Thermal Effects on Hydrodynamic Journal Bearings Lubricated by Magnetic Fluids with Couple Stresses

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***Abstract*-- Based on the momentum and continuity equations for ferrofluid under an applied magnetic field, assuming linear behavior for the magnetic material of the ferrofluid, and using carrying current concentric finite wire magnetic field model, the magnetic force was calculated. A modified Reynolds equation is obtained. It is simultaneously solved with the energy equation numerically by the finite difference technique. The pressure and temperature distributions have been obtained under various couple stress parameter. The solution renders the bearing performance characteristics namely; load carrying capacity, attitude angle of the journal center, frictional force at the journal surface, friction coefficient and bearing side leakage.**

Based on the micro-continuum theory, and by taking into account the couple stresses due to the microstructure additives, the effects of couple stresses on the performance of a finite hydrodynamic journal bearings were studied. The results have shown that fluids with couple stresses are better than Newtonian fluids.

***Index Term*-- Magnetized journal bearings; Ferrofluid lubricants; Static characteristics; Couple stresses.**

I. INTRODUCTION

Recent solutions that devoted to the study of the lubricant properties have been devised with other parameter being considered in predicting the journal bearing behavior, such as load variation, type of lubricant flow region (laminar or turbulent flow), type of lubricant (Newtonian or non Newtonian), inertia and acceleration effects, and magnetic effect in the case of using ferrofluid [1].

Most of these investigations did not consider the thermal aspects (used isothermal solution) and neglecting viscosity variation with change of temperature. In the practical cases, increasing in load capacity leads to increasing the friction force and consequently increasing the temperature. Although many aspects of bearing performance are fully solved, there are still needs for further investigations of the thermal effects on the bearing performance.

The ferrofluid consists of three basic components namely; a base fluid or carrier fluid, ferromagnetic particles and a coating on each particle [2]. The carrier fluid may be a diester base, a hydrocarbon base, an ester base and even a water base; other base fluids, less important for lubrication, are also available.

Ferrolubricants behave like ferrofluids; therefore, these lubricants can be controlled remotely by a magnetic field and they can be positioned exactly where, and only where, wear would be expected to take place. The ability to position the lubricant externally is valuable in clean-environment applications, because the lubricant does not contaminate the environment. The greatest advantages of the magnetic-oil-lubricated bearings are a long life, low friction and reduced noise.

The study of the thermal aspects needs simultaneous solution of the basic Reynolds' equation coupled with the energy equation and heat transfer, thermal distribution problems may also be considered, thus the solution is assumed that the total viscous heating is dissipated in the oil only with negligible influence of the environmental conditions, assuming no heat conducted to or from the oil film to surfaces (adiabatic case).

Applications of ferrofluids are usually based on their controllability by an external magnetic force [3]. Among the various applications in engineering, are those taking advantage of the possibility of collecting and holding firmly small quantities of such fluids in region with highly focused magnetic fields. From this point of view, the ferrofluids were used in liquid seals [4, 5, 6], hydrodynamic braking [7] and also in the lubrication of journal bearings with small or non-existing side leakage [8, 9, 10].

By taking into account the couple stresses due to the microstructure additives, the effects of couple stress on the performance of a finite journal bearing, are presented.

II. GOVERNING EQUATIONS

For a ferrofluid under a magnetic field, the unit volume value of the induced magnetic force is given by [12, 13]:

$$f_m = (curl\,\mathrm{h_m}) \times \mathrm{B} + \mu_o M_g \; grad\, h_m \tag{1}$$

Where B is the magnetic field density vector and ($curl$ $\mathrm{h_m}$) represents the induced free current. The first term, then, can be cancelled and the equation is rewritten as:

$$f_m = \mu_o M_g \; grad\, h_m \tag{2}$$

If a small or moderate applied field is applied; the magnetization of the fluid is approximately proportional to the applied field. $M_g = X_m h_m$, and the induced magnetic force is given by:

$$f_m = \mu_o X_m h_m \; grad\, h_m \tag{3}$$

III. MODIFIED REYNOLDS EQUATIONS

Newtonian ferrofluid lubricant

Starting from the above Navier-Stokes equations, and using the magnetic force as a body external force, the equations of motion are derived for the fluid film,with the following assumptions:

1- The curvature of the fluid film is neglected, since the film thickness in y-direction is very thin compared with the span in x and z-directions. Thus, no gradient of the applied magnetic field across the fluid film, no magnetic force in y-direction and thus no pressure gradient in this direction ($\partial p / \partial y = 0$).

2- The flow is laminar; consequently, neither vortex flow nor turbulence is occurring anywhere in the flow.

3- The fluid inertia force is neglected compared to the viscous force and the induced magnetic force. Thus, $\frac{d\mathrm{v_x}}{dt} = \frac{d\mathrm{v_y}}{dt} = \frac{d\mathrm{v_z}}{dt} = 0.0$

4- No slipping at the bearing surface

5- The lubricant is assumed incompressible; i.e. its density is constant.

6- Except $\partial \mathrm{v_x}/\partial_y$ and $\partial \mathrm{v_z}/\partial_y$, all other velocity gradients are considered negligible.

7- The lubricant is assumed to be Newtonian fluid.

8- No heat conducted to or from the oil film to surfaces (adiabatic case).

The momentum equations, thus, can be written as:

$$0 = -\frac{\partial p}{\partial x} + \lambda \frac{\partial^2 V_x}{\partial y^2} + f_{mx} \tag{4}$$

$$0 = -\frac{\partial p}{\partial z} + \lambda \frac{\partial^2 V_z}{\partial y^2} + f_{mz} \tag{5}$$

Where f_{mx} and f_{mz} are the magnetic force components in circumferential and axial directions respectively.

The boundary conditions are:

$$\mathrm{v_x} = \omega \cdot R\,, \quad \mathrm{v_y} = \mathrm{v_s}\,, \quad \mathrm{v_z} = 0 \quad \text{at } y = h \tag{6}$$

$$\mathrm{v_x} = \mathrm{v_y} = \mathrm{v_z} = 0 \quad \text{at } y = 0 \tag{7}$$

The velocity profiles in circumferential and axial directions are obtained.

$$V_x = \frac{1}{2\lambda}\frac{\partial p}{\partial x}\left(y^2 - yh\right) - \frac{1}{2\lambda} f_{mx}\left(y^2 - yh\right) + \omega R \frac{y}{h} \tag{8}$$

$$V_z = \frac{1}{2\lambda}\frac{\partial p}{\partial z}\left(y^2 - yh\right) - \frac{1}{2\lambda} f_{mz}\left(y^2 - yh\right) \tag{9}$$

These distributions are substituted in the following integrated continuity equation.

$$\int_0^h \frac{\partial \mathrm{v_x}}{\partial x} dx + \int_0^h \frac{\partial \mathrm{v_y}}{\partial y} dy + \int_0^h \frac{\partial \mathrm{v_z}}{\partial z} dz = 0 \tag{10}$$

The generalized pressure equation is obtained in the form:

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\lambda}\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(\frac{h^3}{\lambda}\frac{\partial p}{\partial z}\right) = 6U\frac{\partial h}{\partial x} + \frac{\partial}{\partial x}\left(h^3 f_{mx}\right) + \frac{\partial}{\partial z}\left(h^3 f_{mz}\right) \tag{11}$$

Using equation (3), the components of magnetic force (f_{mx} and f_{mz}) can be obtained.

$$f_{mx} = \mu_o X_m h_m \frac{\partial h_m}{\partial x} \tag{12}$$

$$f_{mz} = \mu_o X_m h_m \frac{\partial h_m}{\partial z} \tag{13}$$

The pressure equation is rewritten in non-dimensional form as:

$$\frac{\partial}{\partial \theta}\left(\frac{H^3}{\bar{\lambda}}\frac{\partial P}{\partial \theta}\right) + \frac{1}{4\nu^2}\frac{\partial}{\partial Z}\left(\frac{H^3}{\bar{\lambda}}\frac{\partial P}{\partial Z}\right) = 6\frac{\partial H}{\partial \theta} + 4\nu^2 \alpha \frac{\partial}{\partial \theta}\left(H^3 H_m \frac{\partial H_m}{\partial \theta}\right) + \alpha \frac{\partial}{\partial Z}\left(H^3 H_m \frac{\partial H_m}{\partial Z}\right) \tag{14}$$

It is a general modified non-dimensional Reynolds equation for Newtonian hydrodynamic lubrication.

Non-Newtonian lubricant with Couple stresses

The modified Reynolds equation is obtained as in G.S. Nada et al [15]:

$$\frac{\partial}{\partial\theta}\left(\frac{G(H,L^*)}{\bar{\lambda}}\frac{\partial p}{\partial\theta}\right)+\frac{1}{4\nu^2}\frac{\partial}{\partial Z}\left(\frac{G(H,L^*)}{\bar{\lambda}}\frac{\partial p}{\partial Z}\right)=6\frac{\partial H}{\partial\theta}$$
$$+4\nu^2\alpha\frac{\partial}{\partial\theta}\left(G(H,L^*)\,H_m\frac{\partial H_m}{\partial\theta}\right)+\alpha\frac{\partial}{\partial Z}\left(G(H,L^*)\,H_m\frac{\partial H_m}{\partial Z}\right)$$
(22)

Where

$$G(H,L^*)=H^3-12(L^*)^2H+24(L^*)^3\tanh\left(\frac{H}{2L^*}\right)$$

, $\alpha=\dfrac{(h_{mo})^2\mu_o X_m c^2}{\lambda_o\omega L^2}$

It is also noted that, as the value of (L^*) equals zero, equation (15) reduced to the Newtonian magnetic lubricant case and the effect of couple stresses vanishes.

IV. Energy Equations

Temperature field in the journal bearing which is generated in oil due to hydrodynamic action and viscous shear under adiabatic conditions can be represented by the following non-dimensional form energy equation [11]:

$$H^2\left[\left(1-\frac{H^2}{6\bar{\lambda}}\frac{\partial P}{\partial\theta}\right)\frac{\partial T}{\partial\theta}-\frac{H^2}{6\bar{\lambda}}\frac{1}{4\nu^2}\frac{\partial P}{\partial Z}\frac{\partial T}{\partial Z}\right]=$$
$$=12\bar{\lambda}\left[1+\frac{H^4}{12\bar{\lambda}^2}\left(\left(\frac{\partial P}{\partial\theta}\right)^2+\frac{1}{4\nu^2}\left(\frac{\partial P}{\partial Z}\right)^2\right)\right]$$
(15)

Equations (14) and (15) are then simultaneously solved numerically using Gauss-Sidel iteration method. The final solution is obtained after successive iterations, beginning with an initial distribution guess of zero values. The results, with no magnetic effects or thermal aspects, have complete agreement with that of Pinkus [14].

5. Bearing Static Characteristics

Load carrying capacity and attitude angle

$$W_\varepsilon=2\int_0^{0.5}\int_0^{2\pi}P\cos\theta\cdot d\theta\cdot dZ$$
$$W_\phi=2\int_0^{0.5}\int_0^{2\pi}P\sin\theta\cdot d\theta\cdot dZ$$
$$W=\sqrt{W_\varepsilon^{\,2}+W_\phi^{\,2}}$$
(16)

The attitude angle, □, is defined as the angle between the line of centers and the load line. Referring to Fig. (2), it is calculated by:

$$\phi=\tan^{-1}\left(\frac{-W_\phi}{W_\varepsilon}\right) \quad (17)$$

Friction force and friction coefficient

The frictional force at journal surface can be given by the dimensionless equation:

$$F=2\int_0^{0.5}\int_0^{2\pi}\left(0.5\frac{\partial P}{\partial\theta}H-0.5H\alpha\nu^2H_m\frac{\partial H_m}{\partial\theta}+\frac{\bar{\lambda}}{H}\right)\cdot d\theta\cdot dZ$$
(18)

The modified friction coefficient is given by:

$$(R/c)f=$$
$$\frac{1}{W}\cdot 2\int_0^{0.5}\int_0^{2\pi}\left(0.5\frac{\partial P}{\partial\theta}H-0.5\alpha\nu^2H_m\frac{\partial H_m}{\partial\theta}+\frac{\bar{\lambda}}{H}\right)\cdot d\theta\cdot dZ$$
(19)

Side leakage

Newtonian lubricant

Non-dimensional form of this equation can be written as:

$$Q=\int_0^{2\pi}-\frac{H^3}{6\,\bar{\lambda}}\cdot\left(\frac{\bar{\lambda}}{4\nu^2}\frac{\partial P}{\partial Z}-\alpha\cdot H_m\frac{\partial H_m}{\partial z}\right)_{z=0.5}\cdot d\theta$$
(20)

Couple stress lubricant

The dimensionless side leakage as in Osman et al [15] is calculated by:

$$Q=\int_0^{2\pi}-\frac{G(H,L^*)}{6\,\bar{\lambda}}\cdot\left(\frac{\bar{\lambda}}{4\nu^2}\frac{\partial P}{\partial Z}-\alpha\cdot H_m\frac{\partial H_m}{\partial z}\right)_{z=0.5}\cdot d\theta$$
(21)

V. Magnetic Field Model

We use a concentric finite wire magnetic field model, Fig. (3). It is the induced magnetic field by a current passing through finite wire located at the shaft center. It produces a magnetic field that is represented by the following non-dimensional form of the equation [1]:

$$H_m(Z)=\sin\tan^{-1}(\nu+2\nu Z)+\sin\tan^{-1}(\nu-2\nu Z)$$
(22)

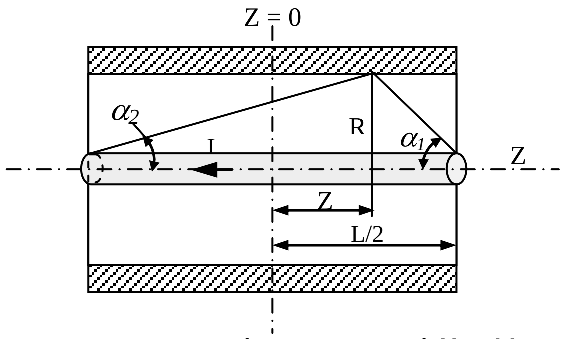

Fig. 3. Concentric finite wire magnetic field model

Bearing Geometry and Boundary Conditions

The examined journal bearing is schematized in Fig. (4). It is an axial feeding cylindrical finite journal bearing. The geometric axes of the journal and bearing are assumed parallel. Reckoning (θ) from the oil admission line, maximum clearance median section, the gap thickness is given by:

$$H = 1 + \varepsilon \cdot \cos\theta \tag{23}$$

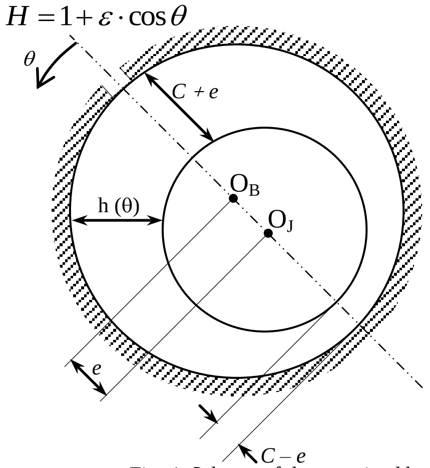

Fig. 4. Scheme of the examined bearing

Considering the boundary conditions, it is clear that the pressure and the temperature are symmetrical about the middle plane of the bearing (Z=0). Thus only half of the bearing has to be calculated.

Used boundary conditions are:

∂P∂Z (θ, 0.0) = 0 Due to bearing symmetry

P (θ, Z) = P (2π, Z) = 0 at the line of oil admission

P (θ, - 0.5) = P (θ, 0.5) = 0 at the bearing ends

And for temperature:

T (0.0, Z) = 0 at the line of oil admission

T (θ, - 0.5) = T (θ, 0.5) = 0 at the bearing ends

VI. Effect of Magnetic Force Coefficient ()

We observe from the result which include an adiabatic solution, when thermal aspects are taken into consideration, that the developed hydrodynamic pressure are lower than those obtained in the isothermal case. This could be related to the introduction of temperature variation in the governing equations. Under isothermal conditions, the lubricant viscosity is taken to be constant whereas in the present adiabatic solution, the viscosity is taken dependent on temperature variation (viscosity decrease exponentially by temperature increase). With lubricant temperature shown an increase during operation, the viscosity is expected to decrease substantially with temperature rise. Hence any decrease in viscosity is associated by decrease in the value of generated hydrodynamic pressure.

It is clear that the peak pressure varies in a manner analogous to the way by which the eccentricity varies. At the far end of the bearing where the eccentricity is assumed to be maximum, the developed hydrodynamic pressure describes maximum values. And at the near end of the bearing where the eccentricity is assumed to be minimum the pressure is found to be minimum. This has been fully observed in the result of the theoretical approach presented.

It is observed from the results that for high eccentricity ratios, the temperature variation along the circumferential direction has a steep gradient in the convergent zone (θ = 0.0 – π) and the gradient flattens in the divergent zone (θ = π – 2π), this could be explained as that in the convergent zone the temperature gradient has been greatly affected by the hydrodynamic pressure gradient and shear action.

In the divergent zone, the negative pressure has been disregarded and assumed to be ambient due to the unknown behavior of cavitated zone. Hence the pressure gradient vanishes and the temperature is assumed to be affected only by the shear action of the possible present oil.

High value of pressure gradient near the divergent zone for the planes ahead to the mid axial plane affected greatly the temperature gradient near this zone (θ = π) by increasing it, while for the planes before the mid axial plane, where lower eccentricity ratios and consequently lower pressure gradient are present there will be lower values of temperature distribution.

The temperature distribution also affected by the inlet condition, i.e. temperature has drop due to the introduction of the oil at ambient conditions through the axial grooving, which the circulating oil has a temperature higher than the supply oil temperature and suddenly drop down at the oil feeding groove.

It is clear that the load capacity is increased slightly with increasing . This is more pronounced at low eccentricity ratios where the hydrodynamic effect is small. At the high eccentricity ratios, the hydrodynamic effect is the major and the magnetic effect can be neglected. The effect of the introduction of the thermal consideration in the analysis of bearings is an expected lower load capacities.

A slightly increase in attitude angle was observed at low eccentricity ratio, but a moderate change have observed at high eccentricity ratios.

There is no significant effect of the magnetic lubrication on the friction force (for the same eccentricity ratio).

A general trend of a lower coefficient of friction is obtained in case of adiabatic solution than that attained under isothermal conditions. Coefficient of friction decreases rapidly for lower values of eccentricity ratio (ε) while retains the slight rate of decrease for higher values of eccentricity ratio.

There is a decrease in side leakage at different eccentricities due to decrease of the attained pressures and then the pressure gradient ($\partial P/\partial Z$) at the end section. So, decrease of the side leakage is obtained.

VII. Effect of Couple Stress Parameter (L*)

In journal bearings lubricated by fluids with couple stress, the couple stress parameter significantly affects the pressure distribution. The results confirm this observation, thus; higher value of non-dimensional pressure comes as results of using a larger couple stress parameter (L*).

Figure (18) is a wire frame map shows the Non-dimensional three dimensional pressure distribution (P) over the bearing half length for couple stress parameter L* = 0.4, length to diameter ratio ν = 1.0 (finite bearing), at low eccentricity ratio ε = 0.1 and with magnetic coefficient α = 0.1

The results show that the couple stress parameter has no significant effect on the temperature distribution at low eccentricity ratios, but have a great effect on the temperature distribution at high eccentricity ratios. As shown in the figure (20) for ε = 0.5; higher value of non-dimensional temperature comes as results of using a larger couple stress parameter (L*).

As a result for using couple stress fluid, the increase of couple stress parameter (L*) leads to increase in both the maximum pressure and load carrying capacity, while decrease the attitude angle and friction coefficient together.

Figure (25) show that effect of couple stresses causes an increase in the value of dimensionless side leakage. This is due to increase of pressure and then the pressure gradient $\partial P/\partial Z$ at Z=0.5 (bearing ends) case increase of side leakage.

Conclusions

1. The thermal aspects affect the lubricant viscosity by reducing it and consequently lower hydrodynamic pressure, reduced load capacity, less frictional force and relatively lower rates of side leakage are obtained in comparison with isothermal results.

2. Under magnetic lubricant, the following conclusions should be considered:

   - The load carrying capacity is increased.
   - Little change of the attitude angle is obtained.
   - The side leakage is highly decreased. It can be completely eliminated by appropriately designing the bearing geometry and magnetic field.

3. For the magnetic lubrication, the increase of the load is not accompanied by increase of the friction losses. Increase of the load without increase of friction force leads to decrease of the friction coefficient. For constant loads, decrease of the operating eccentricity ratio (compared to conventional lubricated bearing) may lead to decrease of the frictional forces.
4. The bearing performance is modified when the magnetic effects are comparable with the hydrodynamic ones, namely; when the bearing operates at low eccentricity ratios (ε) and high values of (α), this requires that the magnetic field to be high, the rotation speed is low and the relative clearance is large. Far from such conditions the hydrodynamic effects prevail considerably and insignificant effect for the magnetic lubrication is obtained.
5. For journal bearing lubricated by magnetic fluids with couple stresses, the following conclusion can be obtained:
   - Both the pressure and load carrying capacity increase with the increase of the couple stress parameter (L*). The increase is more pronounced for bearings operating at higher values of eccentricity ratios.
   - The attitude angle decreases with increasing the couple stress parameter especially at high values of eccentricity ratio.
   - The friction coefficient decreases with increasing couple stress parameter and this influence qualitatively agrees with some previous works.
   - Increasing the side leakage with increasing couple stress parameter.
6. It could be concluded from the above that fluids with microstructure (couple stress) are better lubricants than Newtonian fluids especially if they were prepared to become magnetic fluids. The improvement of the bearing characteristics will cover nearly the whole range of the bearing eccentricity ratios.

Nomenclature

c	bearing radial clearance
C_p	specific heat
D	bearing diameter
e	eccentricity of the journal center
F	dimensionless friction force $F = [F_j (c/R)^2]/\lambda_o \omega Lc$
F_j	friction force at the journal surface
f	friction coefficient $f = F_j/w = (c/R)\, F/w$
f_m	unit volume value of the induced magnetic force
f_{mx}	magnetic force in x direction (circumferential direction)
f_{mz}	magnetic force in z direction (axial direction)
h	lubricant film thickness
H	dimensionless film thickness $H = h/c$
$\mathrm{h_m}$	magnetic field vector
h_m	magnetic field intensity
h_{mo}	characteristic value of magnetic field intensity
H_m	dimensionless magnetic field intensity $H_m = h_m / h_{mo}$
I	strength of the current passing through the wire
J	mechanical equivalent
K	distance ratio parameter $K = R_o / R$
L	bearing length
l	couple stress parameter
L^*	dimensionless couple stress parameter $L^* = l/c$
M_g	magnetization of the ferrofluid
M_{gs}	saturation value of magnetization
p	lubricant pressure

P dimensionless pressure $P = p\,(c/R)^2 / \lambda_o\omega$
q bearing side leakage
Q dimensionless side leakage $Q = 2q / LRc\omega$
R bearing or journal radius
t_o inlet oil temperature
t lubricant temperature
T dimensionless temperature $T = [J\rho Cp(c/R)^2(t-t_o)] / \lambda_o\omega$
U linear velocity of the journal
w load carrying capacity
W dimensionless load carrying capacity $W = [w\,(c/R)^2] / \lambda_o\,\omega\,LR$
$W_□$ dimensionless load capacity component in the eccentricity direction
$W_□$ dimensionless load capacity component in the direction normal to the eccentricity
X_m susceptibility of ferrofluid
Z dimensionless axial distance $Z = z / L$
□ magnetic force coefficient $\alpha = [(h_{mo})^2\ \mu_o\ X_m\ C^2] / \lambda_o\,\omega\,L^2$
□ eccentricity ratio □ $= e / c$
□ attitude angle
λ_o fluid viscosity at inlet temperature t_o
λ fluid viscosity $\lambda = \lambda_o\, e^{-\beta(t-t_o)}$
$\bar{\lambda}$ dimensionless viscosity $\bar{\lambda} = e^{-\zeta T}$
□ angular coordinate □□$= x / R$
$□_o$ permeability of free space or air $□_o = 4□.10^{-7}$ AT/m
ζ oil's property $\zeta = [\beta\,\lambda_o\,N] / J\rho Cp(c/R)^2$
β viscosity temperature index $\beta = 4.91\times10^{-2}\ {}^oC^{-1}$
□ shear stress, dimensionless time
$□_x$ shear stress in circumferential direction

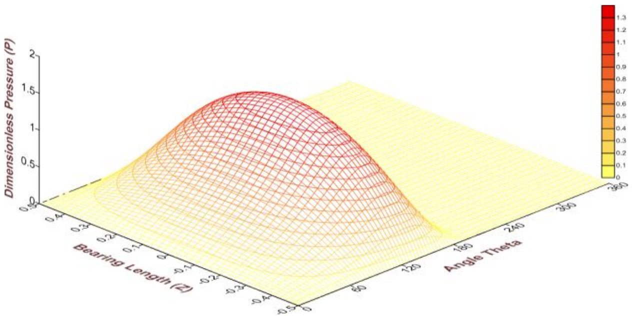

Fig. 5. Non-dimensional 3D pressure distribution (P) at ε = 0.5, α = 0 and ν = 1.0 taking thermal effect and viscosity variation into consideration.

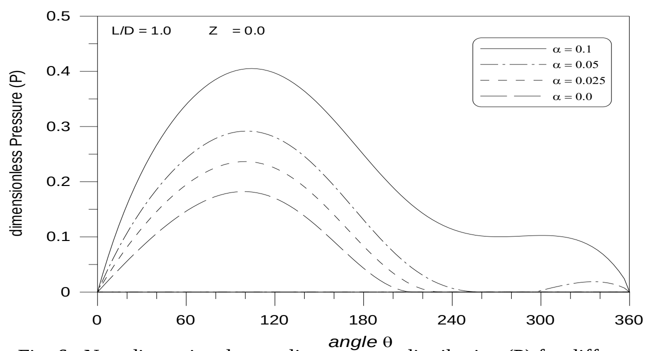

Fig. 6. Non-dimensional centerline pressure distribution (P) for differentα, eccentricity ratio (ε =0.1) and ν = 1.0

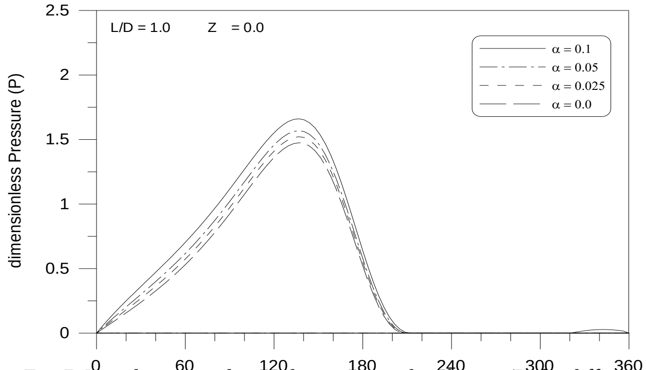

Fig. 7. Non-dimensional centerline pressure distribution (P) for different (α), eccentricity ratio (ε =0.5) and ν = 1.0

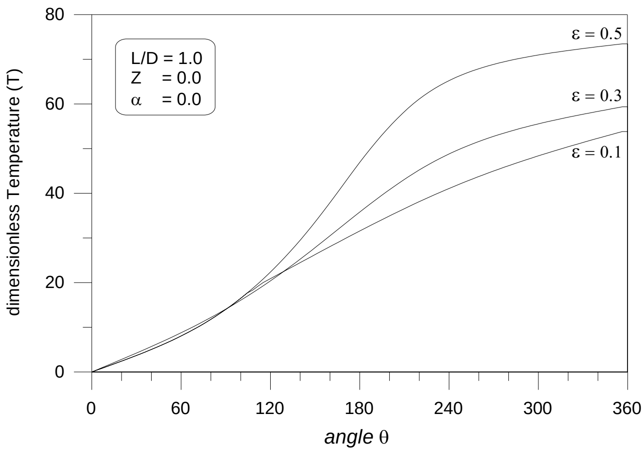

Fig. 8. Non-dimensional Temperature distribution (T) for different (ε), α = 0 and ν = 1.0 at bearing end Z = 0.

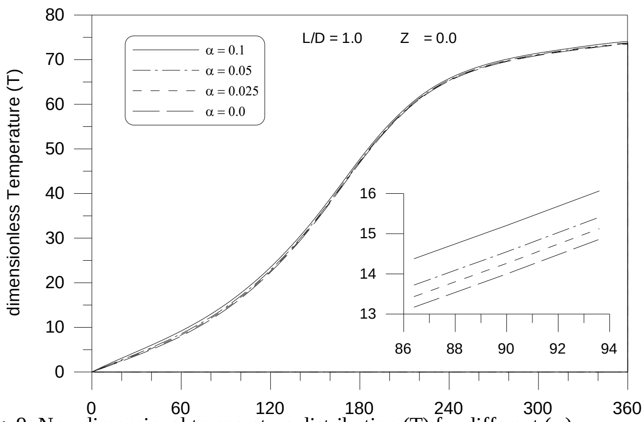

Fig. 9. Non-dimensional temperature distribution (T) for different (α), eccentricity ratio (ε =0.5) and ν = 1.0

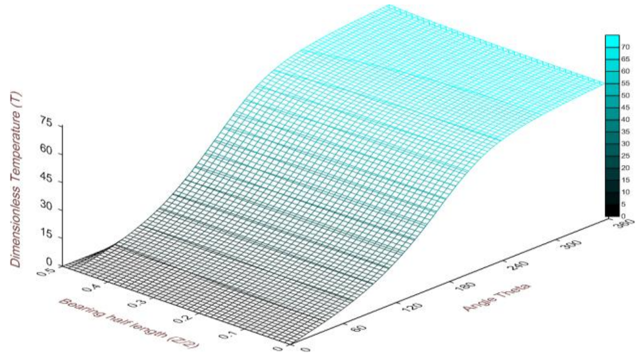

Fig. 10. Non-dimensional 3D Temperature distribution (T) at ε = 0.5, α = 0 and ν = 1.0 over the half length of the bearing.

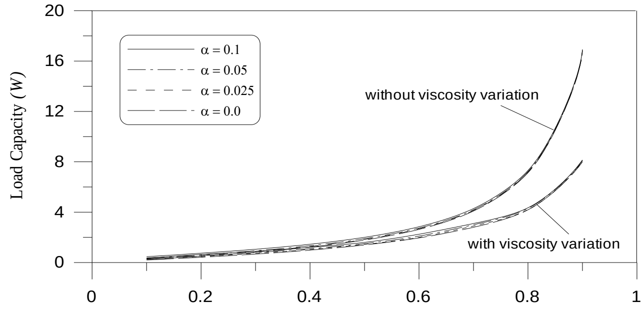

Fig. 11. Non-dimensional Load capacity (W) for different magnetic force coefficient (α), with & without thermal effect.

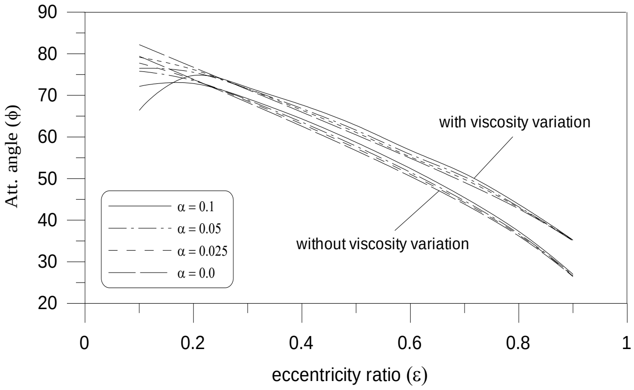

Fig. 12. Attitude angle (ϕ) versus (ε) for different magnetic force coefficient (α), with & without thermal effect

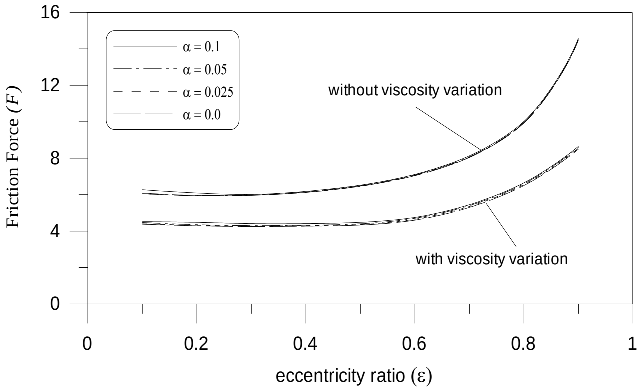

Fig. 13. Non-dimensional friction force (F) versus ε for different α, with & without thermal effect and ν = 1.0

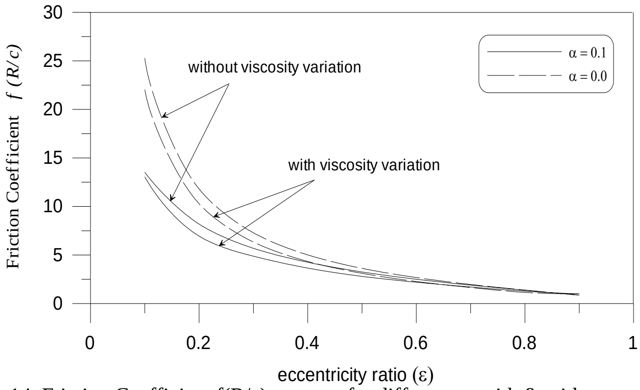

Fig. 14. Friction Coefficient f(R/c) versus ε for different α, with & without thermal effect and ν = 1.0

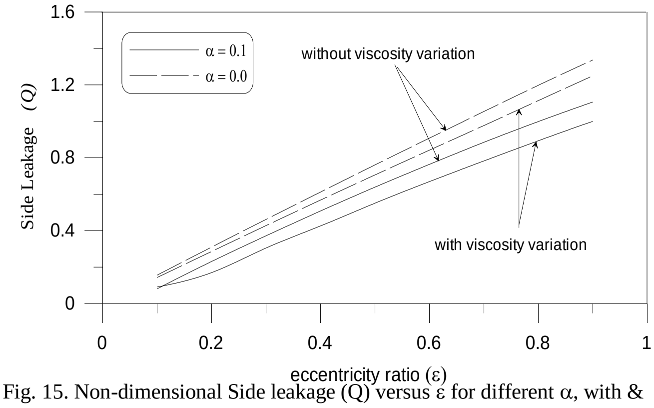

Fig. 15. Non-dimensional Side leakage (Q) versus ε for different α, with & without thermal effect and ν = 1.0

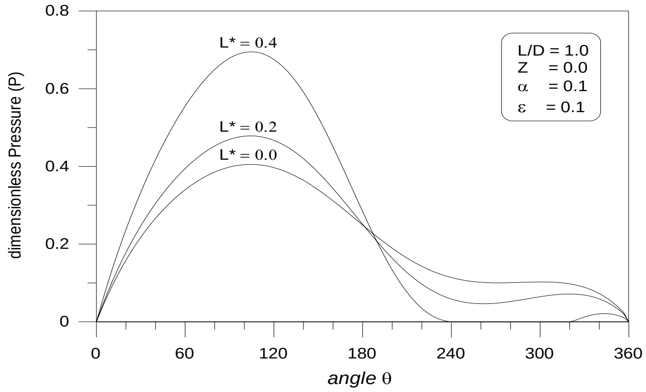

Fig. 16. Non-dimensional centerline pressure (P) for different couple stress parameter L*, ν = 1.0, ε = 0.1 and α = 0.1.

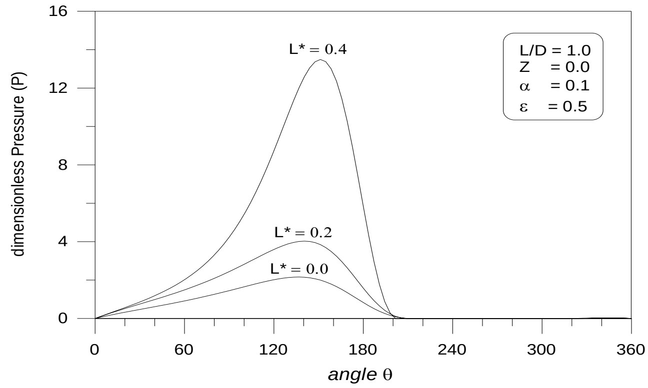

Fig. 17. Non-dimensional centerline pressure (P) for different couple stress parameter L*, ν = 1.0, ε = 0.5 and α = 0.1.

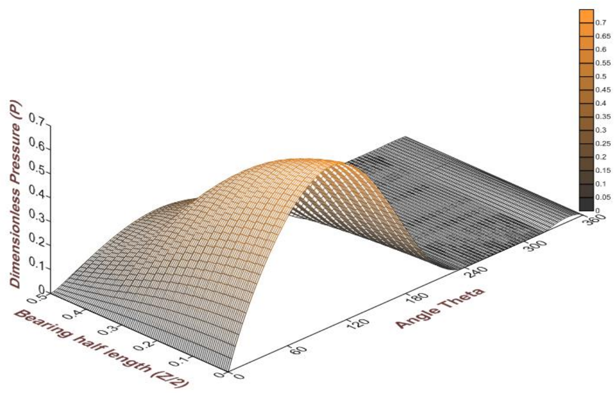

Fig. 18 Non-dimensional 3D pressure distribution (P) for couple stress parameter L* = 0.4, ν = 1.0, ε = 0.1 and α = 0.1.

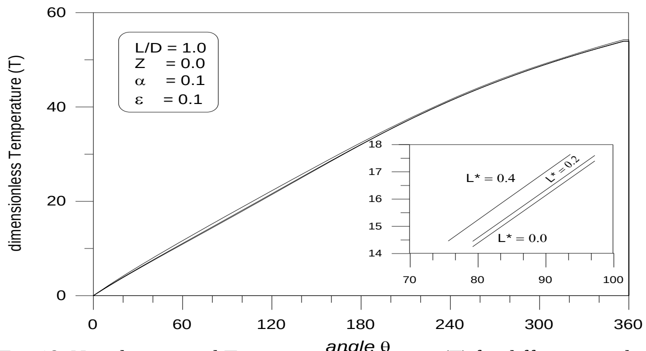

Fig. 19. Non-dimensional Temperature distribution (T) for different couple stress parameter L*, ν =1.0, ε =0.1 and α =0.1

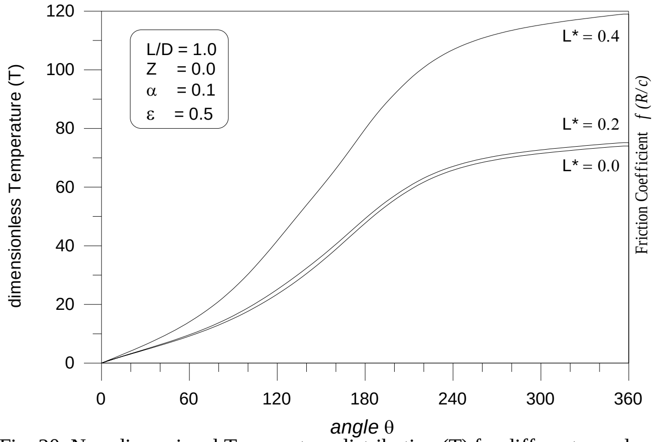

Fig. 20. Non-dimensional Temperature distribution (T) for different couple stress parameter L*,ν =1.0, ε =0.5 and α =0.1.

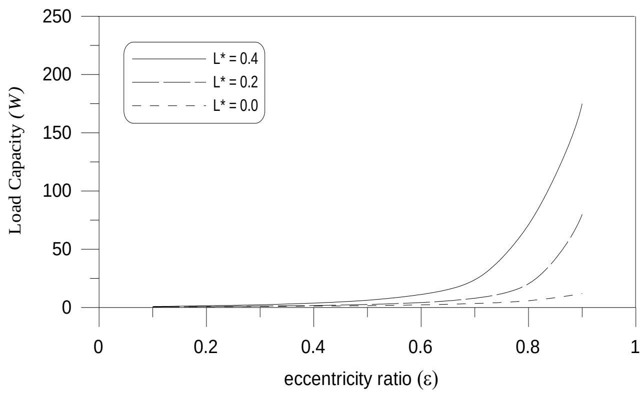

Fig. 21. Non-dimensional Load capacity (W) versus ε for different (L*) and magnetic force coefficient (α = 0.1).

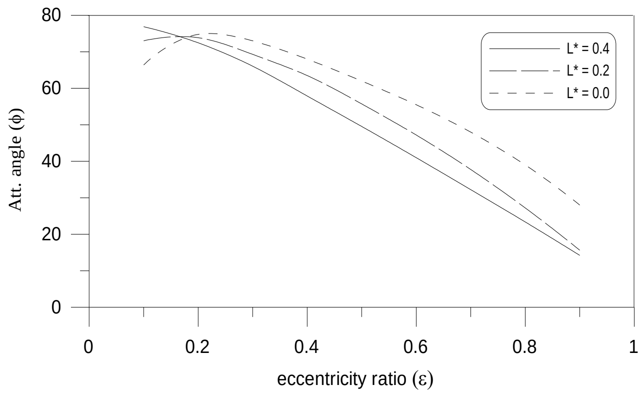

Fig. 22. Attitude angle (ϕ) versus ε for different (L*) and magnetic force coefficient (α = 0.1).

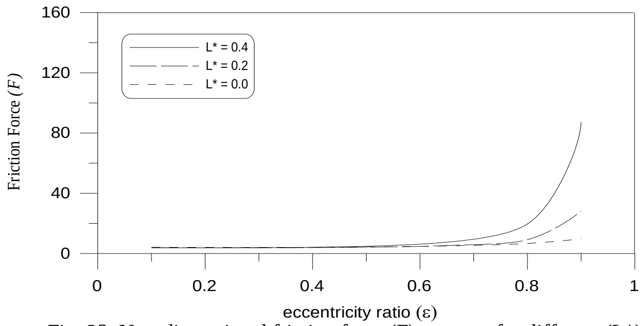

Fig. 23. Non-dimensional friction force (F) versus ε for different (L*) and magnetic force coefficient (α = 0.1).

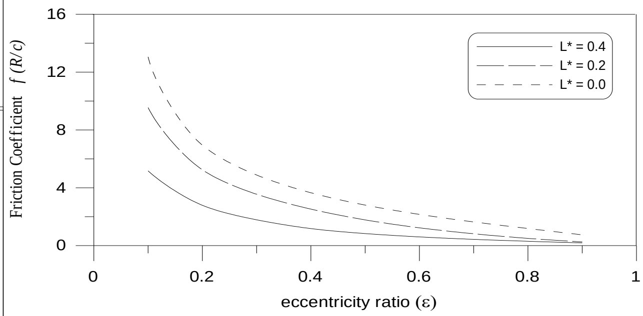

Fig. 24. Friction Coefficient f(R/c) versus ε for different (L*) and magnetic force coefficient (α = 0.1).

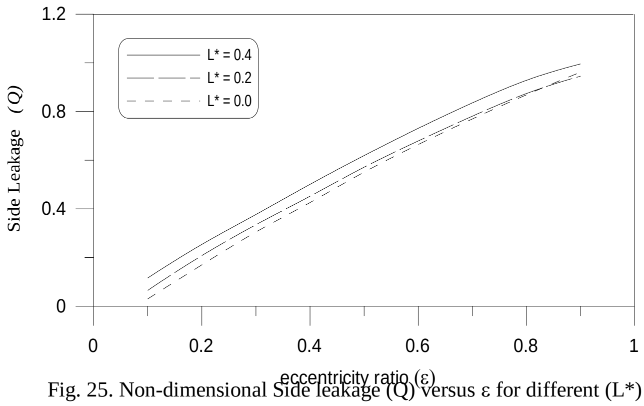

Fig. 25. Non-dimensional Side leakage (Q) versus ε for different (L*) and magnetic force coefficient (α = 0.1).

REFERENCES

[1] G.S. Nada et al "Static and Dynamic Characteristics of Magnetized Journal Bearings Lubricated with Ferrofluid" PhD Thesis in Mechanical Engineering, Faculty of Engineering, Cairo University, 2002.

[2] Saynatjoki, M. and Holmberg, K. "Magnetic Fluids in Sealing and Lubrication –A State of Art Review" Technical Research Center of Finland, February 1982, Vol. 10, P. 119-131.

[3] Moskowitz, R., and Ezekiel, F. D. "Magnetic fluids-something to consider" Instruments & Control system, October 1975, Vol. 48, P. 41-45.

[4] Williams, R. A. and Malsky, H. "Some Experiences Using a Ferrofluid Seals Against a Liquid" IEEE Transactions on Magnetics, March 1980, Vol. Mag.-16, P. 379-381.

[5] Perkovskii, B. M., Karkov, M. S. and Rakhuba, V. K. "Design Problems and Use Limitations for Magnetic-Fluid Seals" Translated from Magnitnaya Gidrodinamika, January 1982, P. 85-93.

[6] Bonvouloir, J. "Experimental Study of High Speed Sealing Capability of Single Stage Ferrofluidic Seals" ASME Journal of Tribology, July 1997, Vol. 119, P. 416-421.

[7] Gorla, J. S. R., Ramalingam, K. and Adluri, I. "Magnetohydrodynamic Breaking" ASME Journal of Tribology, October 1995, Vol. 117, P. 724-728.

[8] Ferrofluid study group, BIAA "On the Magnetic Sealing Capability of Ferrofluid-Lubricated Journal Bearings" Proc. R. Soc. Lond., 1986, Series A, Vol. 404, P. 69-88.
[9] Sorge, F. "A Numerical Approach to Finite Journal bearings Lubricated With Ferrofluid" ASME Journal of Tribology, January 1987, Vol. 109, P. 77-82.
[10] Chang, H. S., Chi, C. Q. and Zhao, P.Z. "A Theoretical and Experimental Study of Ferrofluid Lubricated Four-Pocket Journal Bearings" Journal of Magnetism and Magnetic Materials, 1987, Vol. 65, P. 372-374.
[11] Monira, A. M. "The Effect of Thermal Aspects on the Performance of Hydrodynamically Lubricated Journal Bearings" Master Thesis in Mechanical Engineering, Faculty of Engineering, Cairo University, 1982, No. 2934.
[12] Cowley, M. D. and Rosensweig, R. E. "The Interfacial Stability of Ferromagnetic Fluid" Journal of Fluid Mech., April 1967, Vol. 30, P. 671-688.
[13] Zelazo, R. E. and Melcher, J. R. "Dynamic and Stability of Ferrofluids: Surface Interaction" Journal of Fluid Mech., February 1969, Vol. 39, P. 1-24.
[14] Pinkus, O. and Sternlicht, B. "Theory of Hydrodynamic Lubrication" McGraw-Hill, N.Y., 1961.
[15] G.S. Nada and T.A. Osman "Static Performance of Finite hydrodynamic journal bearings lubricated by Magnetic fluids with Couple Stresses" Tribology Letters Vol. 27, 2007, PP 261 – 268.
[16] T.A. Osman, G.S. Nada and Z.S. Safar "Effect of using current-carrying-wire models in the design of hydrodynamic journal bearings lubricated with ferrofluid" Tribology Letters Vol. 11, No.1, 2001, PP 61-70.
[17] Osman, T. A. "Static Characteristics of Hydrodynamic Magnetic Bearings Working By Non-Newtonian Ferrofluid" Journal of Engineering And Applied Science , Faculty of Engineering, Cairo University, June 1991, Vol. 46, No. 3, P. 521-536.
[18] U.M. Mokhiamer et al. "A study of a journal bearing lubricated by fluids with couple stress considering the elasticity of the liner" Elsevier Science S.A., Wear 224,1999, PP 194–201