Effect of Shaft Misalignment on the Performance of Starved Porous Bearings

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Abstract:

Performance of starved porous journal bearings with misaligned shaft was investigated theoretically under hydrodynamic lubrication condition. The circumferential boundary condition for the oil film pressure is obtained by applying integral momentum equation to the oil film region in the bearing clearance. Two types of misalignment, axial (vertical displacement) and twisting (horizontal displacement), are considered. The analysis is performed on finite length , grooved misaligned bearing for starved inlet conditions. Effect of different degree of starvation and supply pressure are investigated for a particular misaligned condition. The results show that the journal misalignment and oil supply pressure have a marked effect on the static characteristics of the bearing.

الخلاصة

يتضمن البحث دراسة نظرية لأداء المساند المقعدية المسامية ذات الغشاء الزيتي الأعوز تحت ظروف أنحراف المحور الدوار وظروف التزييت الهيدروديناميكي. أستخدمت معادلة الزخم التكاملية لتحديد أمتداد طبقة الزيت. تم أعتماد نوعين من الأنحراف وهما ألأنحراف المحوري (الأزاحة العمودية) والأنحراف البرمي (الأزاحة الأفقية). أجريت الدراسة على مسند مقعدي محدود الطول ويغذى بالزيت عن طريق مجرى محيطي. تمت مناقشة تأثير ظروف عوز مختلفة لطبقة الزيت وضغط التجهيز ولحالة أنحراف محددة على اداء المسند. أظهرت الدراسة بأن أخذ أنحراف المحور الدوار وضغط تجهيز الزيت بعين الأعتبار له تأثير كبير على الأداء السكوني لهذا النوع من المساند.

Introduction

Journal bearings often operate in misaligned condition (the bearing and journal center–lines are not parallel). Since the film thickness is of the order of one – thousandth of the bearing radius, it is very likely that even small variations in operating conditions can cause considerable misalignment. The misalignment can result from non – central loading a moment produced by couple forces on the journal , elastic deflection of the shaft under load, thermal distortion of the shaft and the bearing housing supports, manufacturing error and / or improper installation.

Misalignment can vary in magnitude and direction. The most important cases to be considered are, axial (vertical misalignment), twisting or horizontal misalignment and combination of these can also occur.

An experimental investigation for the mechanism of lubrication in porous bearings was carried out by Kaneko and Obara (1990). The effect of dimensionless oil–feed pressure on frictional characteristics of porous bearings was studied experimentally by Kaneko and Hashimoto (1995). An analytical study of starved porous bearings has been carried out by Cusano (1979), Kaneko et. al.(1997). Kaneko et. al. shows experimentally that negative pressure exists before the trailing end of the oil film region in actual oil film pressure distributions.

Since it is rare that the hydrodynamic bearing is not misaligned then, studying the performance of fluid film bearing with misalignment is an important step toward a reliable design. Pinkus and Bnpara (1979) made an analysis to misaligned grooved journal bearing. They gave charts for a number of different cases which bring out some of the features of grooved bearing misalignment. Jany and Chang (1987) study the performance of misaligned journal bearings with non–Newtonian lubricant under adiabatic condition. Safer *et al.* (1988) made a prediction to the coefficient of friction of misaligned turbulent flow of journal bearing. An empirical formula for the predication of the coefficient of friction was established. Effect of cavitations on the performance of a grooved misaligned journal bearing had been studied by Vijayarraghav and Keith (1989). Effect of various degree of starvation and higher lubricant supply have been considered in this work. Buckholz and Lin (1986) made an analysis to the effect of journal bearing misalignment on load and cavitations for non–Newtonian lubricants. In this work the load–carrying capacity for partial arc journal bearings lubricated by power–law, non–Newtonian fluid is calculated for small values of bearing aspect ratios.

Stanislaw (2005) shows that the bearing successfully carry the extreme load in conditions of misaligned axis of journal and the bearing eliminating the necessity of using self aligning bearings. Vijayaraghavan and Brew (2005) study the effect of misalignment on the performance of planetary gear journal bearing. They discuss the effect of different degree of misalignment on bearing performance. Among the earlier theoretical and experimental contributions, the effect of bearing misalignment on the performance of porous bearings has been neglected.

Governing Equations and Bearing Geometry:

The porous bearing, gap – geometry is show in fig (1). Two independent angles (γ_1 and γ_2) are used to describe the fluid film gap. The non – dimensional fluid film gap given by Jin et. al. (1987) is adopted in this work:

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

Where,

$$\sigma_1 = 2\left(\frac{R}{C}\right)\left(\frac{L}{D}\right)\tan\gamma_1 \qquad \textbf{(2)}$$

$$\sigma_2 = 2\left(\frac{R}{C}\right)\left(\frac{L}{D}\right)\tan\gamma_2 \qquad \textbf{(3)}$$

The two independent misalignment angles (γ_1 and γ_2) are measured from ξ=0; the gap geometry depends on (θ and ξ). 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 the effect of tangential velocity slip (Bevers and Joseph, 1967). For laminar flow and constant viscosity it can be written in dimensionless form as (Kaneko et. al., 1997):

$$\frac{\partial}{\partial\theta}\left(h^{\wedge 3}(1+\zeta_1)\frac{\partial P^{\wedge}}{\partial\theta}\right) + \left(\frac{D_i}{L}\right)^2 \frac{\partial}{\partial Z^{\wedge}}\left(h^{\wedge 3}(1+\zeta_1)\frac{\partial P^{\wedge}}{\partial Z^{\wedge}}\right) = 6\frac{\partial}{\partial\theta}\left(h^{\wedge 3}(1+\zeta_0)\right) - 12\Phi_r\left(\frac{\partial P^{\wedge *}}{\partial r^{\wedge}}\Big|_{r^{\wedge}=1}\right) \qquad \textbf{(4)}$$

where,

$$\left.\begin{aligned}
(\zeta_0) &= \left(\frac{s}{(h^{\wedge} + s)}\right) \\
(\zeta_1) &= \left(3(h^{\wedge}s + 2\alpha^2 s^2)/\{h^{\wedge}(h^{\wedge} + s)\}\right) \\
(s) &= (\Phi c / r_i)^{1/2} / \alpha \\
(\Phi) &= (k r_i / c^3)
\end{aligned}\right\} \qquad \textbf{(5)}$$

The slip coefficient (α) is a dimensionless parameter depending on the material parameter which characterizes the structure of a permeable material within the boundary regions Its value for laminar channel flow has been estimated by Beavers and Joseph (1967) to be (0.1).

Velocity of the fluid flow in the porous matrix can be obtained from Darcy's law. The pressure of the oil flow inside the porous matrix can be governed by the darcy's equation which can be written as:

$$\frac{1}{r^{\wedge}}\frac{\partial}{\partial r^{\wedge}}\left(r^{\wedge}\frac{\partial P^{\wedge^*}}{\partial r^{\wedge}}\right)+\frac{1}{r^{\wedge^2}}\left(\frac{\partial^2 P^{\wedge^*}}{\partial \theta^2}\right)+\left(\frac{D}{L}\right)^2\left(\frac{\partial^2 P^{\wedge^*}}{\partial Z^2}\right)=0 \qquad \textbf{(6)}$$

Circumferential boundary condition for the oil film pressure:

The values of (θ_1 and θ_2) are constant in the direction of (z) (see fig.(1)). The inlet and trailing ends of the oil film are opened to the atmosphere giving ;

$$\left.\begin{array}{l} P^{\wedge}(\theta_1,Z)=P^{\wedge}(\theta_2,Z)=0 \\ P^{\wedge}(\theta,0)=P^{\wedge}(\theta,1)=0 \\ P^{\wedge^*}(r^{\wedge},\theta,0)=P^{\wedge^*}(r^{\wedge},\theta,1)=0 \\ P^{\wedge}(\theta,Z)=P^{\wedge^*}(r^{\wedge},\theta,Z)at(r^{\wedge})=1 \end{array}\right\} \qquad \textbf{(7)}$$

$$P^{\wedge^*}(r_o^{\wedge},\theta,1/2)=P_s^{\wedge}$$

The procedure used by Basim *et al.,* (2006) and (2007) was adopted to determine the values of (θ_1 and θ_2). The first circumferential boundary condition for the oil film is given by the dimensionless integral momentum

$$\bar{M}_{\theta 1}-\bar{M}_{\theta 2}-\bar{M}_{\theta c}-\bar{M}_{\theta b}=0 \qquad \textbf{(8)}$$

While the second circumferential boundary condition for the oil film is:

$$\int_0^1 \frac{\partial P}{\partial \theta} d\bar{Z}=0 \qquad \textbf{(9)}$$

Knowing the values of θ_1, and θ_2, the angular extent β of the oil film is expressed in the form

$$\beta=\theta_2-\theta_1$$

Bearing Parameters:

With the pressure field through the oil film is known, the bearing performance calculation can be carried out as follows.

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

$$\left.\begin{aligned}\left(\hat{W}_R\right) &= -\int_0^1\int_{\theta_1}^{\theta_2}\left(P^{\wedge}(\theta,\xi)\cos\theta\right)d\theta d\xi \\ \left(\hat{W}_T\right) &= \int_0^1\int_{\theta_1}^{\theta_2}\left(P^{\wedge}(\theta,\xi)\sin\theta\right)d\theta d\xi\end{aligned}\right\} \quad \textbf{(10)}$$

The total dimensionless load and the attitude angle (Ψ) are then evaluated as follows:

$$\left.\begin{aligned}\left(\hat{W}\right) &= \sqrt{\left(\hat{W}_R\right)^2 + \left(\hat{W}_T\right)^2} \\ (\Psi) &= \tan^{-1}\left(W_T^{\wedge}/W_R^{\wedge}\right)\end{aligned}\right\} \quad \textbf{(11)}$$

The friction force generated in the bearing clearance is made up of two parts (Cameron 1966). The first part is the solid film extending over the pressure curve which is taken as starting at ($\theta = \theta_1$) and ending at ($\theta = \theta_2$). The second part is the film pressure zone from θ_2 to ($2\pi+\theta_1$). Hence the coefficient of friction is calculated by Kaneko (1994);

$$\mu(\frac{r}{c}) = S\int_0^1\int_{\theta 1}^{\theta 2}\left[\frac{\bar{h}}{2}(1+\frac{1}{3}\xi_1)\frac{\partial \bar{P}}{\partial\theta} + \frac{1}{\bar{h}}(1-\xi_o)\right]d\theta d\bar{Z} + S[\bar{h}(1+\xi_o)|\theta=\theta_2]\int_{\theta 2}^{2\pi+\theta_1}\frac{1}{\bar{h}^2}(\frac{1-\xi_o}{1+\xi_o})d\theta$$

(12)

Method of Solution:

To study the effect of journal misalignment on the performance of porous bearings, equation (1), (4) and (6) have been solved simultaneously satisfying the given boundary conditions using finite differences technique. Gauss Siedel iterative scheme with successive under relaxation was used to determine the oil pressure distribution through the oil film and the porous matrix. To obtain the pressure and the location of the inlet and trailing boundary lines for the oil – film region, the iteration are continued until the following inequalities are satisfied simultaneously.

$$\left(\frac{\sum\sum\sum\left|P_{i,j,k}^{\wedge^*(n+1)} - P_{i,j,k}^{\wedge^*(n)}\right|}{\sum\sum\sum\left|P_{i,j,k}^{\wedge^*(n)}\right|} < 10^{-5}\right) \quad \textbf{(13)}$$

$$\left(\frac{\sum\sum\left|P_{j,k}^{\wedge(n+1)} - P_{j,k}^{\wedge(n)}\right|}{\sum\sum\left|P_{j,k}^{\wedge(n)}\right|} < 10^{-5}\right) \quad \textbf{(14)}$$

$$\left(\left|\mathrm{M}^{\wedge}_{\theta_1} - \mathrm{M}^{\wedge}_{\theta 2} - \mathrm{M}^{\wedge}_{\theta c} - \mathrm{M}^{\wedge}_{\theta_b} / \mathrm{M}^{\wedge}_{\theta_1}\right| < 10^{-3}\right) \quad \textbf{(15)}$$

$$\left(\left|q_{\theta_p} / q_{\theta_c}\right|\right) = \left(\left|\left(\frac{(1+\zeta_{1\theta})}{6(1+\zeta_{0\theta})} h^{\wedge^2} \int_0^1 \frac{\partial P^{\wedge}}{\partial \theta} dz^{\wedge}\right)_{\theta=\theta_2}\right| < 10^{-3}\right) \quad \textbf{(16)}$$

In above equations, (n) and (n+1) denote two consecutive iterations and $P^{\wedge *}_{i,j,k}$ is the nodal pressure at point (i,j,k), in which the signs (i,j,k) represent the grid number in radial, circumferential, and axial directions respectively. A computer program was prepared and written in Fortran – 90 has been used to solve the governing equations of the problem.

Results and Discussion

Figures (2) through (12) present the numerical results obtained during this work. The computer program prepared in this work was tested by comparing the results obtained with that published by Kaneko et. al.(1997)as shown in figure (2). It is clear that there is a good agreement between the obtained and the published results
Four cases represent different misalignment conditions are adopted in the present work, namely:

Case 1: $\gamma_1 = 0, \gamma_2 = 0$ (Aligned journal)
Case 2: $\gamma_1 = 0.0001, \gamma_2 = 0$ (Axial misalignment)
Case 3 $\gamma_1 = 0, \gamma_2 = 0.0002$ (Twisting misalignment)
Case 4: $\gamma_1 = 0.0001, \gamma_2 = 0.0002$ (Combined of axial and twisting misalignment)

Typical film pressure distribution are shown in fig.(3) through fig.(5) for various cases of journal misalignment and specific values of supply pressure and eccentricity ratio. It can be shown from fig.(1) that the pressure decreases when axial (case 1) misalignment taken into consideration. Figures (4) and (5) shows that the maximum pressure increases for cases (3) and (4) indicating that the minimum oil film thickness decreases in these cases. Fig.(5) also show that there is a slight increases in pressure in case (4) in comparison with case (3). The effect of journal misalignment on the oil film extent can be shown in figures (6) for different misalignment cases. It is clear that there is decreases in oil film extent when axial journal misalignment taken into consideration while there is increases in oil film for the other two cases which explains the behavior of pressure curves described before. It can be noticed that the Sommerefeld number increases when the journal axial misalignment taken into consideration than that of alignment case since the load capacity is decreased in this case. The Sommerfeld number is reduced when the twisting misalignment is considered in comparison with that of alignment case. A greater decreases in Sommerefeld number can be shown when both axial and twisting misalignment of the journal are considered in comparison with that of alignment case.

An increase in coefficient of friction has been shown when the axial misalignment is considered when compared with that obtained in alignment case as shown in fig. (7). This is due to the decreases in oil film extent in this cases as discussed before. It is also shown that the coefficient of friction decreases when the journal twisting misalignment is considered when compared with that obtained in alignment case as shown in fig. (8). More decreases in coefficient of friction has been shown when compared with that obtained with the alignment case as shown in fig. (9). It has been shown that the coefficient of friction was not affected for higher value of eccentricity ratio.

Increase in attitude angle for the bearing of journal with axial misalignment as compared with that of aligned journal can be shown in fig. (10). This is due to the decreases in load carrying capacity shown in this case as discussed before. The attitude angle shown to be decreases for the bearing with twisted misalignment journal as shown in fig.(11). More decreases has been noticed for the bearing with journal of both axial and twisting misalignment as compared with the bearing of aligned journal as shown in fig.(12).The effect of supply pressure on the performance of porous bearing can be shown in figures (13), (14). It is clear from figure (13) that the coefficient of friction decreases as the supply pressure increases. This is due to the increase in oil film extent which is consistence with the results obtained by other workers, Kaneko et. al. (1994,1997). The attitude angle increases as the supply pressure increases as shown in figure (14). This is due to the increase of the load carried by the bearing.

Conclusions

The static characteristics under hydrodynamic lubrication condition and journal misalignment have been analyzed for porous journal bearings. Numerical solution was obtained based on starved oil film, and different cases of journal misalignment. The journal misalignment has a marked effect on the static characteristics. Including an axial journal misalignment causes a decrease in oil film extent, increase in Sommerfeld number, increase in friction coefficient and attitude angle when compared with the alignment case. However a slight increase in oil film pressure increase in oil film extent, decrease in Sommerfeld number ,decrease in coefficient of friction and attitude angle has been shown when twisting and combined misalignment were considered in comparison with the alignment case. The adequacy of the present analysis has been confirmed by comparing a sample of results obtained in this work with that published by other worker. The oil supply pressure seems to have a considerable effect on the performance of the porous bearings.

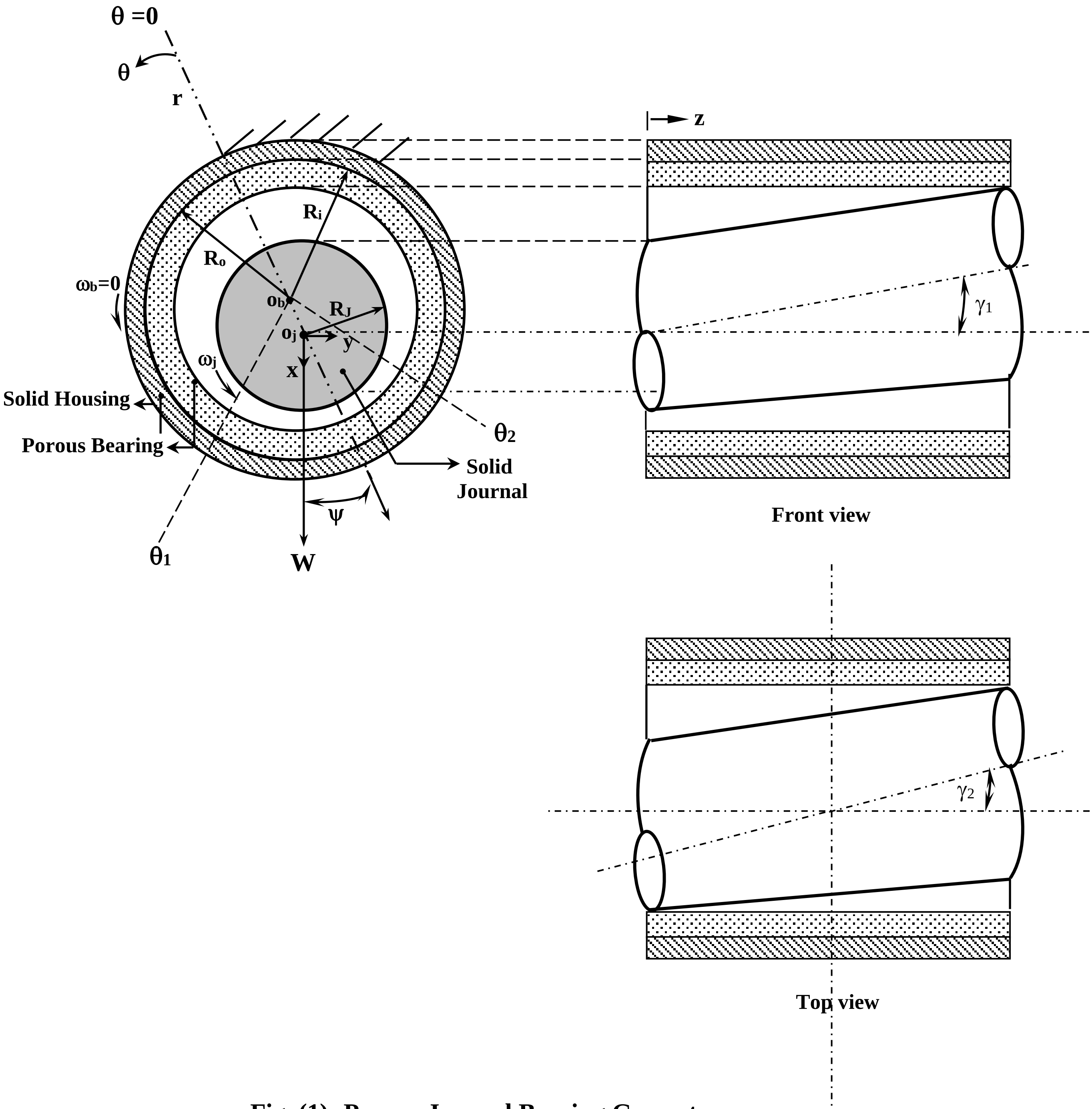
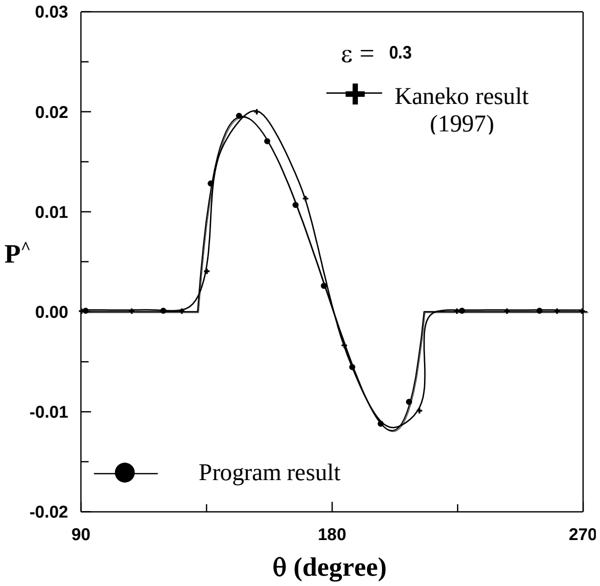
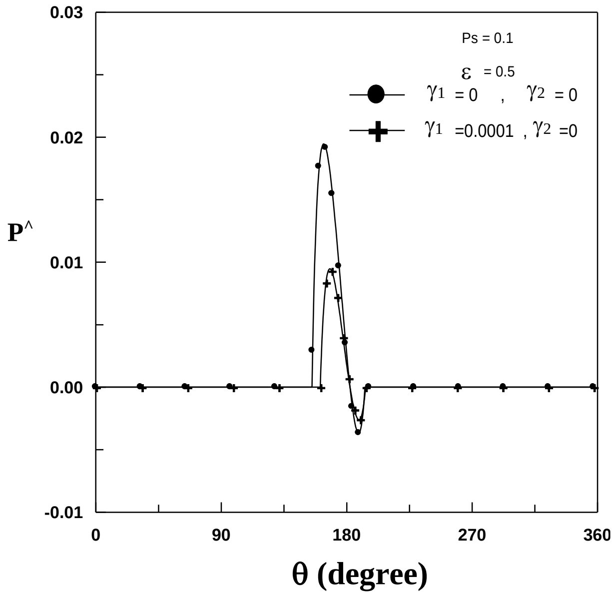


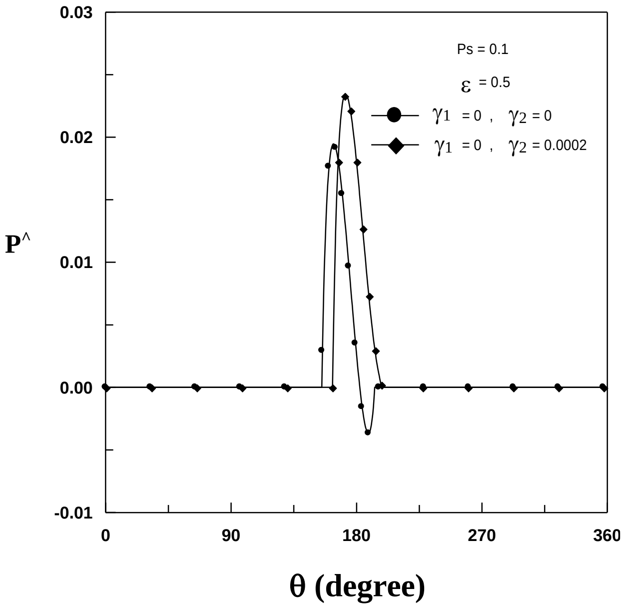
Fig. (1): Porous Journal Bearing Geometry



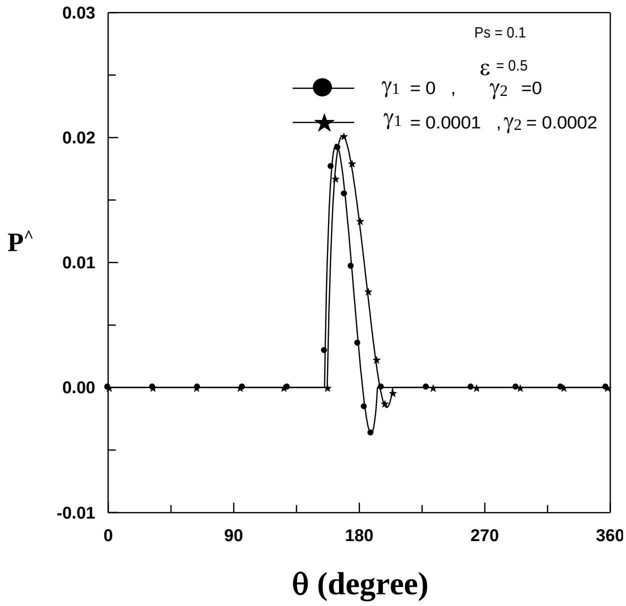
Fig(2): Comparison between obtained results and that published by Kaniko (1997)



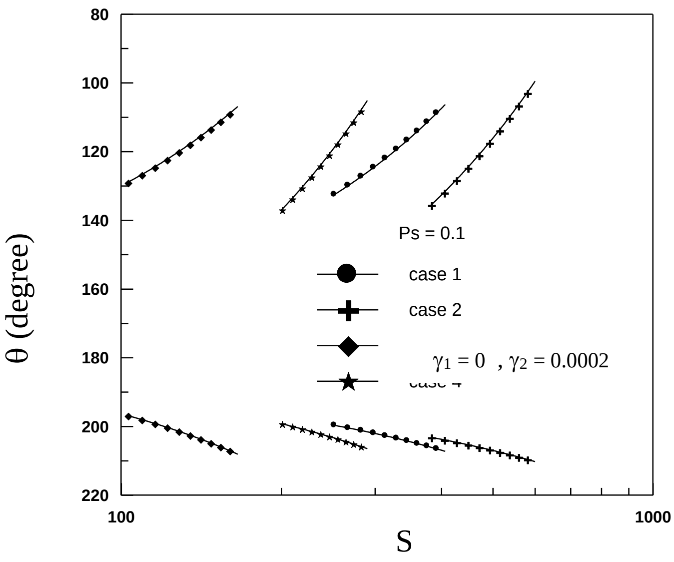
Fig(3): Effect of journal axial misalignment on pressure distribution of the bearing



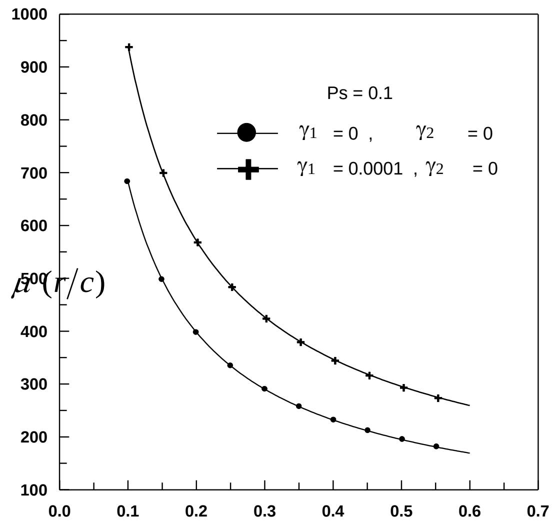
Fig(4): Effect of journal twisting misalignment on the pressure distribution of the bearing



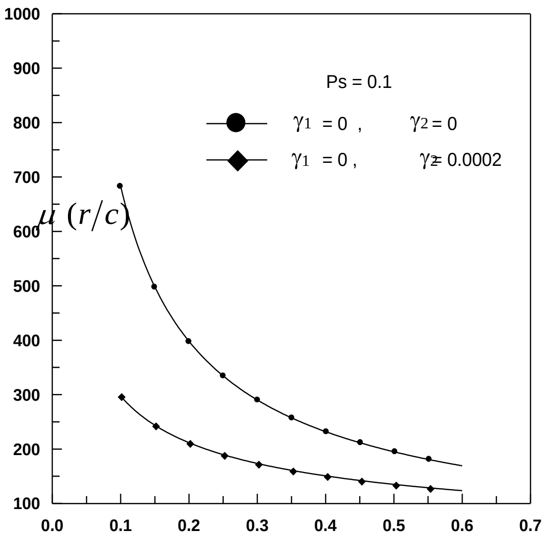
Fig(5): Combined Effect of journal axial and twisting misalignment on the pressure distribution of the bearing



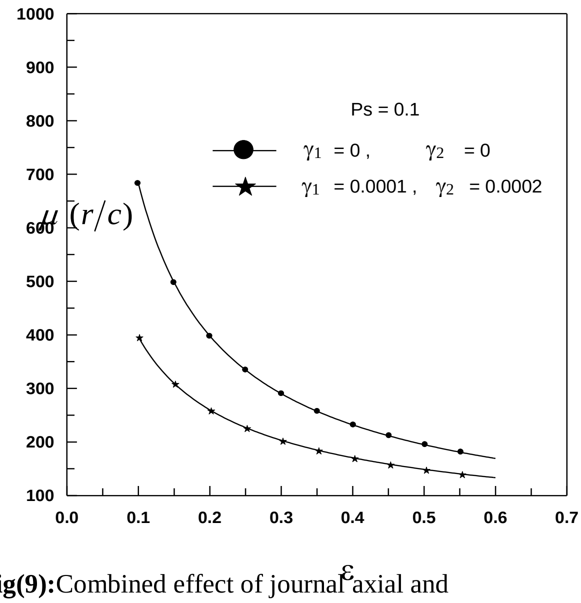
Fig(6): Effect of different journal misalignment on the oil film extent.



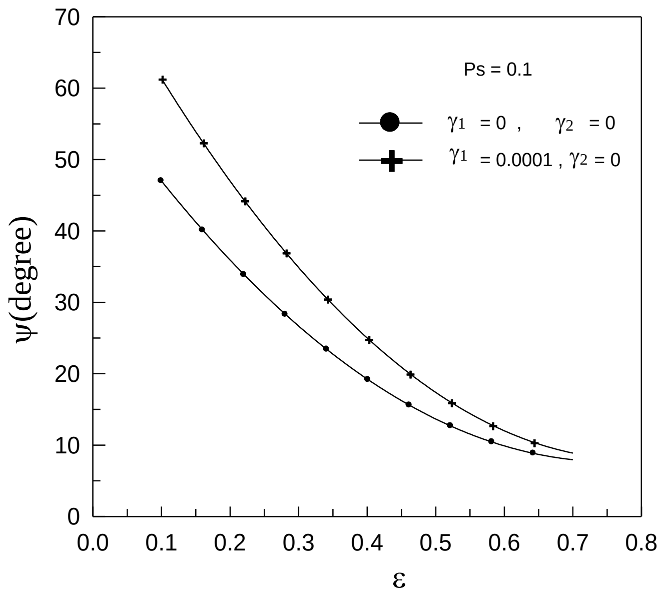
Fig(7): Effect of journal axial misalignment on the coefficient of friction.



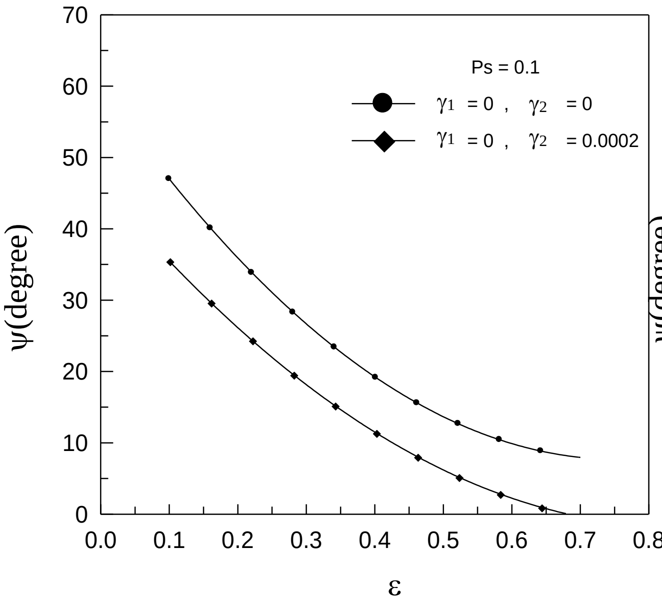
Fig(8): Effect of journal twisting misalignment on the coefficient of friction of the bearing



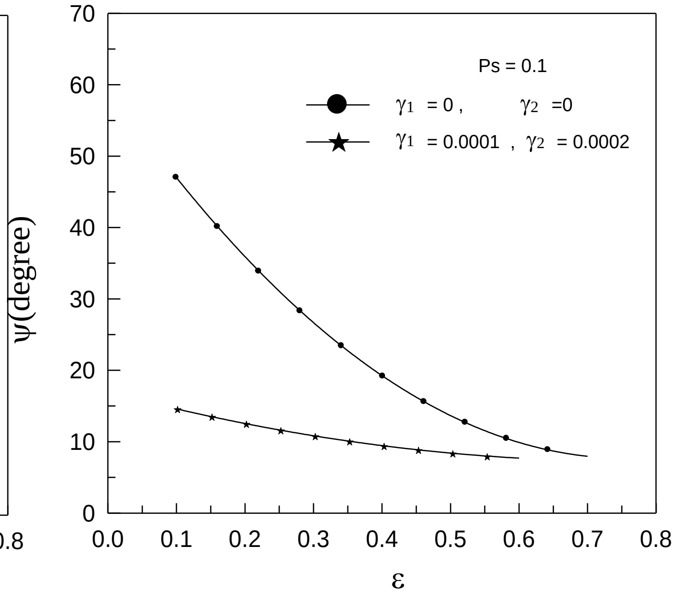
Fig(9): Combined effect of journal axial and twisting misalignment on the coefficient of friction of the bearing



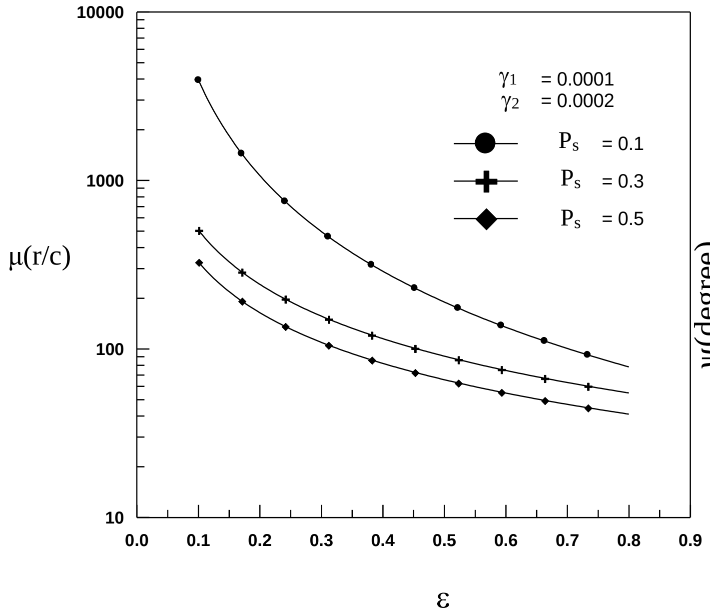
Fig(10): Effect of journal axial misalignment on the attitude angle of the bearing



Fig(11): Effect of journal twisting misalignment on the attitude angle of the bearing



Fig(12): Combined Effect of journal axial and twisting misalignment on the attitude angle of the bearing.



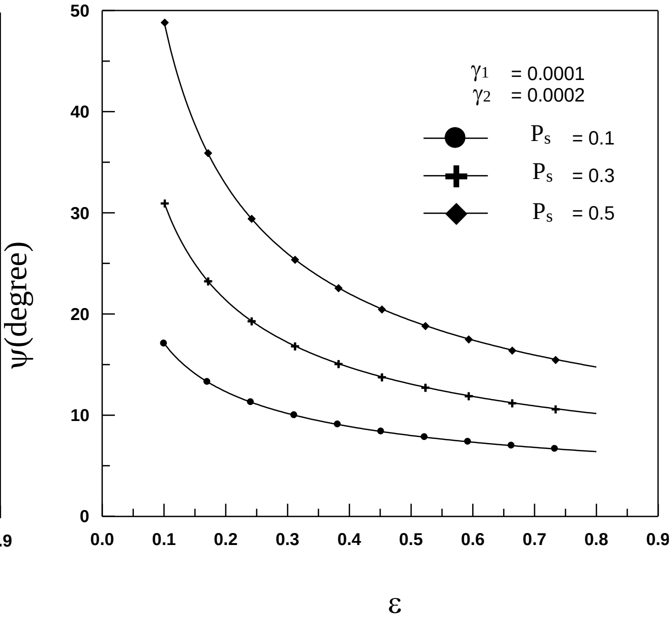


Fig (13): Effect of supply pressure on the friction coefficient

Fig (14): Effect of supply pressure on the attitude angle

References:

Basim A. Abass, Alaa' M. Hussain, and Likaa' Hameed, **(2007)**, "Operating characteristic of porous floating ring journal bearings operating with improved boundary conditions", Journal of Engineering, vol.13, no.1, pp.1296-1311

Beavers, G.S., and Joseph, D.D., (1967), "Boundary conditions at naturally permeable wall", Journal of fluid mechanics, vol. 30, Part 1, Oct., pp. 197-207

Buckholz R. H.and Lin, J.F. (1986), " The Effect of Journal Bearing Misalignment on Load and Cavitations for Non Newtonian Lubricants", Journal of Tribology, vol. 108, October, pp. 645 – 654.

Cusano, C. (1979), "An Analytical Study of Starved Porous Bearings" Transactions of the ASME, vol. 101, January, pp. 38 – 47.

Jin Y. Jang and Chong–Ching Chang, (1987), "Adiabatic Solutions for a Misaligned Journal Bearing with non–Newtonian Lubricants", Tribology International, vol. 20, no. 5, pp. 267 – 275

Kaneko, S. and Hashimoto, Y. (1995), "A Study of the Mechanism of Lubrication in Porous Journal Bearings: Effect of Dimensionless Oil Feed Pressure on Frictional Characteristics", Journal of Tribology, vol., 119, April, pp. 291- 296.

Kaneko, S. Hashimoto Y. and Hojki I. (1997), "Analysis of Oil – Film Pressure Distribution in Porous Journal Bearings Under Hydrodynamic Lubrication Conditions Using an improved Boundary Condition", Journal of Tribology, vol. 119, January, pp. 171 – 178.

Kaneko, S. Obara, S. (1990), "Experimental Investigation of Mechanism of Lubrication in Porous Journal Bearings Part 1 – Observation of Oil Flow in Porous Matrix", Transactions of the ASME, vol. 112, October, pp. 618 – 623.

Kaneko, S. Ohkawa Y. and Hashimoto, Y. (1994), "A Study of the Mechanism of Lubrication in Porous Journal Bearings: Effect of Dimensionless Oil Feed

Pressure on Static Characteristics on Static Hydrodynamic Lubrication Conditions", Journal of Tribology, vol. 116, July, pp. 606 – 611.

Oscar Pinkus and Bupara, S.S. (1979), "Analysis of Misaligned Grooved Journal Bearings", Transaction of the ASME, vol. 101, October, pp. 503 – 508.

Safar Z. S. and Riad, M. S. M. (1988), "Prediction of the Coefficient of Friction of a Misaligned Turbulent Flow Journal Bearing", Tribology International, vol. 21, No. 1, February , pp. 15 – 22.

Stanislaw Strzelecki, (2005), "Operating Characteristics of Heavy Loaded Cylindrical Journal Bearing with Variable Axial Profile", Materials Research, vol. 8, No. 4, pp. 481 – 486.

Vijaraghavan D. and Brew, D.E. (2005), "Effect of Misalignment on the Performance of Planetary Gear Journal Bearings", Brew Papers \ Misalignment Paper\ Misalign Pap. Wpd, April 29.

Vliayaraghav D. and Keith, T.G. (1989), "Effect of Cavitations on the Performance of a Grooved Misaligned Journal Bearing" Wear, vol. 134, pp. 377 – 397

NOMENCLATURE:

The following symbols are used throughout this work.

c Journal Bearing Clearance (m)

D diameter of bearing (m)

F friction force (N)

$h^\wedge$ Dimensionless Film Thickness,($h^\wedge = h/c$)

k Permeability of the Porous Matrix (m^2)

L Length of the Bearing (m)

$M_{\theta 1}$ Circumferential Momentum Flow Rate across Oil Film Surface at Inlet End of Oil – Film Region, i.e. at $\theta = \theta_1$

$M_{\theta 2}$ Circumferential Momentum Flow Rate across Oil – Film Surface at Trailing End of Oil–Film Region, i.e. at $\theta = \theta_2$

$M_{\theta c}$ Circumferential Momentum Flow Rate across Oil–Film Surface at Both Axial Ends.

$M_{\theta b}$ Circumferential Momentum Flow Rate across Oil – Film Surface Adjacent to Inner Surface of the Bearing, i.e. (y=0)

N_j Journal Rotational Speed (r.p.m)

$P^\wedge$ Dimensionless Oil-Film Pressure,$P^\wedge = c^2P/(r^2\, \eta\omega)$

$P^{\wedge *}$ Dimensionless Oil – Film Pressure Inside the Porous Matrix,$P^{\wedge *} = c^2P^*/(r^2\eta\omega)$

P_s Supply Pressure (N/m^2)

$r^\wedge$ Normalized radial coordinate, $r^\wedge = r/r_i$

R_j Journal Radius(m)

r_i Inner Radius(m)

r_o Outer Radius(m)

(S) Sommerfeld Number , $S = (R\eta\omega_j L / W)*(r_i / c)^2$

s Slip parameter

U_j Journal Velocity (m/s)

$W^\wedge$ Dimensionless Load Carrying Capacity, $W^\wedge = W c^2 / \eta\omega_j\, r_i^3 L$

($W^\wedge_r$) 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

Greek Symbols

ε Eccentricity Ratio
η Absolute Viscosity of Oil(pa . s)
θ Angular Coordinate from Maximum Film Thickness Position (Degree)

$\mu^{\wedge}$ Dimensionless Friction Coefficient $\mu^{\wedge} = (R/c)\mu$
ρ Density of oil (kg/m^3)
Φ Permeability parameter (m^2).
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)

Subscript

b Referring to Bearing
j Referring to Journal

Superscript

^ Dimensionless Quantity
* Porous Parameter