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THE EFFECT OF MHD AND COUPLESTRESS FLUID ON THE PERFORMANCE CHARACTERISTICS OF WIDE SLIDER BEARING WITH AN EXPONENTIAL AND SECANT FILM PROFILE –A COMPARATIVE STUDY

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ABSTRACT

The theoretical investigation of comparative study between the performance characteristic of Magnetohydrodynamic wide slider bearing with exponential and secant film profile lubricated with couplestress fluids is made. On the basis of Stokes micro-continuum theory, the modified Reynolds equation is derived and closed form expression for fluid film pressure, load carrying capacity, frictional force and coefficient of friction is obtained. The graphs depicting the build-up between load carrying capacity, frictional force and coefficient of friction is plotted. Computations reveal that, exponential slider has significant load carrying capacity and friction as compared to secant slider. To illustrate the use of present study for engineering application, a design example of slider bearing lubricated with an electrically conducting fluid is also presented.

Key Words: *Couplestress fluid; Magneto-hydrodynamics; Exponential and Secant shaped slider bearings.*

1. INTRODUCTION

In modern machine elements different types of lubricants are selected to meet the specific requirements for bearing operating under severe conditions. In order to enhance the lubricating performance, the non-Newtonian lubricant blended with long chain polymers additives is used. The rheological flow behaviour of Non-Newtonian fluids cannot be described by classical micro-continuum theory. So many micro-continuum theories have been proposed [1-2] and applied in lubrication problems such as continuum model of micro polarfluids [3]. Stokes [4] proposed the simplest micro-continuum theory which permits the presence of couplestresses and body couples. Couplestress fluid model is intended to account for particle size effects.

Slider bearings are often designed to support the axial-component thrust in a rotating shaft. Characteristics of slider bearing lubricated with non-Newtonian couplestress fluid have been studied for slider bearing by Ramanaiah and Sarkar[5]. It is shown that load carrying capacity and frictional force are found to increase but coefficient of friction decreases. Lin. J.R and Yu Ming Lu [6] analysed the steady state performance characteristics of wide parabolic shaped slider bearing with couple stress fluid and found that it has higher load carrying capacity and smaller required flow rate. The effect of couplestress lubricant on pressure and load capacity of wide exponential shaped slider bearing was studied by Mobolaji *et al.* [7]. He used Galerkin finite element method and found that effect of couple stress enhances load carrying capacity of the bearing. J. R Lin *et al.*, [8] studied the dynamic characteristic of wide exponential film shape slider lubricated with couplestress fluid and investigated that it has higher load carrying capacity and better damping and dynamic characteristics. Mobolaji H. Oladeinde and John [9] has done a comparative study between parabolic and inclined slider bearing lubricated with couplestress fluids and found that parabolic slider is more superior to inclined slider in load carrying capacity. N.B Naduvinamani *et al.*, [10] have studied the hydrodynamic lubrication of rough slider bearing with couplestress fluids. He observed that for all film shapes the negatively skewed surface roughness increases the load carrying capacity, frictional force and decreases the coefficient of friction, while in positively skewed surface roughness the reverse trend is observed.

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All the above studies of slider bearing focused on the mechanism lubricated with non-conducting fluids. To avoid unexpected variation with temperature the use of liquid metal lubricant has received extensive interest. Comparing with the conventional non-conducting lubricating oils, liquid metals possesses higher thermal and electrical conductivity. The property of high electrical conductivity implies that hydrodynamic flow can be adjusted by the application of an external magnetic field. This indicates that their use as lubricant is promising. Many authors have investigated the Magnetohydrodynamic performance of slider bearings.

In 1962 Snyder [11] was first to present an analysis of MHD infinite inclined plane slider bearing in the presence of uniform transverse magnetic field. The bearing surfaces are assumed to be electrically conducting. It was observed that a significant increase in load capacity was possible with liquid metal lubricants in the presence of magnetic field. For a given magnetic field load carrying capacity occurs if the bearing surfaces are perfect electrical insulators. Hughes [12] analysed the case of MHD inclined slider bearing under general electrical loading conditions. It was found that significant load increase can be affected by supplying external power. If the Hartmann number is large enough open circuit conditions give rise to greatly increased load capacities. Shukla [13] studied optimum load of MHD slider bearing using film thickness and conductivity functions as bounded control variables. It is shown that if the conductivity of the bearing surface is of step-type function, uniformly applied magnetic field is more advantageous. Hughes [14] analysed the MHD finite step slider bearing in the presence of magnetic field applied both tangentially and transversely to the fluid film. For the transverse field it is found that only a slight increase in pressurization can be affected on open circuit conditions and that short circuit conditions is adverse. Anwar [15] studied the MHD characteristic of inclined slider bearing with arbitrary magnetic field. It is found that a non-uniform applied magnetic field gives higher load capacity as compared with uniform magnetic fields. Lin *et al.* [16] studied the dynamic stiffness and damping characteristic of one-dimensional slider bearing lubricated with electrically conducting fluid. JR Lin [17] studied the dynamic characteristic of MHD wide slider bearing with exponential profile. He observed that improvement of dynamic characteristics is more pronounced for large Hartmann number and small for minimum film thickness. The combined effect of MHD and couple stress is of more interest. Biradar Kasinath [18] studied the combined effect of MHD and couple stress on composite slider bearing. He analysed that the load carrying capacity and frictional force increases in comparison with non-magnetic case, but the coefficient of friction decreases. Mohammed Nabhani and Mohammed El Khlifi[19]studied the inertial couple stress effect on infinitely wide slider bearing and found that couple effect of fluid inertia forces, MHD and non-Newtonian couple stress provides a significant improvement in slider bearing load carrying capacity. Recently Syeda *et al.* [20] studied the combined effect of MHD and couple stresses on the characteristics of different finite plates and found that the effects are prominent in case of circular plates as compared to the other plates.

The aim of the current paper is to conduct a comparative study between MHD exponential and secant shaped slider bearing lubricated with conducting couplestress fluids.

2. MATHEMATICAL FORMULATION

The figure 1 represents the physical configuration of a wide MHD slider bearings lubricated with couplestress fluid with exponential and secant film profile. The bearing is of length L. It consists of two surfaces where x-axis is taken along lower plane across its length while the z-axis is taken across the lubricant film. The lower surface of the bearing is moving with a constant velocity U in its own plane while the upper surface is at rest.

The exponential slider film thickness is described as

$$h = h_0 \exp[-(x/L)\ln r], \tag{1}$$

$$r = \frac{h_1}{h_0} = \frac{d+h_0}{h_0},$$

where $d = h_1 - h_0$ is difference between inlet and outlet film thickness

The secant slider film thickness is described as

$$h = h_0 sec\left\{a\left(1 - x/L\right)\right\}, \tag{2}$$

$$a = sec^{-1}(\delta + 1),$$

where $\delta = \dfrac{h_1 - h_0}{h_0}.$

A uniform transverse magnetic field B_0 is applied perpendicular to the bearing. Assuming that the fluid film is thin and lubricant in the film region is incompressible stokes couple stress fluid in which body forces and body couples are absent. The basic equations governing the hydromagnetic flow for the couplestress lubricant are

$$\frac{\partial u}{\partial x}+\frac{\partial w}{\partial z}=0, \tag{3}$$

$$\mu\frac{\partial^2 u}{\partial z^2}-\eta\frac{\partial^4 u}{\partial z^4}-\sigma B_0^2\,u=\frac{\partial p}{\partial x}+\sigma E_y\,B_0, \tag{4}$$

$$\frac{\partial p}{\partial z}=0. \tag{5}$$

and $M_0=\sqrt{B_0h_0(\sigma/\mu)}$ is the Hartmann number, σ is electrical conductivity and $l=\sqrt{\eta/\mu}$ is the couplestress parameter, μ is the viscosity coefficient and η is the material constant characterizing couplestress.

The relevant no-slip boundary conditions are at the upper surface $z=h$

$$u=0,\quad \frac{\partial^2 u}{\partial z^2}=0,\quad w=0. \tag{6}$$

At the lower surface $z=0$

$$u=U,\quad \frac{\partial^2 u}{\partial z^2}=0,\quad w=0. \tag{7}$$

where u,v,w representsthe components of velocity in x,y,z directions respectively.

If the bearing surfaces are perfect insulators and there is circuit external to the fluid film, then the electric field may be approximated by requiring the net current flow to be zero.

$$\int_{y=0}^{h}\left(E_y+B_0u\right)dz=0 \tag{8}$$

Solving (4) together with boundary conditions (6), (7) and by condition (8) gives the component of velocity as

$$u=-\frac{h_0^2}{\mu M_0^2}\frac{\partial p}{\partial x}\frac{h}{2l}\xi_1+\frac{U}{2}\xi_2 \tag{9}$$

where,

$$\xi_1=\left[\frac{A^2S_1-B^2S_2}{\frac{A^2}{B}\tanh\frac{Bh}{2l}-\frac{B^2}{A}\tanh\frac{Ah}{2l}}\right]\quad \xi_2=\frac{1}{(A^2-B^2)}\left\{A^2S_1-B^2S_2\right\}$$

and

$$S_1=\frac{Sinh\frac{Bh}{l}-Sinh\frac{Bz}{l}+Sinh\frac{B(h-z)}{l}}{Sinh\frac{Bh}{l}}\quad S_2=\frac{Sinh\frac{Ah}{l}-Sinh\frac{Az}{l}+Sinh\frac{A(h-z)}{l}}{Sinh\frac{Ah}{l}}$$

Integrating the continuity equation (2) over the film thickness by the using equation (6), (7) and (9) we get modified one dimensional Reynolds equation in the form

$$\frac{\partial}{\partial x}\left[\frac{6h_0^2h^2}{\mu lM_0^2}\left\{\frac{\left(A^2-B^2\right)}{\frac{A^2}{B}\tanh\frac{Bh}{2l}-\frac{B^2}{A}\tanh\frac{Ah}{2l}}-\frac{2l}{h}\right\}\frac{\partial p}{\partial x}\right]=6U\frac{dh}{dx} \tag{10}$$

Introducing non-dimensional quantities

$$x^*=\frac{x}{L},\; P^*=\frac{p^*h_0^2}{\mu UL},\; l^*=\frac{2l}{h_0},\; h^*=\frac{h}{h_0},\; M_0=B_0h_0\left(\frac{\sigma}{\mu}\right)^{1/2},\; \delta=\frac{h_1-h_0}{h_0},\; r=\delta+1$$

By using non-dimensional quantities equation (10) takes the form

$$\frac{\partial}{\partial x^*}\left\{f(h^*,l^*,M_0)\frac{\partial P^*}{\partial x}\right\}=6\frac{dh^*}{dx^*}, \tag{11}$$

where, $$f(h^*,l^*,M_0)=\frac{12h^{*2}}{l^*M_0^2}\left\{\frac{\left(A^{*2}-B^{*2}\right)}{\frac{A^{*2}}{B^*}\tanh\frac{B^*h^*}{l^*}-\frac{B^{*2}}{A^*}\tanh\frac{A^*h^*}{l^*}}-\frac{l^*}{h^*}\right\}. \tag{12}$$

The ambient boundary conditions

(i) For secant slider bearing is $P^*=0\ at\ x^*=0,1$

(ii) For exponential slider bearing $P^*=0\ at\ x^*=0,-1$

Integrating both sides of equation (11) w. r. t. x^* twice

The non-dimensional pressure in the film region as

$$P^*=6\int_{x^*=0}^{x^*}\frac{h^*}{f^*(h^*,l^*,M_0)}dx^*+C_1\int_{x^*=0}^{x^*}\frac{1}{f^*(h^*,l^*,M_0)}dx^*, \tag{13}$$

where,
For secant slider

$$C_1=-\frac{6\int_{x^*=0}^{1}\frac{h^*}{f^*(h^*,l^*,M_0)}dx^*}{\int_{x^*=0}^{1}\frac{1}{f^*(h^*,l^*,M_0)}dx^*}. \tag{14}$$

For exponential slider

$$C_1=-\frac{6\int_{x^*=0}^{-1}\frac{h^*}{f^*(h^*,l^*,M_0)}dx^*}{\int_{x^*=0}^{-1}\frac{1}{f^*(h^*,l^*,M_0)}dx^*}. \tag{15}$$

The load per unit width is given by $w=\int_0^L pdx$

The dimensionless load carrying capacity W^* is

$$W^*=\int_{x^*=0}^{1}pdx^* \tag{16}$$

$$W^*=6\int_0^1\int_{x^*=0}^{x^*}\frac{h^*}{f^*(h^*,l^*,M_0)}dx^*dx^*+C_1\int_0^1\int_{x^*=0}^{x^*}\frac{1}{f^*(h^*,l^*,M_0)}dx^*dx^* \tag{17}$$

The frictional force on the bearing surface is

$$F=\int_0^L\left(\tau_{zx}\right)_{z=0}dx$$

$$F=\int_0^L\left[-\frac{\mu lM_0^2U}{2h_0^2(A^2-B^2)}\left\{\frac{A^2}{B}\coth\frac{Bh}{2l}-\frac{B^2}{A}\coth\frac{Ah}{2l}\right\}-\frac{h}{2}\frac{\partial p}{\partial x}\right]dx \tag{18}$$

The non-dimensional frictional force is

$$F^*=-\frac{Fh_0}{\mu UL}=\int_0^1\left\{G(h^*,l^*,M_0)+\frac{h^*}{2}\frac{\partial P^*}{\partial x^*}\right\}dx^* \tag{19}$$

where

$$G(h^*,l^*,M_0)=\frac{l^*M_0^2}{4(A^{*2}-B^{*2})}\left(\frac{A^{*2}}{B^*}\coth\frac{B^*h^*}{l^*}-\frac{B^{*2}}{A^*}\coth\frac{A^*h^*}{l^*}\right)$$

$$F^* = \int_0^1 G(h^*, l^*, M_0)dx^* + 3\int_0^1 \left\{ \frac{h^*}{\xi(h^*, l^*, M_0)} \right\} dx^* + \frac{1}{2} C_1 \int_0^1 \left(\frac{1}{\xi(h^*, l^*, M_0)} \right) dx^* \tag{20}$$

$$where \quad \xi(h^*, l^*, M_0) = \frac{12h^*}{l^* M_0^2} \left\{ \frac{\left(A^{*2} - B^{*2}\right)}{\frac{A^{*2}}{B^*} \tanh \frac{B^* h^*}{l^*} - \frac{B^{*2}}{A^*} \tanh \frac{A^* h^*}{l^*}} - \frac{l^*}{h^*} \right\}, \tag{21}$$

The coefficient of friction

$$C = \frac{F^*}{W^*} \tag{22}$$

3. RESULT AND DISCUSSION

The modified Reynolds equations is solved analytically and to study various bearing characteristics such as pressure distribution, load carrying capacity, frictional force, coefficient of friction in the presence of transverse magnetic field, the numerical methods is employed.

3.1 Pressure distribution

The Figure 2 and Figure 3 depicts the variation of non-dimensional pressure P^* for exponential and secant shaped slider bearing. As the value of couplestress parameter increases in the presence of applied magnetic field, it is observed that pressure in the fluid film increases in both the sliders.

3,2 Non-dimensional load carrying capacity:

Figure 4 and Figure 5 shows the comparison between non-dimensional load carrying capacity $W^*$for both the sliders. In Figure 4 comparison between Newtonian fluid (l^*=0) and couplestress fluid (l^*=0.4)is examined. In Figure 5 comparison between non-magnetic (M_0 =0) and magnetic (M_0 =3) case are examined. The plots with solid line correspond to exponential slider bearing pressures and the plots corresponding to dotted lines indicate secant shaped slider bearing. The application of magnetic field reduces the velocity of the lubricant. As a result large amount of fluid is collected in film region which results in the pressure rise. It is also observed that increasing the values of couplestress parameter l^* increases the build-uppressure in both the film profiles. Hence the load carrying capacity increases for the bearings. It is also observed that the effect of magnetic field increased the build-up pressure in the film region, hence, the load carrying capacity of the bearing has increased. From the graphs and calculations, the load carrying capacity of the slider bearing with exponential profile is found to be more prominent. The load carrying capacities of both the sliders are presented in the table format with $\delta = 1.5$in Table-1.

3.3 Non-dimensional frictional force

Figure 6 and 7 represents variations of frictional force F^* against shoulder parameter δ. It is observed that in the presence of applied magnetic field by increasing the value of couplestress parameter frictional force can be increased. The frictional force is more in case of exponential slider than in secant slider.

1.2 The Coefficient of friction

In Figure 8 and 9 the coefficient of friction C is plotted against the shoulder parameter δ. In Figure 8, plot is made showing comparison between Newtonian and couplestress stress fluids for both the sliders. It is observed that in couplestress fluids the coefficient of friction is less in comparison with the Newtonian fluid. It is also observed that exponential slider is having less coefficient of friction than secant slider. Figure 9 represents the comparison between Non-magnetic and magnetic case for both film profiles with increasing value of couplestress parameter and there is decrease in coefficient of friction.

The relative percentage increase in non-dimensional load R_{W^*}, non-dimensional frictional force R_{F^*} and coefficient of frictions R_C is given by

$$R_{w^*} = \{(W^*_{magnetic} - W^*_{non-magnetic}) / W^*_{magnetic}\} \times 100$$

$$R_{F^*} = \{(F^*_{magnetic} - F^*_{non-magnetic}) / F^*_{magnetic}\} \times 100$$

$$R_C = \{(C_{magnetic} - C_{non-magnetic}) / C_{magnetic}\} \times 100$$

and is evaluated for the both slider bearing and is presented in Table-2. It shows that in exponential slider bearing relative percentage increase in non-dimensional load,non-dimensional frictional force and coefficient of friction is more as compared to secant slider.

2. CONCLUSION

The comparative study of slider bearing, with exponential and secant film profile lubricated with couplestress fluid in the presence of transverse magnetic field is presented in this paper.

From the results obtained the following conclusions are made.

- The effect of magnetic field M_0 increases the build-up pressure in both the film profiles.
- The couplestress fluids l^* contain microstructure additives and are sensitive to applied magnetic field M_0. As a result the enhancement in load carrying capacity is visible. It is interesting to note that, as compared with the secant shaped profile, exponential profile is showing significant load carrying capacity.
- The non-dimensional frictional force F^* increases with increasing values of couplestress parameter l^* and the effect is prominent in the bearing with exponential profile.
- The coefficient of friction C decreases with increasing values of couplestress parameter l^* and is less for exponential slider.
- The non-dimensional relative load carrying capacity R_{W^*}, frictional force R_{F^*}, coefficient of friction R_C is also presented in the form of table corresponding to the shoulder parameter $\delta = 1.5$ for both the film profiles.
- From Table-2 it is seen that for $l^* = 0.2$, and $M_0 = 4$ there is significant increase of 73%,199% and 72% in R_{W^*} R_{F^*} and R_C respectively for exponential slider, where as for secant slider profile we observe an increase of 36%,135%,72% in R_{W^*} R_{F^*} and R_C respectively. Hence, it is concluded that exponential slider profile is more prominent to the combined effect couplestresses and external magnetic field as compared to secant slider bearings.
- It is expected that these results helps design engineers to choose the suitable film profile with suitable parameters of couplestress and magnetic field to enhance the life of the slider bearing. To guide the present study a design example is presented in Table -3.

NOMENCLATURE

B_0 applied magnetic field
C coefficient of friction
d Inlet-outlet film thickness difference $(h_1 - h_0)$
F Frictional force
F^* Non-dimensional frictional force $\left(= -Fh_0 / \mu UL\right)$
h Film thickness
h_1 Inlet film thickness
h_0 Outlet film thickness
H non-dimensional film thickness $\left(h / h_0\right)$
l Couplestress parameter
L Bearing length
M_0 Hartmann number
p Pressure in the film region
P^* Non-dimensional pressure
x, y Rectangular co-ordinates
x^* Non-dimensional rectangular coordinates $\left(= x / L\right)$
u, v Velocity component in film region
w Load carrying capacity
W^* Non-dimensional load carrying capacity $\left(= wh_0^2 / \mu UL^2\right)$
δ Non-dimensional inlet-outlet thickness difference $\left(= d / h_0\right)$
η Material constant characterizing couple stress
μ Viscosity coefficient
σ Electrical conductivity

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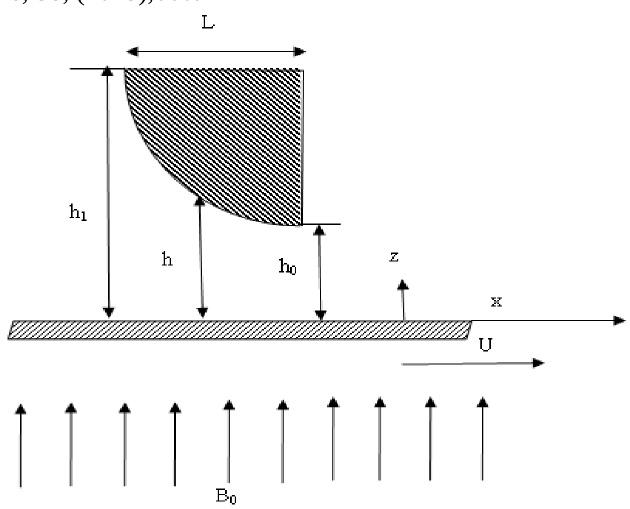

Figure-1: Physical geometry of a wide slider bearing lubricated with MHD and couplestress fluid.

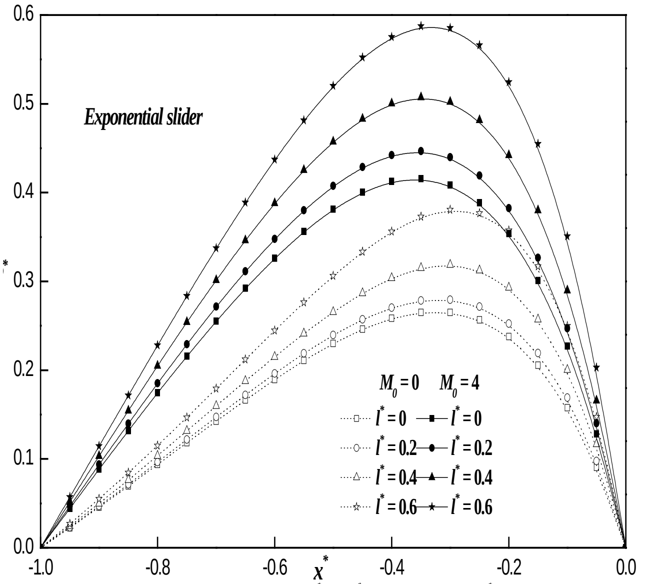

Figure 2 Variation of non-dimensional pressure P^* with x^* for different values of l^* and M_0 with $\delta = 1.5$.

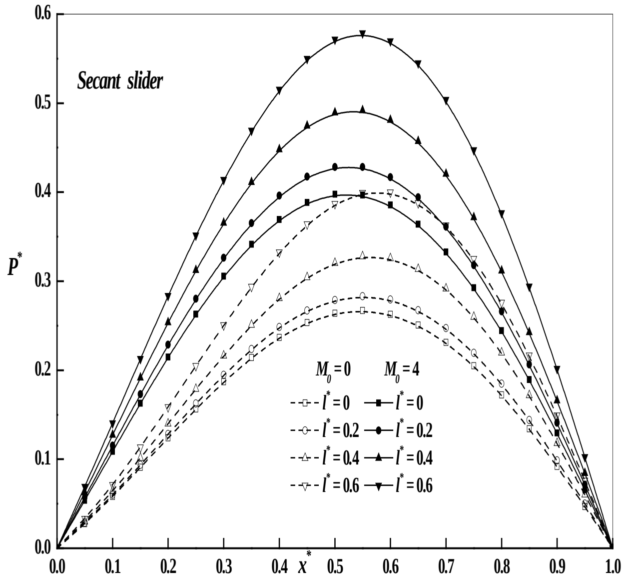

Figure 3 Variation of non-dimensional pressure P^* with x^* for different values of l^* and M_0 with $\delta = 1.5$.

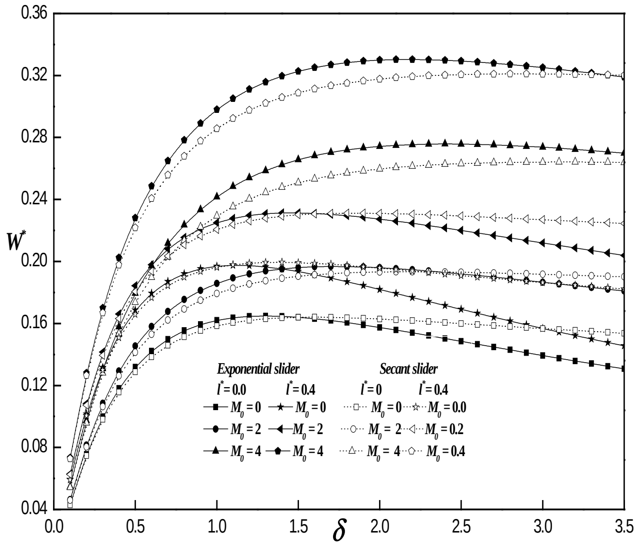

Figure 4 Variation of non-dimensional load W^* with δ for different values of l^* and M_0.

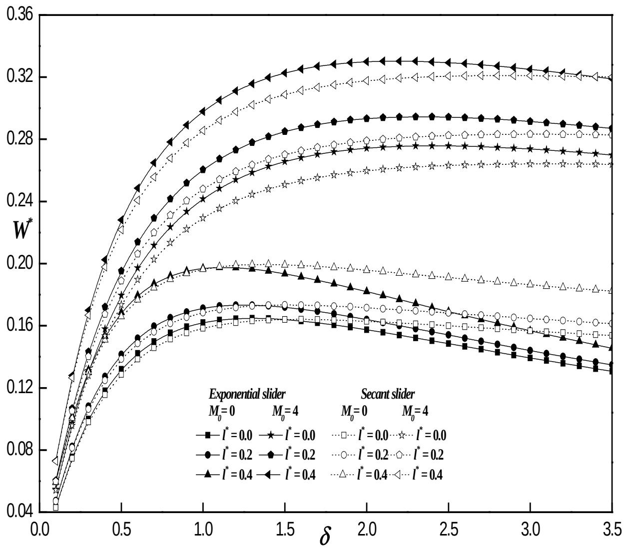

Figure 5 Variation of non-dimentional frictional force F^* with δ for different values of l^* and M_0.

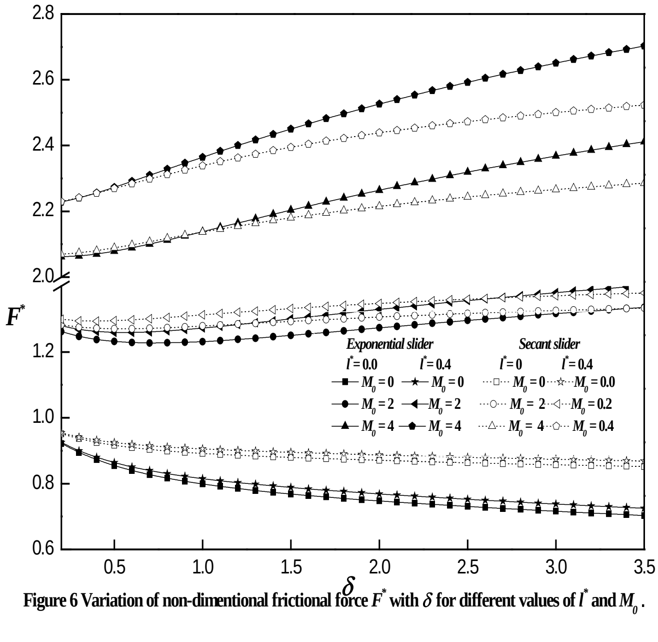

Figure 6 Variation of non-dimentional frictional force F^* with δ for different values of l^* and M_0.

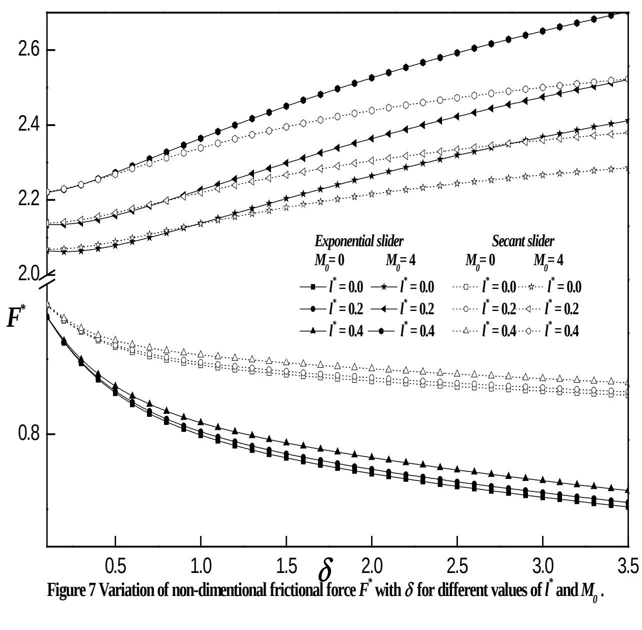

Figure 7 Variation of non-dimentional frictional force F^* with δ for different values of l^* and M_0.

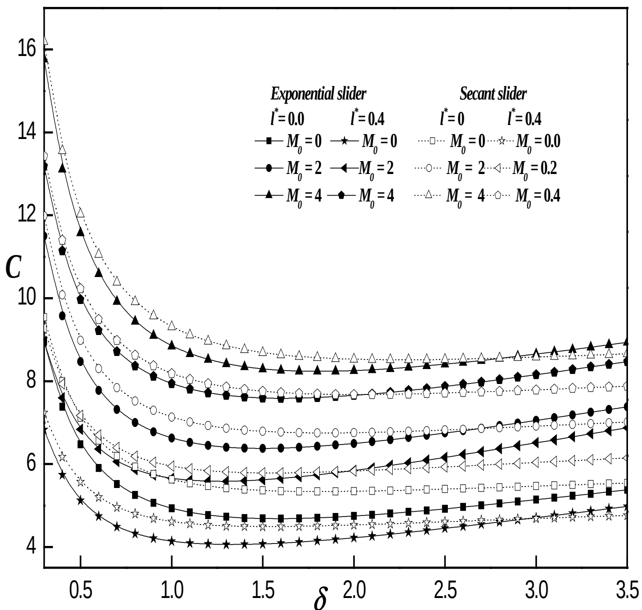

Figure 8 Variation of non-dimentional co efficient of friction C with δ for different values of M_0 and l^*.

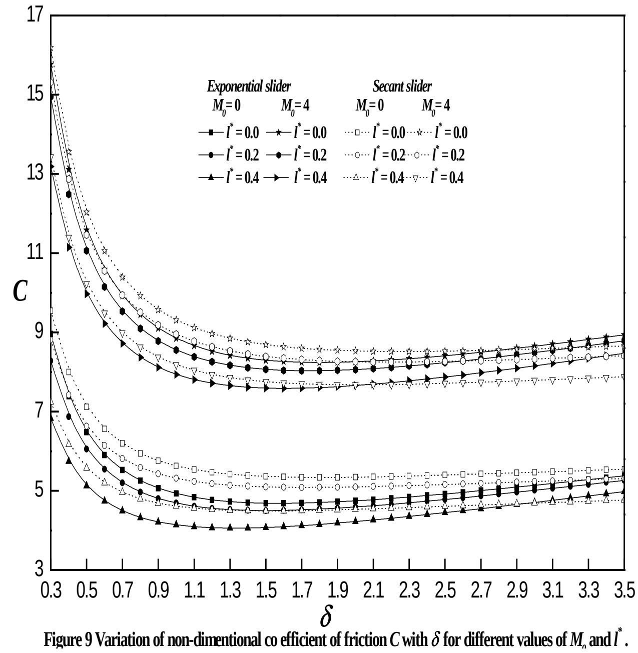

Figure 9 Variation of non-dimentional co efficient of friction C with δ for different values of M_0 and l^*.

Table-1: Comparison of load carrying capacity for exponential and secant slider bearing

Geometry	M_0	$l^*=0.0$	$l^*=0.1$	$l^*=0.2$	$l^*=0.3$	$l^*=0.4$
Exponential	$M_0=0$	0.163993	0.166070	0.171872	0.181074	0.193472
	$M_0=1$	0.172765	0.174990	0.181122	0.190703	0.203485
	$M_0=2$	0.196086	0.198916	0.206249	0.217190	0.231340
	$M_0=3$	0.228418	0.232487	0.242166	0.255692	0.272425
	$M_0=4$	0.265676	0.271737	0.284966	0.302265	0.322709
Secant	$M_0=0$	0.164019	0.166472	0.173345	0.184315	0.199231
	$M_0=1$	0.171123	0.173702	0.180840	0.192089	0.207264
	$M_0=2$	0.190475	0.193582	0.201705	0.213986	0.230121
	$M_0=3$	0.218174	0.222387	0.232511	0.246876	0.264956
	$M_0=4$	0.199231	0.207264	0.230121	0.264956	0.308875

Table-2: Variation of R_{W^*}, R_F and R_C with l^* for different values M_0 with $\delta = 1.5$

l^*	M_0	R_W		R_F		R_C	
		Exponential	**Secant**	**Exponential**	**Secant**	**Exponential**	**Secant**
0	2	19.56974	9.802703	62.70537	38.63665	36.07554	26.25997
	4	62.00448	34.86018	186.7748	127.9884	77.01653	69.05507
	6	111.8292	68.00451	322.6617	233.9711	99.52916	98.78657
2	2	25.76695	9.862531	64.89057	39.33883	31.10785	26.82992
	4	73.76717	36.47115	199.1350	135.7939	72.14660	72.77935
	6	134.0606	74.32710	356.3963	258.5963	94.99027	105.7021
4	2	41.067	8.813749	69.22713	40.07609	19.96145	28.73006
	4	96.78218	33.62550	218.9202	145.4652	62.06716	83.69692
	6	169.7341	70.98951	404.9613	290.0163	87.20695	128.0944

Table-3: Numerical Example of the slider bearings lubricated with an electrically conducting fluid

Physical Parameter	**Notation**	**Range of values chosen**
Steady minimum film thickness	h_{m0}	$1.00\times10^{-4}m$
Difference between the inlet and outlet film thickness	d	$(0.5,1.0,1.5,2.0,2.5)\times10^{-4}m$
Lubricant viscosity	μ	$1.55\times10^{-4}Pa.s$
Electrical conductivity	σ	$1.07\times10^{6}mho/m$
Magnetic field	B_0	$0,0.75,1.5,2.25,3,3.75Wb/m^2$
Material Constant characterizing couplestresses	η	$(1.25,5.0,11.25)\times10^{-15}Ns$

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