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Effect of Journal Misalignment on the Static Characteristics of Porous Journal Bearings Lubricated with Couple Stress Fluid

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Abstract

In this paper, a theoretical study to the effect of journal misalignment on the static characteristics of oil filled porous journal bearing when lubricated with couple stress fluid has been carried out.

The analytical model used through this work is for a bearing with isotropic permeability. Considering isotropic permeability the Reynolds' equation for the oil film is modified to include a so – called filter term and the effect of fluid coupled stress. The pressure equation for the porous medium is obtained from Darcy's law and continuity equation. The equation which was used to evaluate the oil film thickness was modified to include the effect of possible misalignment in longitudinal and transverse directions. The governing equations with appropriate boundary conditions are numerically solved using a suitable numerical technique. A computer program has been prepared to solve the governing equations. The validity of the program has been tested by comparing the results obtained through this work with that published in available works. The comparison shows a good agreement between the results obtained through this work and that published by other workers.

By comparing the behavior of aligned and misaligned bearing it was found that the journal center misalignment has a considerable effect on the performance parameters of the bearing which can not be neglected.

Key words: *Static Characteristics, Self Lubricated Bearing, Misaligned Bearing, Uniform Permeability, Coupled Stress.*

1. Introduction

In most theoretical investigations of hydrodynamic lubrication it has been assumed that the journal and the bearing axis are aligned. This is an unrealistic assumption for the bearing operating with small film thickness since the bearings often operate in misaligned condition. Bearing misalignment can vary in magnitude and direction. The most important cases to be considered are vertical , twisting , and horizontal misalignment or combination of these can also occur.

Most investigators in this area confined their work to solid bearing. A little work has been found related to the behaviour of aligned porous bearing lubricated with coupled stress fluid, while most works found related to study the behaviour of porous bearing lubricated with Newtonian – lubricant as can be seen in references [1 – 8]. It was found that the Newtonian fluid constitutive approximation is not satisfactory engineering approach to many lubrication problems. Hence the effects of non – Newtonian behaviour must be taken into account in the realistic study of these bearings. Many micro continuum theories have been developed to describe the behaviour of non – Newtonian fluids. Airman et. al. [9,10] developed a micro continuum theory to describe the behaviour of fluids containing structure such as polymer. Naduvinamani et. al. [11] used the couple stress theory to analyze the squeeze film lubrication of a short porous bearing. A main conclusion through this work is that under cyclic load the couple stress fluid provide a reduction in journal velocity and an increase in minimum permissible height of squeeze. The surface roughness effect in a short porous journal bearing lubricated with a couple stress fluid was studied by Naduvinamani et. al. [12]. It was observed during this work that the effect of surface roughness on the bearing characteristics are more pronounced for couple stress fluids as compared with Newtonian fluids. Using the couple stress theory to investigate the lubrication mechanism of synovial joints was carried out by Bujurke and Ramesh [13]. They showed that the effects of surface roughness are considerably prounced for poroelastic bearings with couple stress fluid as lubricant compared to classical case.

Recently the flow and heat transfer of couple stress fluid in a porous channel with expanding and contracting walls have been investigated by Srinivasacharya and Srinivasecharyulu [14]. Graphs for velocity components and temperature distribution are presented for different values of the fluid and geometric parameters. Naduvinamani and Patil [15] used a numerical solution to the finite modified Reynolds' equation for couple stress squeeze film lubrication of porous journal bearing. They concluded that under a cyclic load, the effect of couple stress is to reduce the velocity of the journal center and to increase the minimum permissible height of the squeeze film.

So far it seems that all the above researches ignore the effect of journal center misalignment. The effect of bearing misalignment on the bearing characteristics such as the mean load carrying capacity, mean frictional force, and the other bearing characteristics are investigated during this work.

2. Numerical analysis

2.1. Model of the misaligned porous journal bearing

The model of misaligned porous journal bearing lubricated with couple stress fluid shown in Figure (1) is adopted through the present work.

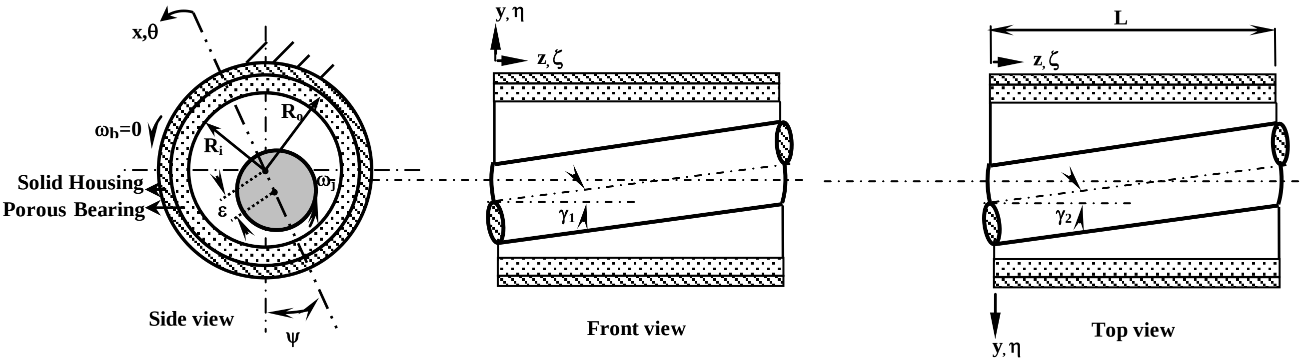


Fig. 1. Geometrical Configuration of the Misaligned Porous Journal Bearing.

2.2. Governing Equations

The governing equation for the pressure distribution in the oil – film is given by the modified Reynolds' equation including a so – called filter term and coupled stress effect. For a journal bearing lubricated with coupled stress fluid with a constant viscosity the modified Reynolds' equation can be written as [12]:-

$$\frac{\partial}{\partial x}\left\{\left[g(h,\tau)+\frac{12k\delta}{(1-\beta)}\right]\frac{\partial P}{\partial x}\right\}+\frac{\partial}{\partial z}\left\{\left[g(h,\tau)+\frac{12k\delta}{(1-\beta)}\right]\frac{\partial P}{\partial z}\right\}=6\mu U\frac{\partial h}{\partial x} \quad \ldots(1)$$

Where:

$\tau=\sqrt{\mu/\eta}$

Equation (1) can be written in dimensionless form as follows:-

$$\frac{\partial}{\partial\theta}\left\{\left[g^{\wedge}(h^{\wedge},l_c^{\wedge})+\frac{12\varphi}{(1-\beta)}\right]\frac{\partial P^{\wedge}}{\partial x}\right\}+\frac{1}{4a^2}\frac{\partial}{\partial z^{\wedge}}\left\{\left[g^{\wedge}(h^{\wedge},l_c^{\wedge})+\frac{12\varphi}{(1-\beta)}\right]\frac{\partial P^{\wedge}}{\partial z^{\wedge}}\right\}=6\frac{\partial h^{\wedge}}{\partial\theta} \quad \ldots(2)$$

Where: [15]

$$g^{\wedge}(h^{\wedge},l_c^{\wedge})=h^{\wedge 3}-12l_c^{\wedge 2}h^{\wedge}+24l_c^{\wedge 3}\tanh\left(\frac{h^{\wedge}}{2l_c^{\wedge}}\right) \quad \ldots(3)$$

$$\beta=\left(\frac{\eta}{\mu}\right)\Big/k \quad \ldots(4)$$

Where β represents the ratio of microstructure size to the porous size. For a vary small β, i.e. β<<1, the polar additives percolate into the porous matrix. It in clear that when β→0, the flow become Newtonian flow.

It can be shown that the couple stress parameter (l_c) has a units of length and it may be regarded as chain length of polar additives and can be evaluated as [15,16]:

$$l_c = \sqrt{\eta/\mu} \qquad \ldots(5)$$

The fluid film thickness in cooperating the effect of journal misalignment in both directions (axial and twisting) can be expressed in non – dimensional form as follows [17]:-

$$h^{\wedge} = 1 + \varepsilon\cos\theta - \zeta\sigma_1\cos\theta + \zeta\sigma_2\sin\theta \qquad \ldots(6)$$

Where:

$$\sigma_1 = 2\left(\frac{R_i}{C}\right)\left(\frac{L}{D}\right)\tan\gamma_1 \qquad \ldots(7)$$

$$\sigma_2 = 2\left(\frac{R_i}{C}\right)\left(\frac{L}{D}\right)\tan\gamma_2 \qquad \ldots(8)$$

The two independent misalignment angles (γ_1 and γ_2) are measured from ξ=0; it is clear from equation (6) that the oil gap geometry depends on (θ and ξ).

Due to the continuity of the fluid motion at the porous matrix, the oil pressure inside the porous matrix satisfies the Laplace equation which can be expressed as [12, 15]

$$\frac{\partial^2 P^*}{\partial x^2} + \frac{\partial^2 P^*}{\partial y^2} + \frac{\partial^2 P^*}{\partial z^2} = 0.0 \qquad \ldots(9)$$

The classical Reynolds' boundary conditions are adopted through this work. The Reynolds' conditions can be expressed as [1,18];

$$\left.\begin{aligned}
&P^{\wedge}(\theta,0) = P^{\wedge}(\theta,1) = P^{\wedge^*}(r^{\wedge},\theta,0) = P^{\wedge^*}(r^{\wedge},\theta,1) = 0\\
&P^{\wedge}(0,z) = P^{\wedge^*}(r^{\wedge},0,Z) = 0\\
&P^{\wedge}(\theta,Z) = P^{\wedge^*}(r^{\wedge},\theta,Z) at (r^{\wedge}) = 1\\
&P^{\wedge^*}(r_o^{\wedge},\theta,1/2) = P_s^{\wedge}\\
&\hat{P} = \frac{\partial \hat{P}}{\partial\theta} = 0 at \theta = \pi + \alpha_c
\end{aligned}\right\} \qquad \ldots(10)$$

Where α_c is the angle at which cavitations starts.

2.3. Bearing Parameters

After evaluating the pressure field through the oil film, the bearing performance parameters can be calculated as follows:-

The radial and tangential components of the load are found as :

$$\left.\begin{aligned}
\left(\hat{W}_R\right) &= -\int_0^1\int_{0_1}^{\pi+\alpha_c}\left(P^{\wedge}(\theta,\xi)\cos\theta\right)d\theta d\xi\\
\left(\hat{W}_T\right) &= \int_0^1\int_0^{\pi+\alpha_c}\left(P^{\wedge}(\theta,\xi)\sin\theta\right)d\theta d\xi
\end{aligned}\right\} \qquad \ldots(11)$$

The total bearing load carrying capacity can be expressed in dimensionless form as;

$$\left(\hat{W}\right) = \sqrt{\left(\hat{W}_R\right)^2 + \left(\hat{W}_T\right)^2} \qquad \ldots(12)$$

The attitude angle can be found as:-

$$(\Psi) = \tan^{-1}\left(W_T^{\wedge}/W_R^{\wedge}\right) \qquad \ldots(13)$$

The friction force can be expressed in dimensionless form as [16] ;

$$F_h^{\wedge} = \int_0^1\int_0^{\pi+\alpha_c}\left(\frac{1}{h^{\wedge}} + \frac{h^{\wedge}}{2}\frac{\partial P^{\wedge}}{\partial\theta}\right)d\theta dz^{\wedge} \qquad \ldots(14)$$

The coefficient of friction can be evaluated as:

$$\mathrm{f(R/C)} = \frac{F_h^{\wedge}}{W^{\wedge}} \qquad \ldots(15)$$

The oil side leakage flow for the porous journal bearing can be evaluated as [16];

$$Q_s^{\wedge} = \int_0^1\int_0^{\pi+\alpha_c}\left.\frac{\partial P^{\wedge}}{\partial z^{\wedge}}\right|_{z^{\wedge}=1} h^{\wedge} * \left\{y^{\wedge^2}h^{\wedge^2} - y^{\wedge}h^{\wedge^2} + 2l_c^{\wedge^2}\left[1 - \frac{\cosh\left(\frac{2y^{\wedge}h^{\wedge} - h^{\wedge}}{2l_c^{\wedge}}\right)}{\cosh\left(\frac{h^{\wedge}}{2l_c^{\wedge}}\right)}\right]\right\}d\theta dy^{\wedge} \qquad \ldots(16)$$

2.4. Method of solution

The pressure distribution in the oil film can be obtained by solving the modified Reynolds' equation (1), the Laplace equation (9) coupled with the modified oil film thickness equation (6) with the Reynolds' boundary conditions equation

(10) simultaneously using iterative numerical scheme. The field of solution is divided into grid spacing (N_1=8, N_2=180, N_3=12); each has a mesh size $\Delta r^{\wedge}$ * $\Delta\theta$ *$\Delta z^{\wedge}$ for porous matrix and grid spacing (N_2=180, N_3=12); each has a mesh size $\Delta\theta$ *$\Delta z^{\wedge}$ for oil film thickness.

Gauss Siedel iterative scheme with successive under relaxation has been used to solve the governing equation of the problem. The iterations are stooped when the following convergence criteria are satisfied.

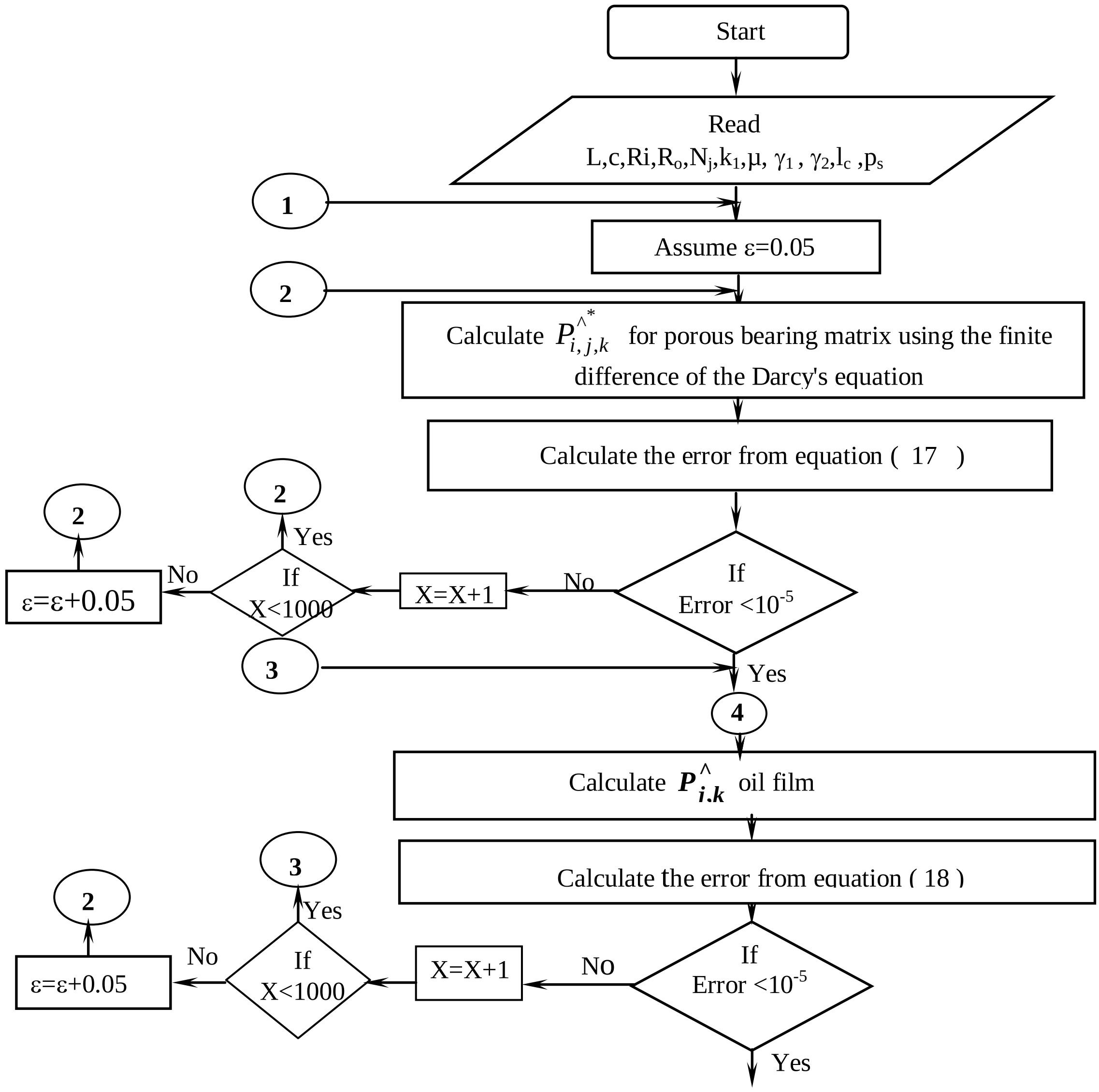
For the oil pressure inside the porous matrix the convergence criterion can be expressed as:-

$$\left(\frac{\sum\sum\sum \left| P_{i,j,k}^{\wedge *(n+1)} - P_{i,j,k}^{\wedge *(n)} \right|}{\sum\sum\sum \left| P_{i,j,k}^{\wedge *(n)} \right|} < 10^{-5} \right) \qquad \ldots(17)$$

While for the oil pressure in the bearing oil film the convergence criterion can be expressed as:-

$$\left(\frac{\sum\sum \left| P_{j,k}^{\wedge (n+1)} - P_{j,k}^{\wedge (n)} \right|}{\sum\sum \left| P_{j,k}^{\wedge (n)} \right|} < 10^{-5} \right) \qquad \ldots(18)$$

A computer program written in FPRTRAN – 90 language has been used to solve the governing equations of the problem. Figure (2) shows the flow chart of the computer program used during this work.



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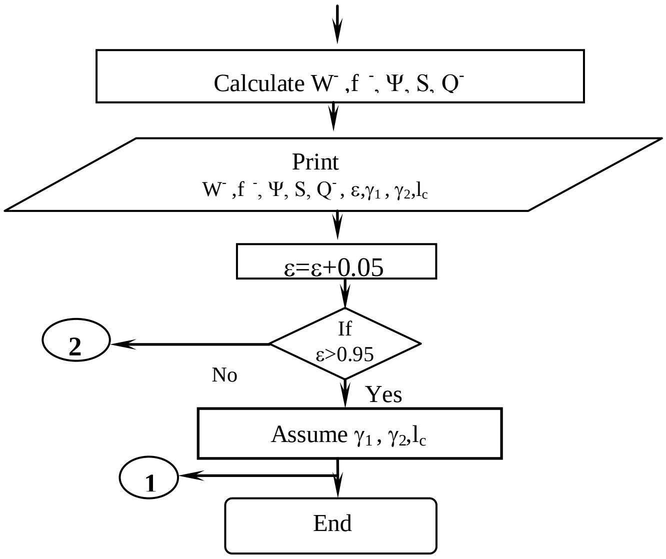

Fig. 2.Flow Chart of the Computer Program.

3. Results and discussion

The combined effect of couple stress and journal center misalignment on the performance of porous journal bearing is predicted through this work. The effect of couple stress is taken into consideration on the basis of stokes couple stress fluid model for lubricant. A finite porous journal bearing operated under steady conditions is analyzed. All bearing characteristics such as the load carrying capacity (W), frictional coefficient f(R/C) , friction force, the lubricant side leakage and the attitude angle are functions of the couple stress parameter (l_c) and the eccentricity ratio (ε).

The computer program prepared to solve the governing equations of the present work is verified by comparing the obtained results with those obtained by Mokhiamer (1999). Figures (1) and (2) shows a comparison between the pressure distribution obtained in this work with that obtained by Mokhiamer (1999) for eccentricity ratio of (0.4, 0.6) respectively. It can be shown from these figures that the maximum percentage of error between the obtained and the published results is (6.25% and 5%) respectively, while Figure (3) shows that the maximum error between the attitude angle obtained during this work in comparison with that obtained by Mokhiamer (1999) did not exceed (6%). The percentages of error mentioned above are based on the difference between the calculated and published data divided by the published data. It is clear from the above figures that the results obtained through this work are in a good agreement with the published results and give a reasonable reliability to the program used to analyze the problem of the present work.

Figure (4) shows that the load carrying capacity of the bearing decreases when combined misalignment (axial and twisting) which gives an indication that the oil film thickness increases when the bearing suffers from the combined misalignment is taken into consideration. A maximum decrease of (28%) in load carrying capacity is noticed in this case. A greater decreases in load carrying capacity is noticed when axial misalignment of the bearing is taken into consideration especially for low values of eccentricity ratio. The maximum reduction is calculated and found to be about (35%) , which indicates an increase in oil film thickness in this range of eccentricity ratios. A slight increase in load carrying capacity is noticed when the twisting misalignment is considered especially for low values of eccentricity ratio (i.e. to about ε = 0.4) while it has no effect for higher values of eccentricity ratio.

A decrease in attitude angle is seen when combined axial and twisting misalignment is taken into consideration as shown in Figure (5). A maximum decrease is calculated and found to be (20%). This can be attributed to the variation in load carrying capacity components mentioned before. An increase in attitude angle is noticed when axial misalignment of the bearing is taken into consideration. This is due to the increase in oil film thickness in this case.

A slight decrease in coefficient of friction is shown when the twisting misalignment is considered as shown in Figure (6) , which indicates a lower shearing rate of the oil in this case. An increase in coefficient of friction is shown when twisting and combined misalignment is taken into consideration. This is due to the increase of the shearing rate of the lubricant in this case.

Figure (7) shows that the side leakage flow rate of the bearing deceases when the axial misalignment of the journal bearing is taken into consideration. This can be attributed to the increase in oil film thickness, which affects the velocity component in axial direction, and the pressure gradient in circumferential direction. A decrease in side leakage flow rate of (20%), is noticed in this case. An increase in side leakage flow rate has been shown when axial and twisting misalignment of the journal bearing have been taken into consideration. A higher increase in side leakage is noticed when the twisting misalignment only is taken into consideration which indicate the increase in oil film thickness of the lubricant.

Figure (8) shows that the load carrying capacity for a porous bearing lubricated with couple stress fluid is higher than that obtained when the bearing lubricated with Newtonian lubricant for different values of the eccentricity ratios. This can be attributed to the higher viscosity of the couple stress lubricant than that of Newtonian lubricant. It is clear from this figure that a higher enhancement is obtained when the bearing is lubricated with couple stress lubricant of higher couple stress parameter (l_c). About (40%) enhancement in load carrying capacity is shown for the bearing lubricated with a lubricant of couple stress parameter of ($l_c = 0.2$).

The bearing attitude angle is seen to have lower values for the bearing lubricated with couple stress lubricant than that lubricated with Newtonian lubricant as shown in Figure (9). This is attributed to the increase in load carrying capacity component mentioned above when lubricating the bearing with couple stress fluid. The decrease in attitude angle becomes higher when the bearing lubricated with couple stress lubricant of higher couple stress parameter.

A decrease in friction coefficient of the bearing has is when the bearing lubricated with couple stress lubricant rather than that lubricated with Newtonian lubricant as shown in Figure (10). The decrease in coefficient of friction become higher when the bearing lubricated with a couple stress lubricant with higher couple stress parameter. This can be explained if we know that the load carrying capacity of the bearing is increased in this case, since the coefficient of friction is inversely proportional to the value of the load.

Figure (11) shows a slight increase in oil side leakage from the bearing lubricated with couple stress lubricant than that for lubricated with Newtonian lubricant. A couple stress parameter (l_c) is shown to have a slight effect on the oil side leakage of the bearing. A slight increase in oil side leakage is shown with increasing values of (l_c). The increase oil side leakage can also be attributed to the effect of combined misalignment of the bearing which causes larger oil film thickness.

The effect of length to diameter ratio of the bearing on the load carrying of the bearing is shown in Figure (12) . It is clear that the load carrying capacity increase for the bearing with higher length to diameter ratio, while the coefficient of friction decreases as shown in Figure (13).

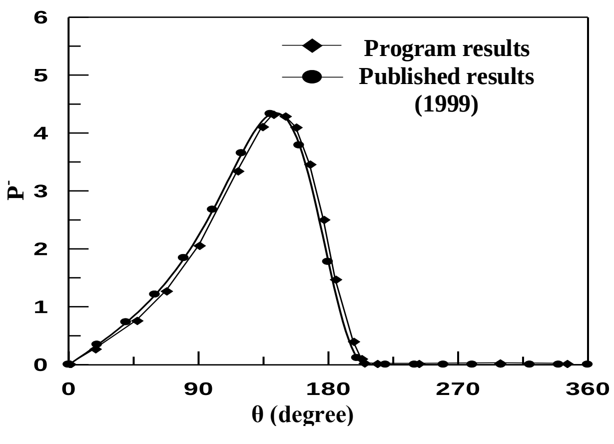

Fig. 1. Comparison Pressure Distribution Between Present and Published Result U.M.Mokhiamer et. al. (1999) at P_s= 0.1, l_c^-=0.4,R/L=0.5 ,γ_1=0.0 , γ_2=0.0 and ε=0.4.

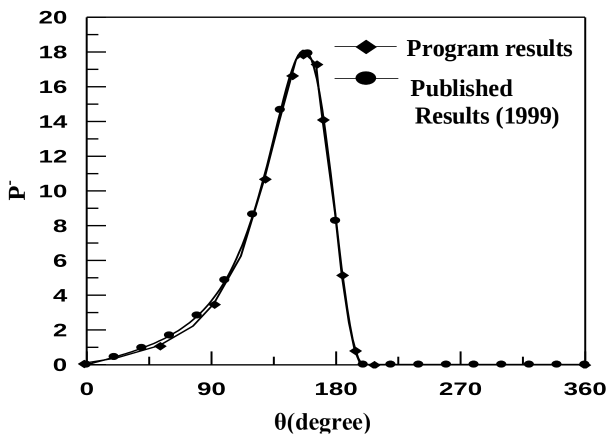

Fig. 2. Comparison Pressure Distribution Between Present and Published Result U.M.Mokhiamer et. al. (1999) at P_s= 0.1, l_c^-=0.4,R/L=0.5 ,γ_1=0.0 , γ_2=0.0 and ε=0.6.

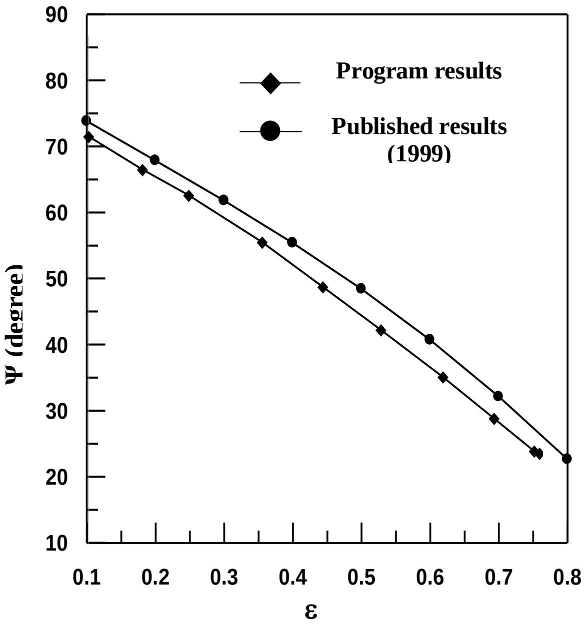

Fig. 3. Comparison attitude angle versus eccentricity ratio between present and published result U.M.Mokhiamer et. al. (1999) at P_s= 0.1, l_c^- =0.4,R/L=0.5 ,γ_1=0.0 , γ_2=0.0 and ε=0.4.

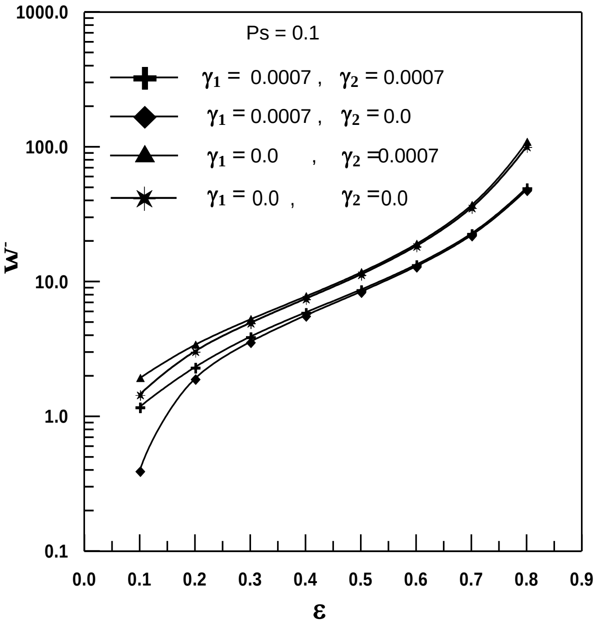

Fig. 4. Load carrying capacity verse eccentricity ratio for different misalignment ratios at l_c^- = 0.4.

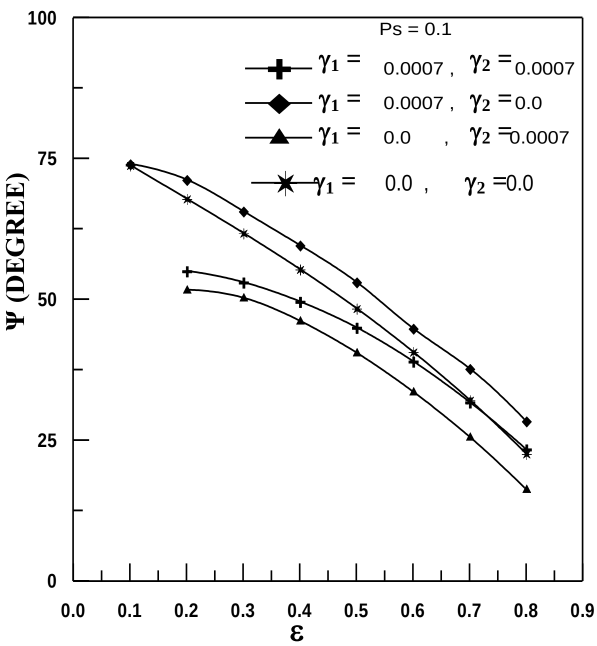

Fig. 5. Attitude angle verse eccentricity ratio for different misalignment ratios at l_c^- = 0.4.

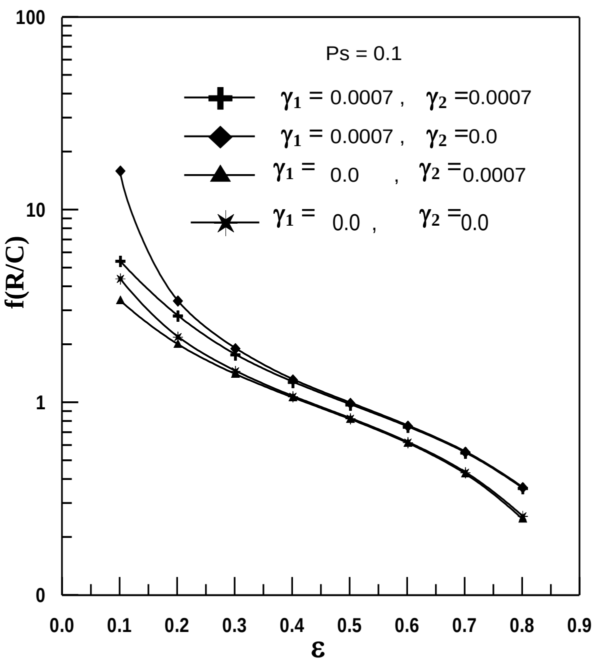

Fig. 6. Coefficient of friction verse eccentricity ratio for different misalignment ratios at l_c^- = 0.4.

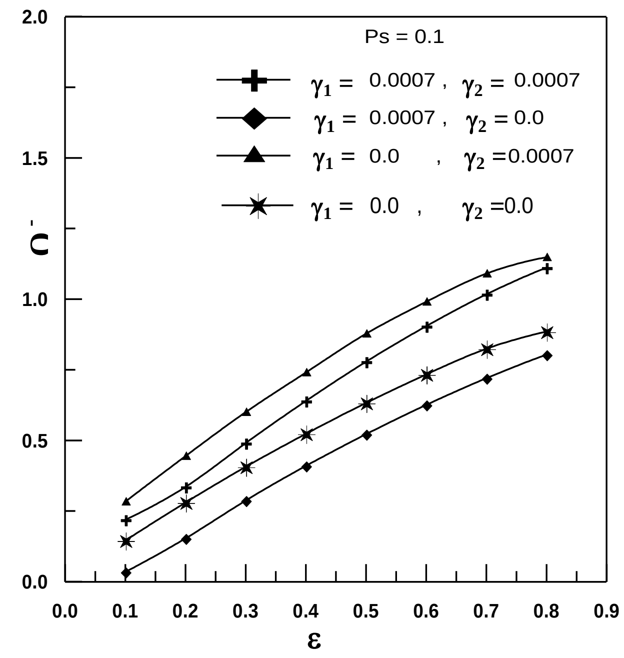

Fig. 7. Dimensionless Side Flow Verse Eccentricity Ratio for Different Misalignment Ratios at $l_c^- = 0.4$.

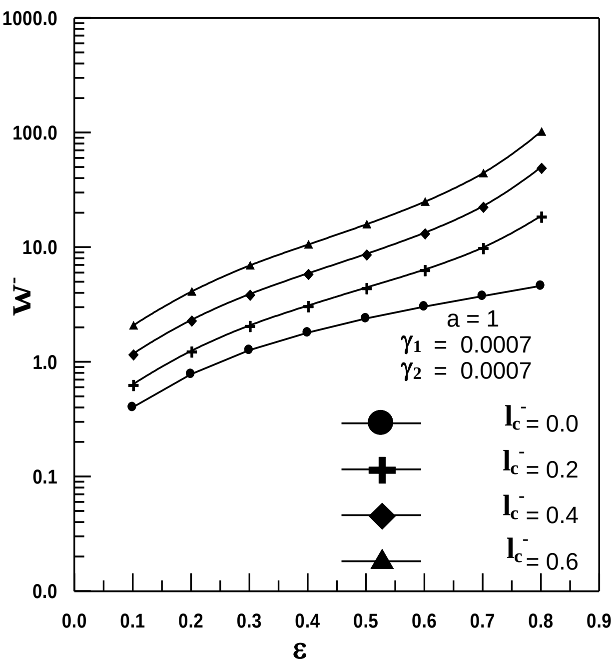

Fig. 8. Load Carrying Capacity Verse Eccentricity Ratio for Various Values of Couple Stress parameter.

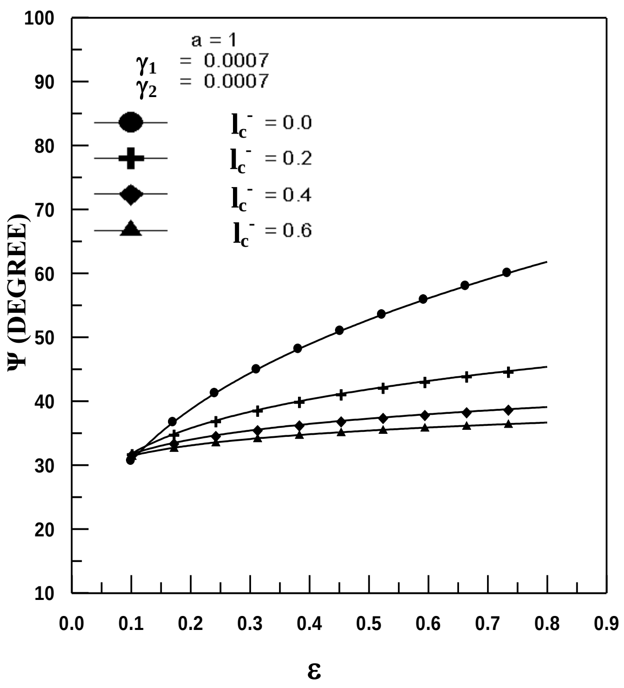

Fig. 9. Attitude Angle Verse Eccentricity Ratio for Various Values of Couple Stress Parameter.

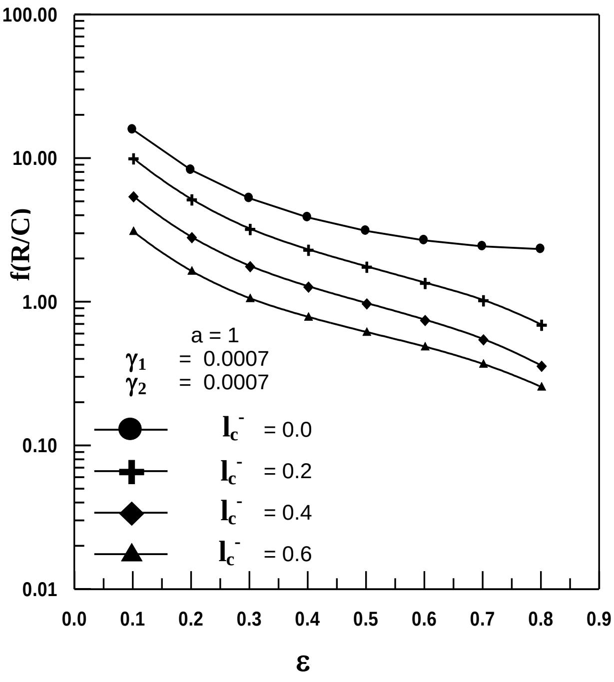

Fig. 10. Coefficient of Friction Verse Eccentricity Ratio for Various Values of Couple Stress Parameter.

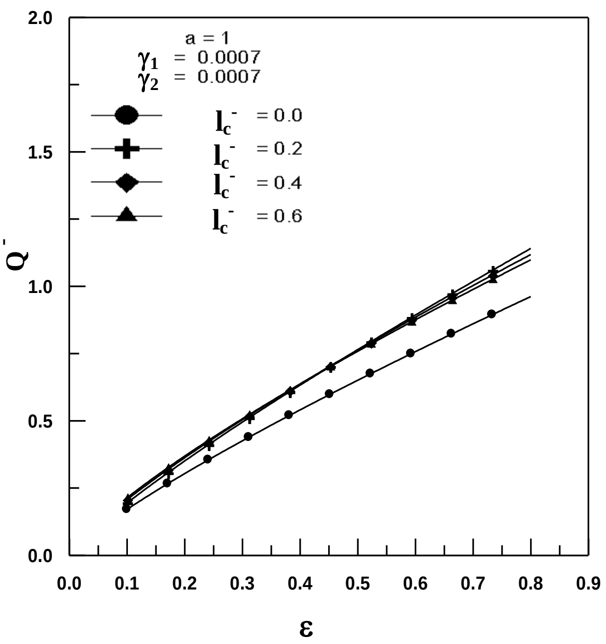

Fig. 11. Dimensionless Side Flow Verse Eccentricity Ratio for Various Values of Couple Stress Parameter.

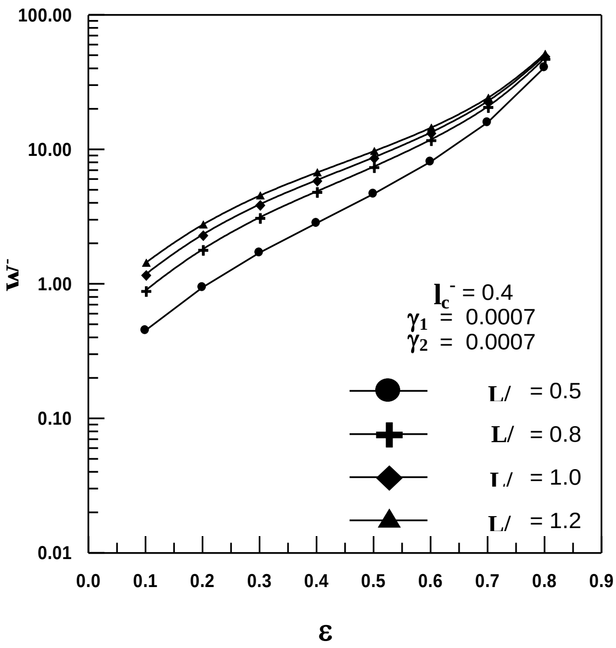

Fig. 12. Load Carrying Capacity Verse Eccentricity Ratio for Various Values of L/D Ratios.

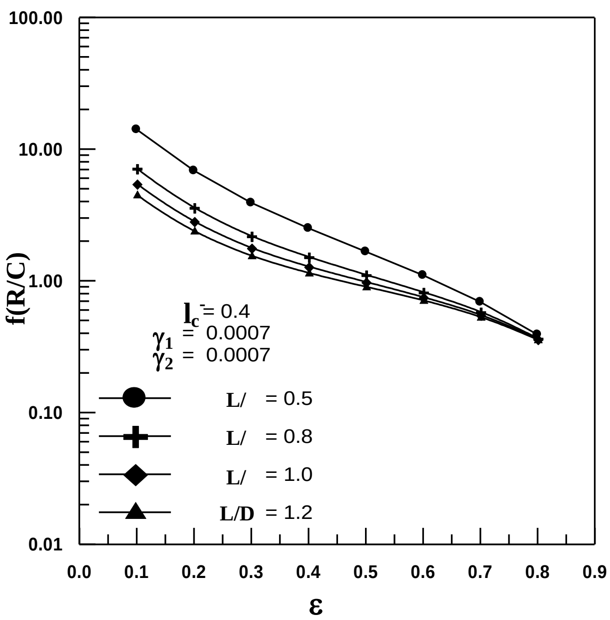

Fig. 13. Friction Factor Verse Eccentricity Ratio for Various Values of L/D Ratios.

4. Conclusions

On the basis of microcontinum theory developed by Stokes, the present investigation reveals the effect of couple stresses on the performance of misaligned finite length porous journal bearing. The results obtained through this work leads to the following main conclusions:-

1- The journal center misalignment has a considerable effect on the performance parameters of the bearing which can not be neglected. The presence of combined misaligned (axial and twisting) causes a decreases of the load carrying capacity and attitude angle.
2- The presence of bearing axial misalignment causes a slight increase in load carrying capacity and attitude angle, while a decrease in side leakage of the bearing, is noticed in this case.
3- Using the coupled stress fluid as a lubricant seems to improve some of the steady state characteristics of the porous bearings in comparison with the bearing lubricated with Newtonian lubricant. An increase in load carrying capacity, a decrease in attitude angle, friction coefficient and slight increase in side leakage flow have been noticed.

5. Nomenclature

The following symbols are used throughout this work.

a	ratio of length to diameter
c	Journal Bearing Clearance (m)
D	diameter of bearing (m)
F	friction force (N)
$F^\wedge$	Dimensionless friction force, $F^\wedge = (Fc/\mu\omega_j r^2 L)$
$f^\wedge$	Dimensionless Friction Coefficient, $f^\wedge = (R/c)f$
$h^\wedge$	Dimensionless Film Thickness, $(h^\wedge = h/c)$
L	Length of the Bearing (m)
L_c	characteristics length of the additive
$L_c^\wedge$	dimensionless coupled stress parameter, $L_c^\wedge = L_c/c$
k	Permeability parameter (m^2)
N_j	Journal Rotational Speed (r.p.m)
$P^\wedge$	Dimensionless Oil-Film Pressure, $P^\wedge = c^2P/(r^2\,\mu\omega_j)$
$P^{\wedge*}$	Dimensionless Oil – Film Pressure Inside the Porous Matrix, $P^{\wedge*} = c^2P^*/(r^2\mu\omega_j)$
P_s	Supply Pressure(N/m^2)
$Q^\wedge_s$	Dimensionless side leakage flow, $Q^\wedge_s = Q_s\,L/U_jR^2c$
$r^\wedge$	Normalized radial coordinate, $r^\wedge = r/r_i$
R_j	inner radius of porous bearing (m)
R_o	outer radius of porous bearing (m)
r_i	Inner Radius (m)
r_o	Outer Radius (m)
S	Sommerfeld Number, $S = (R\mu\omega_jL / W)*(r_i / c)^2$
U_j	Journal Velocity (m/s)
$W^\wedge$	Dimensionless Load Carrying Capacity, $W^\wedge = (W\,c^2 / \eta\omega_j\, \boldsymbol{r_i^3}\, L)$
$W^\wedge_r$	Dimensionless Component of Oil Film Force Along the Line of Centers
$W^\wedge_T$	Dimensionless Component of Oil Film Force Perpendicular to the Line of Centers
$y^\wedge$	Dimensionless bearing coordinates in axial direction, $y^\wedge = y/h$
$Z^\wedge$	non – dimensional axial co- ordinate $Z^\wedge = z/L$

Greek Symbols

ε	Eccentricity Ratio
μ	Absolute Viscosity of lubricant (pa . s)
η	material constant responsible for the couple stress parameter
θ	Angular Coordinate from Maximum Film Thickness Position (Degree)
ρ	Density of oil (kg/m^3)
δ	porous layer thickness (m)
Φ	Permeability parameter, $\Phi = k*\delta/c^3$
ψ	Attitude Angle (degrees)
r, θ, z	Bearing coordinates in radial, circumferential and axial directions.
γ_1, γ_2	tilt angles (rad)
σ_1, σ_2	two independent misalignment parameters
ξ	Normalized axial coordinate (z/L)
ω_j	journal rotational speed (rad/sec)

Subscript

b	Referring to Bearing
j	Referring to Journal

Superscript

^	Dimensionless Quantity
•	Porous Parameter

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تأثير انحراف المحور على الخصائص الساكنة للمساند المقعدية ذاتية التزييت المشبعة بمائع ذو أجهاد مزدوج

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الخلاصة

يتضمن هذا البحث دراسة نظرية لتأثير أنحراف المحور على الخصائص الساكنة للمساند المقعدية ذاتية التزييت المشبعة بمائع ذو أجهاد مزدوج. الموديل التحليلي المستخدم في البحث الحالي يختص بالمساند ذات النفاذية منتظمة التوزيع. مع الأخذ بنظر الاعتبار النفاذية المنتظمة للمسند المقعدي فان معادلة رينولدز عدلت لتأخذ بنظر الاعتبار تأثير النفاذية ونوعية الزيت (الزيت مزدوج الأجهاد). معادلة الضغط للطبقة المسامية استحصلت من معادلة دارسي ومعادلة الاستمرارية. المعادلة التي استخدمت لاحتساب سمك طبقة الزيت عدلت لتأخذ بنظر الاعتبار تأثير انحراف المحور بالاتجاهين الطولي والمستعرض. تم حل المعادلات الحاكمة مع الشروط الحدية المناسبة عدديا باستخدام تقنية عددية مناسبة. تم أعداد برنامج حاسوبي لحل المعادلات الحاكمة وتم التأكد من موثوقية البرنامج عن طريق مقارنة بعض النتائج المستحصلة في هذا البحث مع تلك المنشورة في بعض البحوث المتوفرة وقد أظهرت عملية المقارنة تقاربا جيدا في النتائج المستحصلة مع تلك المنشورة. ومن خلال مقارنة تصرف المساند المنحرفة والغير منحرفة وجد بأن لانحراف محور المسند المقعدي تأثيرا لا يمكن إهماله.