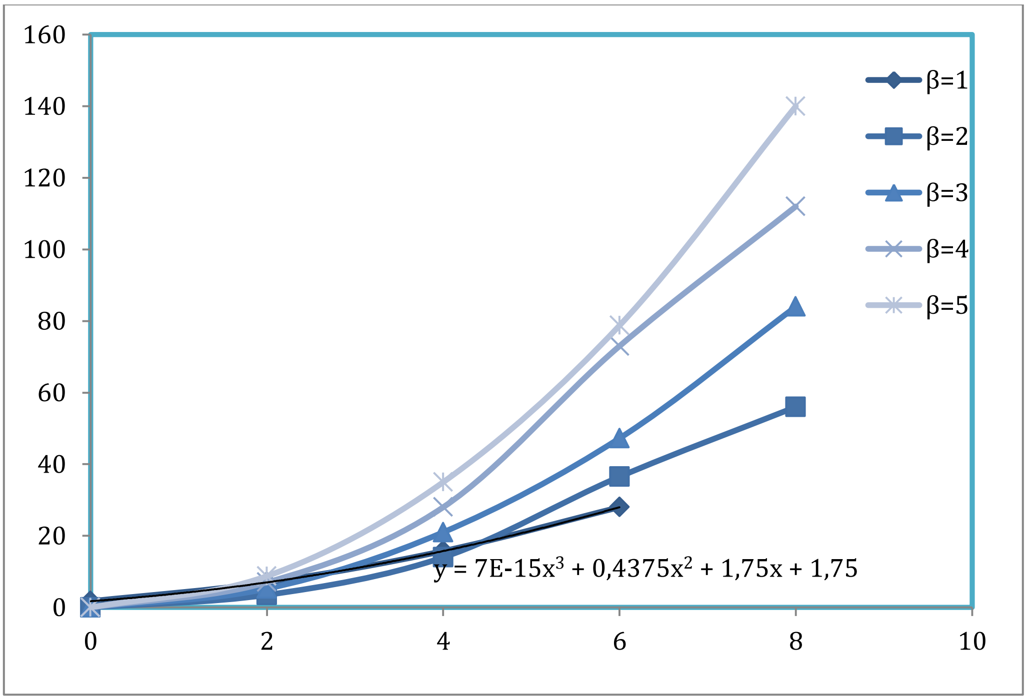


Figure-1 Variation of dimensionless pressure $P^*$ with Hartmann number $Ha$ for $\alpha = 2$ and $\beta = 1, 2\ 3, 4, 5$.

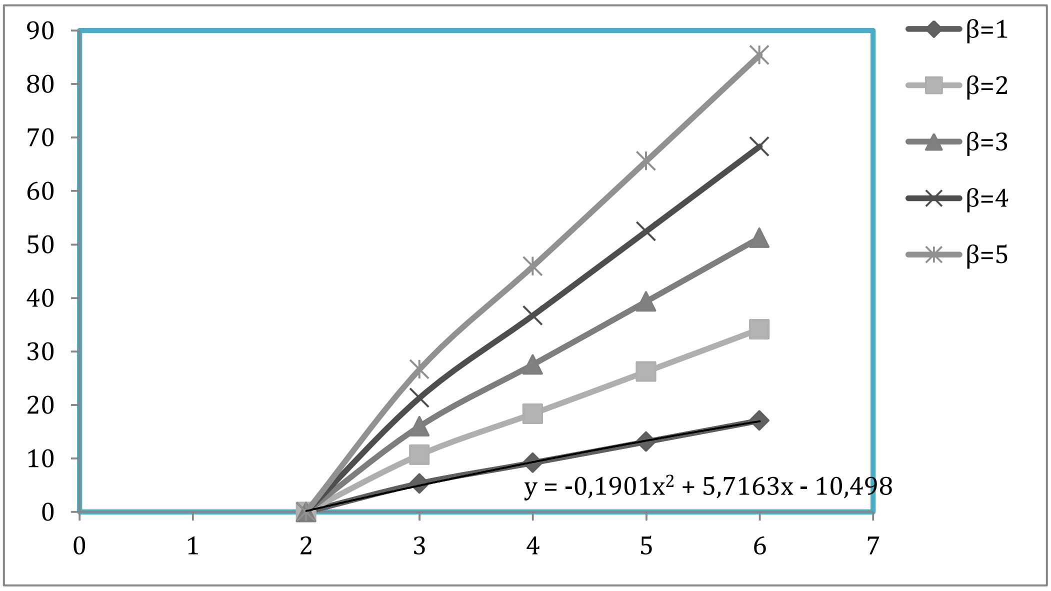


Figure-1 Variation of dimensionless pressure $P^*$ with $\alpha$ for Hartmann number $Ha$=2 and $\beta = 1, 2\ 3, 4, 5.$

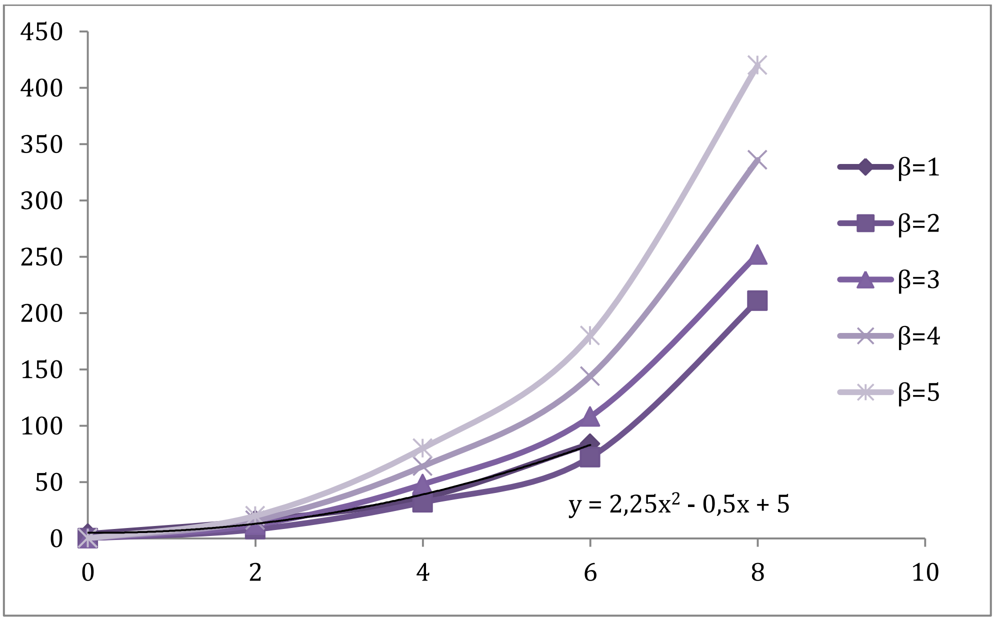


Figure-1 Variation of dimensionless Load Capacity $W^*$ with Hartmann number $H_a$ for $\alpha = 2$ and $\beta = 1, 2\ 3, 4, 5.$

The aspect ratios α and β vary with the density and viscosity of fluid, that increase with increasing the magnetic field. The table and figure show that dimensionless pressure P* and Load carrying capacity W* both varies with α and β, hence P* and W* both increase with increasing the magnetic field. The P* and W* also increases with Hartmann number $H_a$. The parabolic equation for P* and W* with $H_a$ can be found with method of least square approximation as follows:

$$P^* = 0.437\, H_a^2 + 1.75\, H_a + 1.75; \beta = 1$$

$$W^* = 2.25\, H_a^2 - 0.5\, H_a + 5; \beta = 1$$

## 6. CONCLUSION

From the discussion of results, it is noted that the magnetic field increases the density and viscosity of the fluid. The film pressure varies with density and viscosity hence it increases with the intensity of magnetic field. Thus the load carrying capacity of the bearing is enhanced by the application of the magnetic field. In presence of magnetic field there is increase in ratio $\alpha, \beta$ that leads to increased pressure as well as load carrying capacity in the bearing.

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