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Three-Dimensional Thermohydrodynamic Analysis of Journal Bearing Lubricated by Magnetic Fluids with Couple Stresses

Ruaa Braz Dahham[1], Noora A Hashim[2,*], Fatima Mohammed K Al-Fatlwe[3]

1. Ministry of Education – Najaf Education Directorate – School Building Division – Najaf – Iraq 2. University of Kufa – Mechanical Engineering Department – Najaf – Iraq 3. Al-Furat Al-Awsat Technical University – Engineering Technical College – Najaf – Iraq
*Correspondence author: nooraa.alkhalidi@uokufa.edu.iq

ABSTRACT

Couple stresses because of magnetic particles additives on the lubrication performance of a journal bearing system was embraced and scrutinized in the current examinations theoretically. The governing equations of Reynolds (pressure field), energy (temperature field), and heat conduction (temperature field through the solids) are coupled and solved instantaneously in term of temperature and viscosity. After these set of equation pressure is obtained, it is used to obtain the bearing characteristic. It was found that the load currying capacity and maximum pressure are increased due the employing of magnetic fluids along with the couple stresses, as well as side leakage flow and the friction coefficient are decreased. It can be also concluded that the magnetic fluids with couple stresses are better lubricant than magnetic fluid only ($\bar{l} = 0$), Newtonian fluids, and couple stress fluid ($\lambda = 0$).

Keywords: Coupled stress, Magnetic fluids, load currying capacity.

INTRODUCTION

Hydrodynamic bearings have been used in various applications of mechanical industry since a long time to support rotating shafts of heavy machines. Studies had been conducted to improve the performance of the bearing. Magnetic force thrust bearing lubrication reduces the surface stresses, and it is possible to avoid the direct contacting of mating surfaces for sufficiently strong field (Burcan *et al.* 2004).

Urreta *et al.* (2009) analyzed two different fluids: one Ferro fluid and one magnetorheological fluid. Magnetorheological fluid achieved good performance as active fluids in bearing. Some researchers (Andharia *et al.* 2019; Lin 2016; Patel *et al.* 2015; Shukla *et al.* 2013) have used magnetic fluid as a lubricant in order to improve the tribological performance of a rough bearing with different conditions but they studied the hydrodynamic performance of the fluid only without thermal effect. The polymer additives were embraced in a magnetic fluid to serve as a lubricant by Hu and Xu (2017) under couple stress condition. The reported outcomes show that the (Pmax) which represents the maximum pressure is getting increase as both the magnetic force coefficient and the parameter of couple stress increase along the bearing centerline. Hu *et al.* (2022) developed a theoretical computation model based on magnetic fluids supporting force parameter and couple-stress factor but ignoring the thermal

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effect. Considering elastohydrodynamic effect of bearing liner, the results indicated that maximum pressure of oil increases with increase in both magnetic fluids parameter, and couple stress factor. It was also concluded that maximum pressure of oil increases with the decrease in elastic deformation coefficient of the liner. Singh and Ahmad (2011) and Yin *et al.* (2013) studied bearing lubricated with magnetic fluid considering thermal effects. Singh *et al.* (2011) used porous-inclined slider bearing and solved energy equation in two dimensions only. Yin *et al.* (2013) also ignored the third dimension in solution of heat field and a lot of factors were out of consideration.

The bearing load capacity relies on lubricant thermophysical properties, especially its temperature. As the oil temperatures vary, the load currying capacity would vary as well in return. Hence, the prediction of bearing performance is strongly affected by the lubricant's thermal status. The main objective of this work is to establish a complete three-dimension thermohydrodynamic analysis for journal bearing undergoing the combined effect of magnetic force and couple stresses behavior of the lubricant. The effect of the temperature difference of the solid part of the bearing (bush) on the performance characteristics of the bearing lubricant is also taken into consideration.

THEORETICAL ANALYSIS

The global coordinates system that is used in the oil film of the journal bearings is the Cartesian coordinates system (x, y, z). The (y) coordinate matches the line centers, the (x) coordinate is in the rotation direction, and the (z) coordinate is perpendicular to the (x, y) plane as shown in Fig. 1. The nondimensional coordinates $(\theta, \overline{y}, \overline{Z})$ are used to solve the governing equations. The $(\theta, \overline{r})$ coordinates are used to describe the temperature distribution through the solid body of bearing.

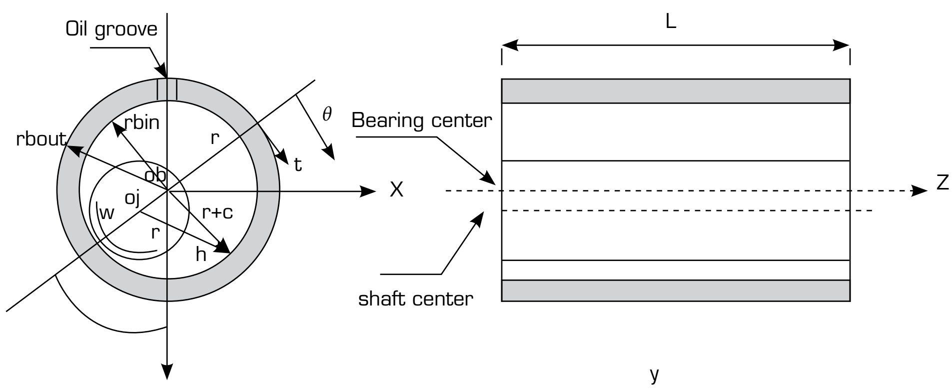

Figure 1. Geometry of journal bearing and coordinate system.

For incompressible flow, Stokes equation (Stokes 1966), which addresses both magnetic fluid with couple stresses can be written as shown in Eqs. 1 and 2:

$$\rho\frac{DV}{Dt} = -\nabla P + F_m + \frac{1}{2}\rho\nabla\times C + \mu_{av}\nabla^2 V - \eta\nabla^4 V \tag{1}$$

$$\nabla\cdot V = 0 \tag{2}$$

Equation 2, which is called *continuity equation* is usually used in such application. The term on the left side of Eq. 1 represents inertia effects, and those on the right side are the pressure gradient, magnetic effects, body couples and viscous terms. It is assumed that the fluid film is thin, the inertia effects and body couples are absent. Then for steady state the fluid equations governing the motion of lubricant can be written in Eqs. 3–6:

$$\frac{\partial p}{\partial x} = F_{mx} + \mu_{av}\frac{\partial^2 u}{\partial y^2} - \eta\frac{\partial^4 u}{\partial y^4} \tag{3}$$

$$\frac{\partial p}{\partial y} = 0 \tag{4}$$

$$\frac{\partial p}{\partial z} = F_{mz} + \mu_{av}\frac{\partial^2 w}{\partial y^2} - \eta\frac{\partial^4 w}{\partial y^4} \tag{5}$$

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0 \tag{6}$$

Where (F_{mx}) holds for one entity of magnetic force in the lateral direction, as the second entity (F_{mz}) denotes to another one in the axial direction. For this issue of bearing and journal surfaces, the applicable boundary conditions are as follows:

$$\begin{aligned} &u|_{y=0} = 0,\, v|_{y=0} = 0\,,\; w|_{y=0} = 0 \\ &\frac{\partial^2 u}{\partial y^2}|_{y=0} = 0, \frac{\partial^2 w}{\partial y^2}|_{y=0} = 0 \\ &u|_{y=h} = U,\, v|_{y=h} = 0\,,\; w|_{y=h} = 0 \\ &\frac{\partial^2 u}{\partial y^2}|_{y=h} = 0, \frac{\partial^2 w}{\partial y^2}|_{y=h} = 0 \end{aligned} \tag{7}$$

By integrating Eqs. 3 and 5, and using above boundary conditions, the velocity components are obtained as in Eqs. 8, 9 (Hu *et al.* 2022).

$$u = U\frac{Y}{h} + \frac{1}{2\mu_{av}}\left(\frac{\partial p}{\partial x} - F_{mx}\right)\left\{y(y-h) + 2l^2\left[1 - \frac{\cosh\left(\frac{2y-h}{2l}\right)}{\cosh\left(\frac{h}{2l}\right)}\right]\right\} \tag{8}$$

$$w = \frac{1}{2\mu_{av}}\left(\frac{\partial p}{\partial z} - F_{mz}\right)\left\{y(y-h) + 2l^2\left[1 - \frac{\cosh\left(\frac{2y-h}{2l}\right)}{\cosh\left(\frac{h}{2l}\right)}\right]\right\} \tag{9}$$

Reynolds equation

The discretization of Reynolds equation can be achieved by integrating the continuity equation with respect to y and substituting the velocity components (Eq. 10) (Nada *et al.* 2010).

$$\frac{\partial}{\partial x}\left[\frac{1}{\mu_{av}}\left(\frac{\partial P}{\partial x} - F_{mx}\right)\left(h^3 - 12l^2h + 24l^3\tanh\frac{h}{2l}\right)\right] + \frac{\partial}{\partial z}\left[\frac{1}{\mu_{av}}\left(\frac{\partial P}{\partial z} - F_{mz}\right)\left(h^3 - 12l^2h + 24l^3\tanh\frac{h}{2l}\right)\right] = 6U\frac{\partial h}{\partial x} \tag{10}$$

By following isothermal and linear behavior approach as recommended by both Osman *et al.* (2001) and Hu and Xu (2017) in the current case, which assumed to be applicable and appropriate to the magnetic fluid. Then, the magnetic force could be expressed by Eq. 11 (Yin *et al.* 2013).

$$F_m = \mu_o X_m h_m \nabla h_m \tag{11}$$

Where μ_o denotes the permeability of the free surface, X_m represents the magnetic fluid susceptibility, and eventually the entity h_m holds for the intensity of magnetic field. Due to the symmetry about the x-axis, the applied field should be $\frac{\partial h_m}{\partial x} = 0$.

Consequently, $F_{mx} = 0$

The dimensionless Modified Reynolds equation is given by Eq. 12.

$$\frac{\partial}{\partial\theta}\left\{\frac{\overline{f}(\overline{h},\overline{l})}{\overline{\mu}_{av}}\frac{\partial\overline{p}}{\partial\theta}\right\}+\left(\frac{R}{L}\right)^2\frac{\partial}{\partial\overline{z}}\left\{\frac{\overline{f}(\overline{h},\overline{l})}{\overline{\mu}_{av}}\frac{\partial\overline{p}}{\partial\overline{z}}\right\}=6\frac{\partial\overline{h}}{\partial\theta}+\Lambda\frac{\partial}{\partial z}\left\{\frac{\overline{f}(\overline{h},\overline{l})}{\overline{\mu}_{av}}\overline{h}_m\frac{\partial\overline{h}_m}{\partial z}\right\} \tag{12}$$

Where: $\overline{f}(\overline{h},\overline{l})=\left\{\overline{h}^3-12\overline{l}^2\overline{h}+24\overline{l}^3\tanh\frac{\overline{h}}{2\overline{l}}\right\}$, $\overline{P}=\frac{PC^2}{\mu_{in}UR}$, $\overline{\mu}_{av}=\frac{\mu_{av}}{\mu_{in}}$, $\theta=\frac{X}{R}$.

$\overline{h}=\frac{h}{c}$, $\overline{l}=\frac{l}{c}$, couple stress parameter, $\Lambda=C^2\mu_o X_m h_{mo}^2/\omega\mu_{in}L^2$.

The pressure boundary conditions are: $\overline{P}|_{\overline{z}=\pm1/2}=\frac{\partial\overline{P}}{\partial\overline{Z}}|_{\overline{z}=0}=0$, $\frac{\partial\overline{P}}{\partial\theta}|_{\theta=\theta_{cav}}=0.0$ and $\overline{p}\,|_{\theta=\theta_{cav}}=0.0$, $\overline{P}|_{\theta=(2\pi-\varphi)}=p_{in}$.

Where θ_{cav}Represents the cavitation zone.

Magnetic Field Model

Distribution of the magnetic field along the axial direction is assumed to be symmetric, and it can be given as Eq. 13 (Yin *et al.* 2013). Equation 14 shows it in dimensionless form.

$$h_{mz}=h_{mc}-(h_{mc}-h_{me})\left(\frac{2z}{L}\right)^2 \tag{13}$$

$$\overline{h}_m=1-4(1-\beta)\overline{Z}^2 \tag{14}$$

Where: $\overline{h}_m=\frac{h_m}{h_{mc}}$, $\overline{Z}=\frac{z}{L}$; β can be defined as the ratio of the magnetic field strength at the end section h_{me} to the same value at the middle section h_{mc}.

The Modified Energy Equation

In the case of incompressible steady-state couple stress lubrication, the generated temperature field due to viscous shear and hydrodynamic actions defined as Eq. 15 (Wang *et al.* 2001). Equation 16 shows it in dimensionless form.

$$\rho C_o\left\{u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}\right\}=\frac{\partial}{\partial y}\left(k_{oil}\frac{\partial T}{\partial y}\right)+\mu\left\{\left(\frac{\partial u}{\partial y}\right)^2+\left(\frac{\partial w}{\partial y}\right)^2-l^2\left(\frac{\partial u}{\partial y}\frac{\partial^3 u}{\partial y^3}+\frac{\partial w}{\partial y}\frac{\partial^3 w}{\partial y^3}\right)\right\} \tag{15}$$

$$p_l\left(\overline{u}\frac{\partial\overline{T}}{\partial\theta}+p\frac{\partial\overline{T}}{\partial\overline{y}}+\overline{w}\left(\frac{R}{L}\right)\frac{\partial\overline{T}}{\partial\overline{z}}\right)=\frac{1}{\overline{h}^2}\frac{\partial^2\overline{T}}{\partial\overline{y}^2}+O \tag{16}$$

Where: $p_l=\frac{\rho C_0 UC^2}{k_{oil}R}$, $p=-\frac{1}{\overline{h}}\left[\frac{\partial}{\partial\theta}\left(\overline{h}\int_0^{\overline{y}}\overline{u}\,d\overline{y}\right)+\frac{R}{L}\frac{\partial}{\partial\overline{Z}}\left(\overline{h}\int_0^{\overline{y}}\overline{w}\,d\overline{y}\right)\right]$, $O=\beta_o\frac{\overline{\mu}_{av}}{\overline{h}^2}\left[\left(\frac{\partial\overline{u}}{\partial\overline{y}}\right)^2+\left(\frac{\partial\overline{w}}{\partial\overline{y}}\right)^2-\frac{\overline{l}^2}{\overline{h}^2}\left(\frac{\partial\overline{u}}{\partial\overline{y}}\frac{\partial^3\overline{u}}{\partial\overline{y}^3}+\frac{\partial\overline{w}}{\partial\overline{y}}\frac{\partial^3\overline{w}}{\partial\overline{y}^3}\right)\right]$, $\beta_o=\frac{\mu_o U^2}{k_{oil}T_{in}}$.

Oil Viscosity: Temperature Equation

The energy and Reynolds equations are solved numerically with the oil viscosity equation which can be defined in dimensionless form by Eq. 17 (Nada *et al.* 2010; Nassab and Moayeri 2002):

$$\bar{\mu}_{av} = exp^{-\alpha T_{in}(\bar{T}_{av}-1)} \tag{17}$$

Where $\bar{T}_{av}$ is the dimensionless average temperature of cross-film that can be given in Eq. 18.

$$\bar{T}_{av}(\theta,\bar{z}) = \int_0^1 \bar{T}\,(\theta,\bar{y},\bar{z})d\bar{y} \tag{18}$$

Where α is the temperature-viscosity coefficient.

Oil Film Thickness

The oil film thickness for the conventional journal bearing can be expressed in dimensionless form as in Eq. 19 (Abass and Mohammed 2017).

$$\bar{h} = \frac{h}{c}(1+\varepsilon\cos\theta) \tag{19}$$

Heat Transfer Equation

By assuming a steady state condition, the dimensionless equation of heat transfer in the bush can be write as Eq. 20 (Wang *et al.* 2001).

$$\frac{\partial^2\bar{T}_b}{\partial\bar{r}^2} + \frac{1}{\bar{r}}\frac{\partial\bar{T}_b}{\partial\bar{r}} + \frac{1}{\bar{r}^2}\frac{\partial^2\bar{T}_b}{\partial\theta^2} + \left(\frac{R}{L}\right)^2\frac{\partial^2\bar{T}_b}{\partial\bar{z}^2} = 0 \tag{20}$$

The Characteristics of Bearing

Attitude angle and the dimensionless load capacity can be related together as shown in Eqs. 21–24 (Yin *et al.* 2013).

$$\bar{W}_r = \int_0^1\int_0^{2\pi}\bar{P}\cos\theta.d\theta.d\bar{Z} \tag{21}$$

$$\bar{W}_t = \int_0^1\int_0^{2\pi}\bar{P}\sin\theta.d\theta.d\bar{Z} \tag{22}$$

$$\bar{W} = \sqrt{\bar{W}_r^2 + \bar{W}_t^2} \tag{23}$$

$$\varphi = \tan^{-1}\left(-\frac{\bar{W}_t}{\bar{W}_r}\right) \tag{24}$$

The frictional force dimensionless equation at surface of journal and friction coefficient can be defined as Eqs. 25 and 26 (Nada *et al.* 2010):

$$\bar{f}_r = \frac{f_r}{\mu_{in}UL(R/C)} = 2\int_0^{2\pi}\int_0^{0.5}\frac{\bar{\mu}_{av}}{\bar{h}} + \frac{\bar{h}}{2}\left(\frac{\partial\bar{P}}{\partial\theta} + \left(\frac{L}{R}\right)^2\Lambda\bar{h}_m\frac{\partial\bar{h}_m}{\partial\theta}\right)\,d\bar{z}\,d\theta \tag{25}$$

$$C_f = \frac{\bar{f}_r}{\bar{W}}\left(\frac{r}{c}\right) \tag{26}$$

The dimensionless side leakage is calculated by Eq. 27:

$$Q = 2\int_0^{2\pi} -\frac{\overline{f}(\overline{h},\overline{l})}{6\overline{\mu}_{av}}\left(\left(\frac{R}{L}\right)^2 \frac{\partial \overline{p}}{\partial \overline{z}} - \Lambda \overline{h}_m \frac{\partial \overline{h}_m}{\partial \overline{z}}\right) d\theta \tag{27}$$

METHOD OF SOLUTION

The dimensionless governing equations (Reynolds, energy and heat transfer equations) are solved simultaneously by using finite difference method. Its corrected by comparing the results with those published by Kuznetsov *et al.* (2011).

In the present analysis the field of solution is divided into 42 sections in the circumferential direction, 6 sections across the oil film thickness, 20 sections in the axial direction, and 8 divisions in the radial direction of bearing as shown in Fig. 2.

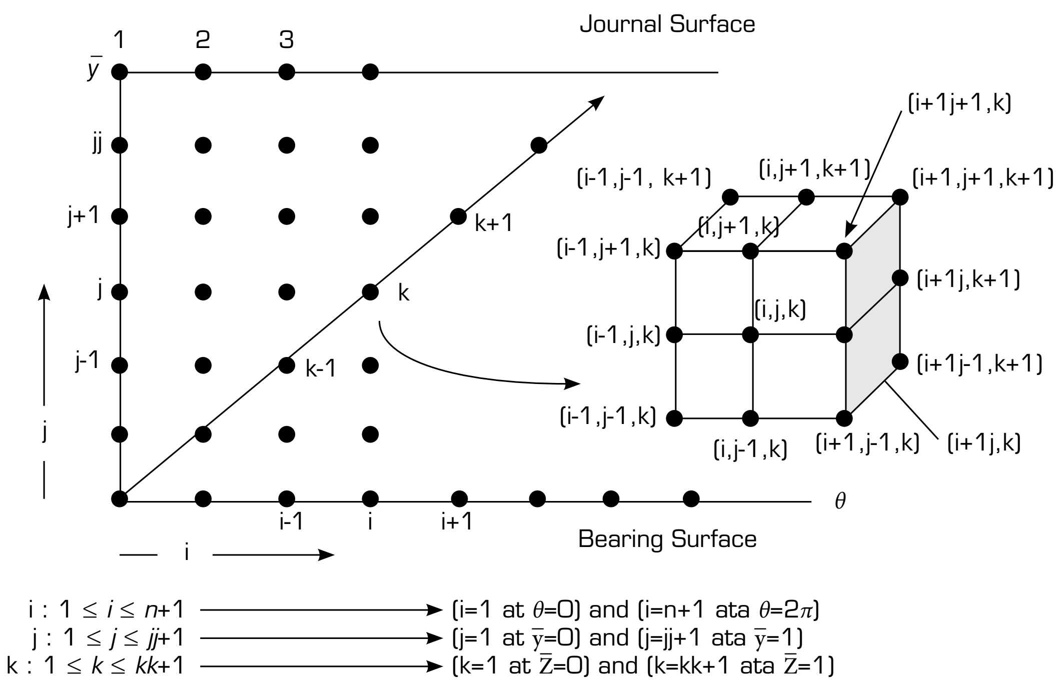

Figure 2. Grid generation for oil film of journal bearing.

At first, initial pressure distribution and initial oil temperature distribution are assumed and then the viscosity of oil and the fluid velocity components are calculated with their derivatives. Secondly, the energy and heat transfer equations are simultaneously solved. Next, with under relaxation of 10^{-6}, the new oil-film temperature is used to compute the new viscosity field. Finally, the viscosity is subsequently used to solve the average Reynolds equation and simultaneous solutions for the equations are obtained iteratively.

RESULTS AND DISCUSSION

Thermohydrodynamic model has been adopted through this work. The effect of magnetic fluids on the performance of the bearing has been considered by using different values of magnetic force coefficient (Λ), while the effect of the couple stress fluid was discussed by using different values of couple stress parameter ($\overline{l}$). The journal bearing dimensions and the oil properties used in the calculations are shown in Table 1. Supply of oil is achieved by an oil groove at the top of the bearing.

Table 1. bearing parameters and lubricant properties used in THD analysis.

Parameter	Symbol	Value and unit
Length of bearing	L	0.08 m
Radius of external bearing	R_{bout}	0.1 m
Radius of journal	R	0.05 m
Radial clearance	C	0.000152 m
Inlet lubricant temperature	T_{in}	40 °C
Ambient temperature	T_a	40 °C
Pressure of inlet lubricant	P_{in}	70,000 pa
Rotational speed	N	2,000 rpm
density of the lubricant	ρ	860 kg·m^{-3}
Viscosity of inlet lubricant	μ_{in}	0.0277 pa·s
Coefficient of temperature - viscosity	α	0.34
Specific heat of lubricant	C_o	2000 J·kg^{-1}·°C
Thermal conductivity of lubricant	$Koil$	0.13 W·m^{-1}·°C
Bush convection heat transfer coefficient	$hconv$	80 W·m^{-2}·°C
Bush thermal conductivity	K_b	250 W·m^{-1}·°C
Groove angle		18°

Figure 3 shows the effect of magnetic fluids with couple stresses on centerline pressure distributions. It can be noted from this figure that a higher pressure of oil film is obtained for the fluid of higher magnetic force coefficient. More than 35% increase in P_{max} when magnetic force coefficient increase from $\Lambda = 0$ to $\Lambda = 0.2$. On the other hand, more than 35% and 134% increase in P_{max} when couple stress parameter increases from $\bar{l} = 0$ to $\bar{l} = 0.2$, 0.4, respectively when $\Lambda = 0$. This can be explained by the lower thickness of oil film value in the bearing with higher couple stress parameter. The increase in P_{max} becomes higher when the bearing lubricated by magnetic fluids with couple stresses.

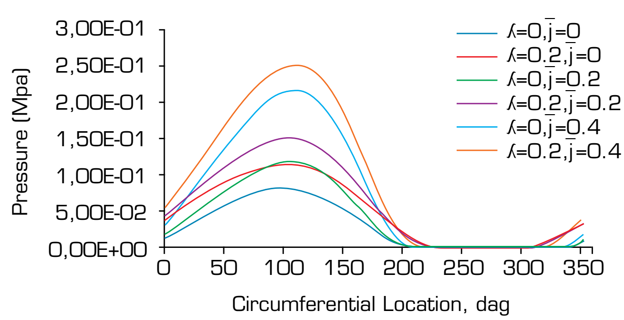

Figure 3. Effect of Λ, $\bar{l}$ on circumferential pressure distribution (e = 0.1).

Figure 4 shows the magnetic force coefficient effect on load capacity for low values of eccentricity ratio. The load capacity changes directly with magnetic force coefficient. The load capacity increases by 24.8% when the magnetic force coefficient changes from $\Lambda = 0$ to $\Lambda = 0.2$) for e = 0.35, $\bar{l} = 0$. This increase becomes more obvious by adding the effect of couple stress fluids especially with high eccentricity ratio values as shown in Figs. 4 and 5. In case of the higher value of eccentricity ratio, the effect of hydrodynamic is the major while the effect of magnetic is so small and can be neglected. more than 2.5 times increasing achieved in load capacity when couple stress parameter increases from ($\bar{l}$=0) to ($\bar{l}$=0.2) for Λ=0, e=0.8.

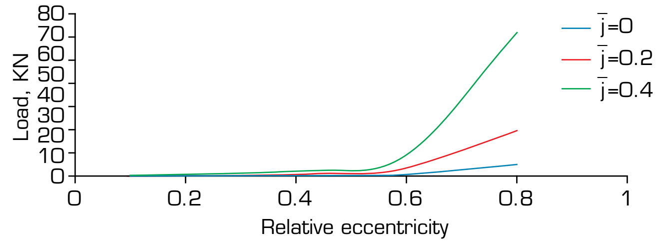

Figure 4. Load capacity (N) with eccentricity ratio for different values of λ, $\bar{l}$.

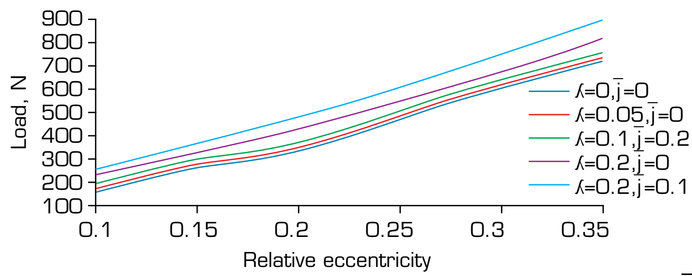

Figure 5. Load capacity versus eccentricity ratio for different values of $\bar{l}$, $\lambda = 0$.

Figure 6 shows the influence of magnetic force coefficient and couple stress parameter on maximum oil film temperature for different eccentricity ratio values. As shown, there is no change in maximum oil film temperature with magnetic force coefficient. With eccentricity ratio up to 0.7, maximum oil film temperature decreases with increase in couple stress parameter. This increase will be more obvious with increase eccentricity ratio value.

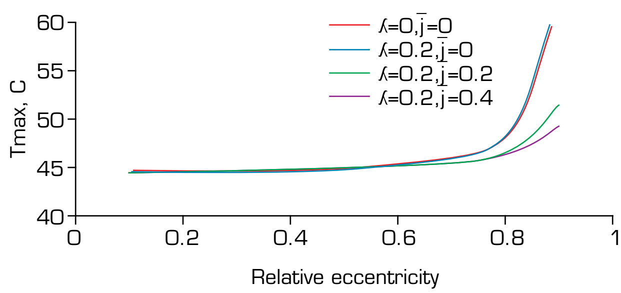

Figure 6. Maximum temperature versus eccentricity ratio for different values of λ, $\bar{l}$.

The effect of magnetic force coefficient and couple stress parameter on the friction force can be shown in Fig. 7. It is noticed that there is no clear effect of magnetic fluid on the frictional force, which means that the magnetic fluid can improve load carrying capacity without any increase in the frictional force. It is also shown that the friction force simply increases with the increase couple stress parameter. This can be explained due to the increase in hydrodynamic effect with couple stress fluids.

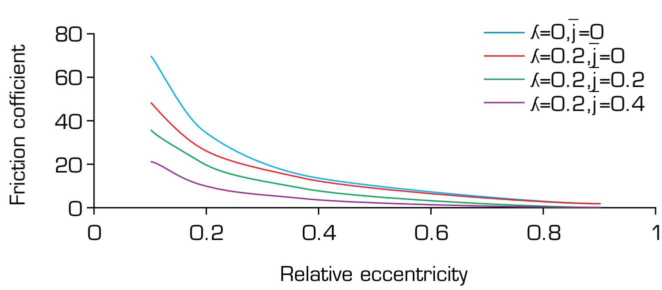

Figure 7. Friction force (N) with the eccentricity ratio for different values of λ, $\bar{l}$.

Figure 8 shows the coefficient of friction for a bearing working under different values of eccentricity ratios. As shown, the bearing has lower coefficient of friction when the magnetic force coefficient and couple stress parameter are taken into consideration.

This means that the increase in friction force (with an increase in the couple stress parameter) is lower than the increase in load capacity because the friction coefficient is the ratio between them. The coefficient of friction decreases by 72.2% when the magnetic force coefficient and couple stress parameter changes from ($\Lambda=0$, $\bar{l}=0$) to ($\Lambda=0.2$, $\bar{l}=0.4$) for e=0.4.

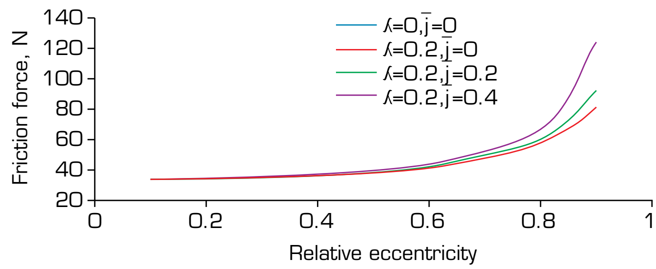

Figure 8. Friction coefficient with relative eccentricity ratio for different values of Λ, $\bar{l}$.

The effect of couple stress parameter and magnetic force coefficient on the side leakage flow can be observed in Fig. 9. It seems that the side leakage flow of the oil decreases by 36% when the bearing working with magnetic fluid compared to Newtonian fluid (at e=0.8). For the couple stress fluid and ($\Lambda=0$), the side leakage flow is often the same as that Newtonian fluid. For the journal bearing lubricated by magnetic fluids with couple stress, the side leakage flow is higher compared to journal bearing lubricated by magnetic fluids without couple stress.

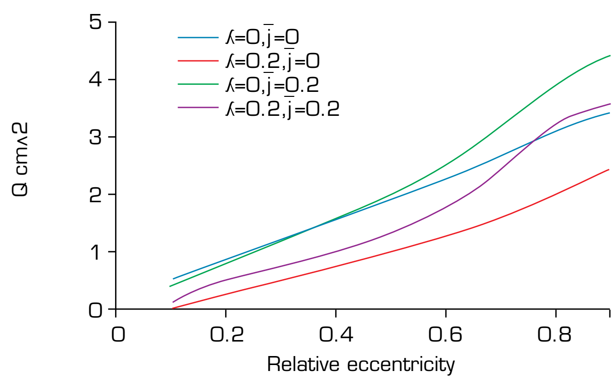

Figure 9. Side leakage flow with eccentricity ratio for different values of Λ, $\bar{l}$.

CONCLUSION

Without couple stress ($\bar{l}$ = 0), the magnetic fluid improved P_{max} and load capacity. No obvious effect can be noted for the magnetic force coefficient on the friction force. As a result, coefficient of friction decreases with increase magnetic force coefficient but this effect decreases when the bearing working under high eccentricity ratio. The magnetic fluid also reduce side leakage flow.

In case of the higher value of eccentricity ratio, the effect of hydrodynamic is the major while the effect of magnetic is so small and can be neglected.

There is a considerable improve in P_{max} and load capacity when the effect of couple stresses is taken into consideration. Although friction force increased, friction coefficient decreases with the increase in couple stress parameter and this mean that the increase of friction force (because $\bar{l}$ increasing is lower than load capacity increasing).

At high eccentricity ratios the effect of couple stresses are more effective. From the above results it can be concluded that the magnetic fluids with couple stresses are better lubricant than (magnetic fluid only ($\bar{l}$=0), Newtonian fluids, and couple stress fluid (Λ=0).

AUTHORS' CONTRIBUTION

Conceptualization: Dahham RB; **Methodology:** Dahham RB; **Software:** Dahham RB, **Validation:** Dahham RB and Hashim NA; **Formal analysis:** Dahham RB and Al-Fatlwe FMK; **Investigation:** Dahham RB; **Resources:** Dahham RB; **Supervision:** Hashim NA; **Project administration:** Hashim NA; **Writing – Original Draft:** Dahham RB; **Writing – Review & Editing:** Hashim NA.

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REFERENCES

Abass BA, Mohammed NF (2017) Thermohydrodynamic characteristics of worn journal bearing lubricated with oil containing nanoparticles additive. Al-Nahrain Journal for Engineering Sciences 20(3):526-543.

Andharia PI, Lakhnotra SB, Pandya H (2019) A study of longitudinally rough Rayleigh step bearing formed by a magnetic fluid. J Appl Sci Comput VI (III):760-766.

Wang A, Pan J, Wu H, Ye J (2022) Structural design and lubrication properties under different eccentricity of magnetic fluid bearings. Appl Sci 12(14):7051. https://doi.org/10.3390/app12147051

Nada GS, Abdel-Jaber GT, Abdo HS (2010) Thermal effects on hydrodynamic journal bearings lubricated by magnetic fluids with couple stresses. Int J Mech Mechatron Eng 12(5):12-20.

Burcan J, Burcan-Piotrowicz E, Slawinska A (2004) Investigation on magnetic fluids as lubricant. Indian J Eng Mater Sci 11:338-342.

Lin JR (2016) Longitudinal surface roughness effects in magnetic fluid lubricated journal bearings. J Mar Sci Technol 24(4):5. https://doi.org/10.6119/JMST-016-0201-1

Kuznetsov E, Glavatskih S, Fillon M (2011) THD analysis of compliant journal bearings considering liner deformation. Tribol Int 44(12):1629-1641. https://doi.org/10.1016/j.triboint.2011.05.013

Nassab SAG, Moayeri MS (2002) Three-dimensional thermohydrodynamic analysis of axially grooved journal bearings. Proc Inst Mech Eng J: J Eng Tribol 216(1):35-47. https://doi.org/10.1243/1350650021543870

Osman TA, Nada GS, Safar ZS (2001) Static and dynamic characteristics of magnetized journal bearings lubricated with ferrofluid. Tribol Int 34(2001):369-380. https://doi.org/10.1016/S0301-679X(01)00017-2

Patel MP, Patel HC, GM Deheri (2015) Magnetic fluid lubrication of an infinitely long slider bearing with rough surfaces. J Serb Soc Comput Mech 9(2):10-18.

Hu R, Xu C (2017) Influence of magnetic fluids' cohesion force and squeeze dynamic effect on the lubrication performance of journal bearing. Adv Mech Eng 9(9):1-13. https://doi.org/10.1177/1687814017732671

Hu R, Xu C, Maiti R, Mcquade C, Yang Y (2022) The lubrication characteristics of magnetized journal bearings affected by magnetic fluids considering the elasticity of its liner. Adv Mech Eng 14(6):1-10. https://doi.org/10.1177/16878132221108263

Shukla S, Deheri G (2013) Effect of slip velocity on magnetic fluid lubrication of rough porous Rayleigh step bearing. J Mech Eng Sci 4:532-547. https://doi.org/10.15282/jmes.4.2013.17.0050

Singh JP, Ahmad N (2011) Analysis of a porous-inclined slider bearing lubricated with magnetic fluid considering thermal effects with slip velocity. J Braz Soc Mech Sci & Eng 33(3):351-356. https://doi.org/10.1590/S1678-58782011000300011

Stokes VK (1966) Couple stresses in fluids. Phys Fluids 9:1709-1715. https://doi.org/10.1063/1.1761925

Urreta H, Leicht Z, Sanchez A, Agirre A, Kuzhir P, Magnac G (2009) Hydrodynamic bearing lubricated with magnetic fluids. J Phys: Conf Ser 149:012113. https://doi.org/10.1088/1742-6596/149/1/012113

Wang XL, Zhu KQ, Wen SZ (2001) Thermohydrodynamic analysis of journal bearings lubricated with couple stress fluids. Tribol Int 34(5):335-343. https://doi.org/10.1016/S0301-679X (01)00022-6

Yin X, Ma J, Fang Y, Jin S (2013) Analysis of fluid-thermal coupling in Ferro fluid bearing used in high speed machines. Paper presented 2013 International Conference on Electrical Machines and Systems (ICEMS), 1399-1404, https://doi.org/10.1109/ICEMS.2013.6713248