Proceedings of ICFDP9:
Ninth International Congress of Fluid Dynamics & Propulsion
December 18-21, 2008, Alexandria, Egypt

ICFDP9-EG-273

Characteristics of Magnetized Turbulent Flow Journal bearings Lubricated with Ferrofluid

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ABSTRACT
This paper is concerned with the characteristics of hydrodynamic turbulent flow journal bearing lubricated with Ferro fluid. Based on the momentum and continuity equation and using the mixing length concept to express turbulent stresses a pressure differential equation (modified Reynolds equation) was obtained. Assuming linear behavior for the magnetic material of the Ferro fluid, the magnetic force was calculated and the magnetic pressure resulting from it was incorporated in the modified Reynolds equation. Solutions by finite difference methods were obtained and rendered the bearing performance characteristics. It was found that magnetic effect is more pronounced for laminar flow bearings at lower values of eccentricity ratio than that of turbulent flow bearings

KEYWORDS:
Journal Bearing, Ferrofluid, Turbulent Flow, Magnetic field

INTRODUCTION
Ferrofluids are an interesting group of liquids, because they have liquid properties and act like a ferromagnetic material. Many properties of the Ferrofluid are similar to the base fluid. As regards preparation, characteristics and applications of Ferrofluids, many references are available. Among the various applications in engineering, are those taking the advantage of introducing an external force into the fluid to facilitate control, position, orientation and movement of the fluid. The most usual applications are in sealing, filtering, separation, ink-jet printing and heat transfer[1-3]. Magnetic fluids have also been used in the lubrication of journal bearings with some advantages over conventional lubrication.

Only a few and incidental references have been made to the field of lubrication. Analytical investigations have already been executed [4, 5]. Osman and et al. [6] were concerned with the static and dynamic characteristics of the hydrodynamic journal bearings lubricated with Ferro fluid. Based on the momentum and continuity equations, a pressure differential equation (modified Reynolds equation) was obtained. Assuming linear behavior for the magnetic material of the Ferro fluid, the magnetic force was calculated. The magnetic pressure resulting from the magnetic force was incorporated into the Reynolds equation and it was not separately treated. The solution rendered the bearing performance characteristics, namely, load carrying capacity, attitude angle, frictional force at the journal surface, friction coefficient and bearing side leakage.

Throughout the last few decades, there has been a great deal of interest both from an analytical and experimental view point in the operation of bearings beyond the laminar regime. An approximation method was first introduced by Smith and Fuller [7] who considered that the velocity profiles in the fluid film may be regarded as super-position of Couette and Poiseuille flows. Tao [8] and Golubev [9] suggested another simplified method for short bearings. They assumed that the circumferential pressure gradient, is negligible in comparison with the pressure gradient and the latter can be estimated by analogy with flows through ducts.

At present the most common methods used to describe turbulent stresses are basically of two types. One method is derived from mixing length theories and the other is derived from logarithmic profiles of fully developed turbulent pipe flows. Constantinescu [10, 11] assumed a linear relationship of the mixing length $\pounds_M$. Chou and Saibel [12] considered a parabolic law for the mixing length. Constantinescu [13]showed that whether taking linear, parabolic, etc. law of variation of the mixing length, it has no qualitative influence on the shape of the velocity profiles. This was confirmed recently by Z Zhang et al. [14]. Pressure distributions were obtained for infinitely long bearing. When the variations of the attitude angle with Reynolds number were plotted, good agreements with the experimental measurements were obtained when using Reynolds boundary conditions;

$$\{P = P_a \text{ at } \theta = \theta_1,\ P = P_a \text{ and } \frac{\partial P}{\partial \theta} = 0 \text{ at } \theta = \theta_2\}$$

MODIFIED REYNOLDS NUMBER

For a Ferrofluid under a magnetic field, the unit volume value of the induced magnetic force is given by

$$f_m = \mu_o X_m h_m \operatorname{grad} h_m$$

Consistent with the basic assumption of creeping flow and the type of geometry which is particular to lubricating films i.e., the inertia forces and the derivations of velocity components with the velocity derivatives in the direction and making use of the assumption that turbulence posses special homogeneity surfaces, and using the magnetic force as a body external force, the equation of motion for the fluid film are derived as follow

$$0 = -\frac{\partial P}{\partial x} + \frac{\partial}{\partial y}\left\{\eta \frac{\partial u}{\partial y} - \overline{\rho u`v`}\right\} + f_{mx} \quad (1)$$

$$0 = -\frac{\partial P}{\partial z} + \frac{\partial}{\partial y}\left\{\eta \frac{\partial w}{\partial y} - \overline{\rho v`w`}\right\} + f_{mz} \quad (٢)$$

Using the mixing length concept to express the turbulent flow stresses, and assuming a linear variation of the mixing length as was adopted by Constantinescu, and assuming axial symmetric applied magnetic field, the modified Reynolds equation for turbulent flow of Ferrofluid becomes

$$\frac{\partial P}{\partial x}\left\{\frac{h^3}{\eta K_x}\rho\frac{\partial P}{\partial x}\right\} + \frac{\partial}{\partial z}\left\{\frac{h^3}{\eta K_z}\rho\frac{\partial P}{\partial z}\right\} = \frac{U}{2}\frac{\partial(\rho h)}{\partial x} + \frac{\partial}{\partial z}\left\{h^3 h_m \frac{\partial h_m}{\partial z}\right\}$$

$$\frac{\partial P}{\partial Z} \quad (3)$$

Where

K_x and K_z are constants and given by Constantinescu as

$$K_x = 12 + 0.53\,(R_e K^2)^{0.725}$$
$$K_z = 12 + 0.296\,(R_e K^2)^{0.65} \quad (4)$$

Constantinescu assumed that K is equal to 0.3

Using the non dimensional parameters, the modified turbulent Reynolds equation becomes

$$\frac{\partial}{\partial \theta}\left\{\frac{H^3}{K_x}\frac{\partial \overline{P}}{\partial \theta}\right\} + \frac{1}{4v^2}\frac{\partial}{\partial Z}\left\{\frac{H^3}{K_z}\frac{\partial \overline{P}}{\partial Z}\right\} = 6\frac{\partial H}{\partial \theta} + \alpha\frac{\partial}{\partial Z}\left\{H^3 H_m \frac{\partial H_m}{\partial Z}\right\} \quad (5)$$

The RHS of this equation contains the wedge action effect, and the magnetic effect to field gradient in the axial direction.

BEARING SCHEME

The examined journal bearing is schematized in figure (1), it is an axial feeding cylindrical finite journal bearing. The geometric axes of the journal and bearing are assumed parallel. Then, the non – dimensional film thickness is given by

$$H = 1 + \varepsilon \cos \theta \quad (٦)$$

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

The boundary conditions used are:

$$\overline{P}(\theta, 0) = 0,\ \overline{P}(\theta, 0.5) = 0,\ \overline{P}(0, Z) = \overline{P}(2\pi, Z) = (\quad (٧)$$

$0 \quad 0$

At film rupture and reformation boundaries, free Reynolds boundary conditions are used:

$$\bar{P}(\theta_2) = \left.\frac{\partial \bar{P}}{\partial \theta}\right|_{\theta=\theta_2} = 0 \qquad (٨)$$

θ_2 is the film rupture or reformation angle; it is not a prescribed value but it is determined during calculations.

MAGNETIC FIELD MODEL

The axial parabolic distribution magnetic field is used. It is represented by the following equation

$$h_m(z) = h_{mc} - (h_{mc} - h_{me})(2z/L)^2 \qquad (٩)$$

In non dimensional form, it is given by

$$H_m(z) = 1.0 - 4(1-ß)Z^2 \qquad (١٠)$$

Where $H_m = h_m / h_{mc}$ and $Z = z/L$ (١١)

ß are the ratio of the magnetic field strength at end section (h_{me}) to its value at the middle section (h_{mc}). It is an important parameter that determines the gradient of the magnetic field .Negative magnetic gradient is required to obtain positive induced magnetic pressures and the resultant load carrying capacity will then be increased. This can be achieved for values of ß ranging from 0 to less than 1. If ß =1.0, there is no field variation ($\partial H_m/\partial Z$ = 0.0) and Equation (5) will be turned into the Reynolds equation for turbulent flow hydrodynamic lubrication and the bearing will then perform as a conventional bearing .On the other hand, if the magnetic field variation is such that $\partial H_m/\partial Z$ is positive (for ß>1.0), negative magnetic pressures will be induced and the bearing performance is then decreased.

BEARING PERFORMANCE CHARACTERISTICS

Integration of the pressure over the bearing area gives the non –dimensional load carrying capacity, calculated by

$$W = \sqrt{W_\varepsilon^2 + W_ø^2} \qquad (١٢)$$

$$W_\varepsilon = 2\int_0^{0.5}\int_0^{2\pi} \bar{P}\cos\theta \, d\theta \, dZ \qquad (١٣)$$

$$W_ø = 2\int_0^{0.5}\int_0^{2\pi} \bar{P}\sin\theta \, d\theta \, dZ \qquad (١٤)$$

The Sommerfeld number, S, can be determined as

$$S = \frac{1}{\pi W} \qquad (١٥)$$

The attitude angle, ø, is calculated by

$$ø = \tan^{-1}\left\{\frac{W_ø}{W_\varepsilon}\right\} \qquad (١٦)$$

Non-dimensional frictional force at the journal surface can be given by

$$\bar{F} = 2\int_0^{0.5}\int_0^{2\pi} \left\{ 0.5 \frac{\partial \bar{P}}{\partial \theta} H + \frac{b}{H} \right\} d\theta \, dZ \qquad (١٧)$$

Where F={(c ... c}F ... and b is a control number. Inside the active zone region (full film thickness), b=1.0 and outside this region (partial film thickness), it is calculated by b = h_{min} / h< 1.0; where h_{min} is the minimum film thickness.

The side leakage can be obtained by integrating the axial velocity component v_z across the end section. In general, it is calculated by

$$Q = \int_0^{2\pi} -\frac{H^3}{6}\left\{\frac{1}{4\nu^2}\frac{\partial \bar{P}}{\partial Z} - \alpha \frac{\partial H_m}{\partial Z}\right\}_{Z=0.5} d\theta \qquad (١٨)$$

ANALYSIS

The performance characteristics of a magnetized turbulent flow bearing, with length to diameter ratio ν=1.0 and magnetic parameter β=0.75, are determined and compared to those of a conventional lubricated bearing, i.e. a bearing which has zero magnetic force coefficient α.

The effect of this field model on the pressure distribution for laminar and turbulent flow bearing are illustrated in Figures (2) and (3) respectively, for eccentricity ratio=0.2 .A small eccentricity ration was chose)14(Osman et al.[6] has shown that the influence of the magnetic force coefficient on the bearing performance characteristic is significant at lower values of eccentricity ratios. Figure (2) shows that the peak value of the pressure is almost twice when α is increased from 0 to 0.2, while for turbulent flow bearings of Reynolds number of 5000, the peak pressure is only 1.5 times for the same range when α is increased .

Fig.(4) shows the influence of magnetic force coefficient (α) on non-dimensional load capacity for different eccentricity ratio values(ε) for laminar flow bearing. As shown, at low eccentricity ratio, there is a considerable increase in the load as α is increased. This improvement of the load is decreased with the increase of ε until it becomes negligible at higher eccentricity ratios. From a conventional laminar flow bearing to a magnetic bearing (i.e. α=0.0 to α=0.2), the load carrying capacity increases about 60% at ε=0.2, this value is decreased to 20% at ε=0.3, and becomes 6% at ε=0.6.

The same phenomena is observed for turbulent flow journal bearing at Reynolds number=5000 as shown in Figure (5), but the magnetic effect is less pronounced, thus from a conventional bearing to a magnetic turbulent flow bearing the load carrying capacity increases by only 25% at ε=0.2 .

Figures (6) and (7) show the effect of magnetic force coefficient (α) on non-dimensional frictional force. As shown, there is no magnetic effect on the friction force for both laminar and turbulent flow bearings.

The increase in pressure, by the magnetic effect, may not have any significant effect on the pressure gradient ($\partial\overline{P}/\partial\theta$) and the friction force is nearly constant, this means that the magnetic lubrication gives higher load carrying capacity without increases in the friction force and the power losses are not affected.

Fig.(8) and (9) shows the effect of magnetic force coefficient (α) on non-dimensional side leakage. As shown there is a considerable decrease in the side leakage with the increase of the coefficient α, for laminar and turbulent flow respectively but the effect is more pronounced for the laminar flow bearing.

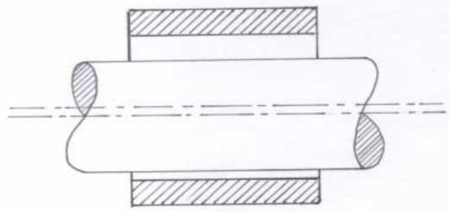

Fig. (1) Schematic diagram of bearing configuration . (Conventional bearing)

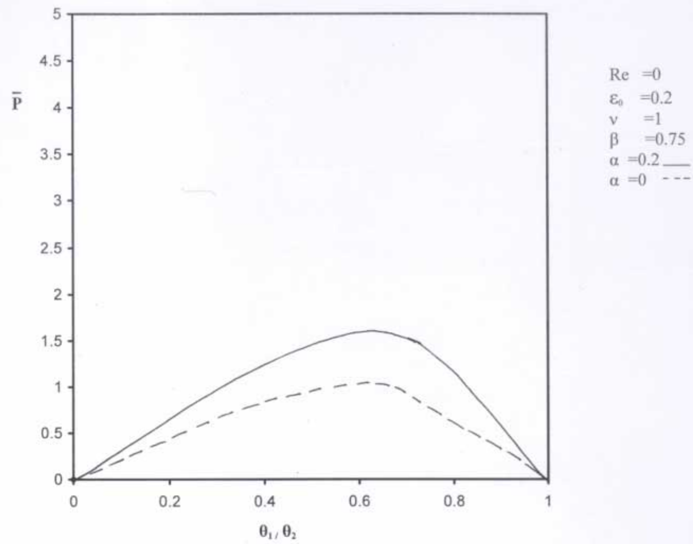

Fig. (2) Non –dimensional pressure distribution in the circumferential direction for different magnetic force coefficient ε_0=0.2 , ν=1
(Laminar flow bearing)

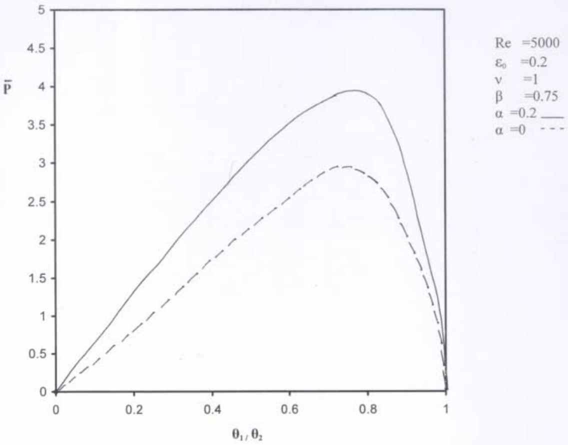

Fig. (3) Non –dimensional pressure distribution in the circumferential direction for different magnetic force coefficient ε_0=0.2 , ν=1 (Turbulent flow bearing)

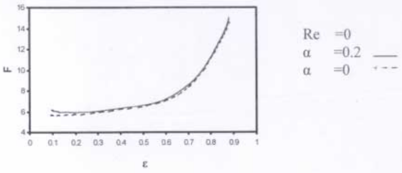

Fig. (6) Non –dimensional friction force (F) versus eccentricity ratio (ε) and magnetic force coefficient (α), ν=1 and β=0.75

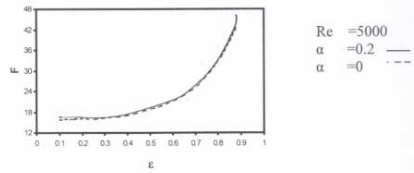

Fig. (7) Non –dimensional friction force (F) versus eccentricity ratio (ε) and magnetic force coefficient (α), ν=1 and β=0.75

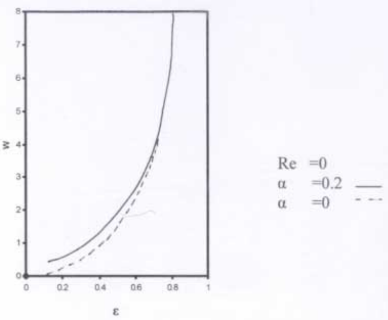

Fig. (4) Non –dimensional load capacity (W) versus ε and α , ν=1 and β=0.75

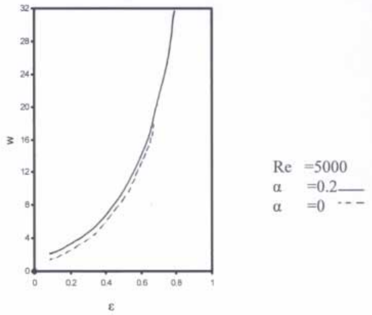

Fig. (5) Non –dimensional load capacity (W) versus ε and α , ν=1 and β=0.75

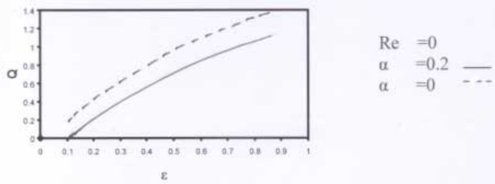

Fig. (8) Non –dimensional side leakage (Q), for the two bearing configurations versus eccentricity ratio (ε) and magnetic force coefficient (α), ν=1 and β=0.75 (Unsealed bearing)

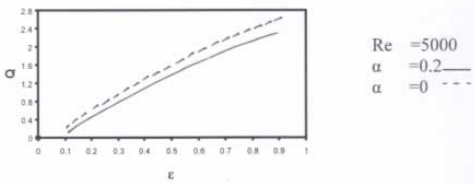

Fig. (9) Non –dimensional side leakage (Q), for the two bearing configurations versus eccentricity ratio (ε) and magnetic force coefficient (α), ν=1 and β=0.75 (Unsealed bearing)

CONCLUSIONS

The bearing performance characteristics for laminar flow bearings are modified when the magnetic effects are comparable with the hydro-dynamic effects, i.e., when the bearing operating under higher magnetic field at lower eccentricity ratio. For turbulent flow bearing with higher rotational speeds the magnetic effects become less pronounced and the increase in load carrying capacity from conventional to magnetic turbulent flow bearing is far less than that obtained in case of laminar flow bearing, the benefit of the magnetic force is only noticed in the side leakage, where the side leakage is highly decreased for both laminar and turbulent flow bearing. It can be completely eliminated by appropriately designing the bearing geometry and the magnetic field.

NOMENCLATURE

c bearing clearance

D bearing diameter

e eccentricity of the journal center

F friction force at the journal surface

$\bar{F}$ dimensionless friction force $\bar{F} = \dfrac{F\,(c/R)^2}{\eta\,\omega LC}$

f_m unit volume value of the induced magnetic force

h lubricant film thickness

H dimensionless film thickness H=h/c

h_m magnetic field intensity

h_{m0} characteristic value of magnetic field intensity

H_m dimensionless of magnetic field intensity $H_m = h_m/h_0$

L bearing length

m_g magnetization of the Ferro fluid

$\bar{P}$ dimensionless pressure $\bar{P} = \dfrac{P\,(c/R)^2}{\eta\,\omega}$

p lubricant pressure

Q dimensionless side leakage $Q = \dfrac{2q}{LRc\omega}$

q bearing side leakage

R bearing or journal radius

R_e Reynolds number

U velocity of the rotor

u, v, w Fluid velocity in the x, y, z directions

u`, v`, w` Perturbation of the fluid velocity in the x, y, z directions

W dimensionless load-carrying capacity $W = \dfrac{w\,(c/R)^2}{\eta\omega LR}$

w load-carrying capacity

$W\varepsilon$ dimensionless load capacity component in the eccentricity direction

$W_{\o}$ dimensionless load capacity component in the direction normal to the eccentricity

Xm susceptibility of Ferro fluid

χ, y, z Cartesian coordinates

Z dimensionless axial distance Z = z/L

α magnetic force coefficient $\alpha = \dfrac{(h_{m0})^2\,\mu_0\,{}^x\,mg^2}{\eta\omega\,L^2}$

ε eccentricity ratio $\varepsilon = e/c$

ø attitude angle

η fluid viscosity

θ angular coordinate $\theta = \chi / R$

μ_0 permeability of free space or air $\mu_0 = 4\pi \times 10^{-7}$ AT/m

ω angular speed

ν length to diameter ratio L/D

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