

AN INVESTIGATION TO THE COMBINED EFFECT OF NON-UNIFORM PERMEABILITY AND SHAFT MISALIGNMENT ON THE PERFORMANCE OF POROUS BEARINGS

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ABSTRACT

A porous bearing with misaligned journal and non uniform permeability has been investigated through the present work. The Reynolds' boundary condition has been used to determine the boundary of the effective oil film. Two types of misalignment, axial (vertical displacement) and twisting (horizontal displacement) with different values have been adopted. Two models are considered to analyze a finite porous bearing with misaligned shaft and non uniform permeability. In the first model the permeability a cross the radial direction has been varied, while in the other model the axial permeability has been varied. Effect of different supply pressure and misalignment are investigated for a particular bearing permeability. It was found that there is a slight effect to the supply pressure on the performance parameters of the porous bearing. Varying the permeability in radial and axial direction cause an increase of load carrying capacity with slight increase in coefficient of friction. It is also noticed that the journal misalignment has a considerable effect on the bearing performance.

Keywords:Hydrodynamic Lubrication, Porous Oil Bearing, Misaligned Journal Bearing, Non-Uniform Permeability, Static Characteristics.

الخلاصة :

تضمن هذا البحث دراسة للمساند المقعدية ذاتية التزيت ذات النفاذية المتغيرة والمحور المنحرف. تم استخدام الشرط الحدي لرينولدز لتحديد حدود طبقة الزيت. تم اعتماد نوعين من الانحراف وهما الانحراف المحوري (العمودي) والانحراف البرمي. اعتمد نموذجان لدراسة تأثير تغير النفاذية باتجاهي النصف قطري والمحوري للمساند المسامية المحددة الطول والمحور المنحرف. لقد وجد إن لضغط تجهيز الزيت تأثير طفيف على خصائص المساند المسامية، وان الحمل الهيدروديناميكي يزداد بتغير النفاذية باتجاهي النصف قطري والمحوري مع زيادة طفيفة جدا بمعامل الاحتكاك. كذلك تم ملاحظة أن للمحاور المنحرفة تأثير على خصائص المسند.

Jang and Chang (1987) presented an adiabatic solution for a finite width hydrodynamic misalignment journal bearing with non Newtonian lubricants obeying the power law model. Both vertical and horizontal misalignment has been taken into consideration during this work.

A modified version of the cavitations algorithm which automatically predict film rupture and reformation in bearings is used to analyze misaligned grooved journal bearing by Vijayaragha Van and Keith (1989).

Stanislaw (2005) shows that the bearing successfully carry the extreme load in conditions of misaligned axis of journal and the bearing eliminatory 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 combined effect of bearing misalignment and non-uniform permeability on the performance of porous bearings have 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 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 velocity slip (Bever and Joseph, 1967). For laminar flow and constant viscosity the Reynolds" equation can be written in dimensionless form as Kaneko (1989):

$$\frac{\partial}{\partial \theta} \left(h^3 (1 + \zeta_{1\theta}) \frac{\partial P^*}{\partial \theta} \right) + \left(\frac{D_i}{L} \right)^2 \frac{\partial}{\partial Z} \left(h^3 (1 + \zeta_{1Z}) \frac{\partial P^*}{\partial Z} \right) = 6 \frac{\partial}{\partial \theta} \left(h^3 (1 + \zeta_{0\theta}) \right) - 12 \Phi_r \left(\frac{\partial P^*}{\partial r} \right)_{r=1} \quad (1)$$

where,

$$\left. \begin{aligned} (\zeta_{0\theta}) &= \left(\frac{s_\theta}{h + s_\theta} \right) \\ (\zeta_{1\theta}) &= \left(3(h s_\theta + 2\alpha^2 s_\theta^2) / \{h (h + s_\theta)\} \right) \\ (\zeta_{1Z}) &= \left(3(h s_Z + 2\alpha^2 s_Z^2) / \{h (h + s_Z)\} \right) \end{aligned} \right\} \quad (2)$$

The terms $(\zeta_{0\theta})$, $(\zeta_{1\theta})$ and (ζ_{1Z}) result from the tangential velocity slip, where,

$$\left. \begin{aligned} (s_\theta) &= (\Phi_\theta c / r_i)^{1/2} / \alpha \\ (s_Z) &= (\Phi_Z c / r_i)^{1/2} / \alpha \\ (\Phi_r) &= (k_r r_i / c^3) \\ (\Phi_\theta) &= (k_\theta r_i / c^3) \\ (\Phi_Z) &= (k_Z r_i / c^3) \end{aligned} \right\} \quad (3)$$

The nondimensional fluid film gap given by Jin et.al. (1987) is adopted in this work:

$$\hat{h} = 1 + \varepsilon \cos \theta - \xi \sigma_1 \cos \theta + \xi \sigma_2 \sin \theta \quad (4)$$

Where,

$$\sigma_1 = 2\left(\frac{R}{C}\right)\left(\frac{L}{D}\right) \tan \gamma_1 \quad (5)$$

$$\sigma_2 = 2\left(\frac{R}{C}\right)\left(\frac{L}{D}\right) \tan \gamma_2 \quad (6)$$

The two independent misalignment angles (γ_1 and γ_2) are measured from $\xi=0$; the gap geometry depends on (θ and ξ).

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 Kaneko and Doi (1989);

$$\frac{1}{r^\wedge} \frac{\partial}{\partial r^\wedge} \left(r^\wedge \Phi_r \frac{\partial P^\wedge}{\partial r^\wedge} \right) + \frac{1}{r^\wedge{}^2} \frac{\partial}{\partial \theta} \left(\Phi_\theta \frac{\partial P^\wedge}{\partial \theta} \right) + \left(\frac{D_i}{L} \right)^2 \frac{\partial}{\partial Z^\wedge} \left(\Phi_Z \frac{\partial P^\wedge}{\partial Z^\wedge} \right) = 0 \quad (7)$$

The permeability of the porous bearing has been varied in radial and axial direction as shown in figure (1).

Boundary conditions for the oil film pressure:

The pressure boundary conditions can be expressed as:

$$\left. \begin{aligned} P^\wedge(\theta, 0) &= P^\wedge(\theta, 1) = 0 \\ P^\wedge(r^\wedge, \theta, 0) &= P^\wedge(r^\wedge, \theta, 1) = 0 \\ P^\wedge(\theta, Z) &= P^\wedge(r^\wedge, \theta, Z) \text{ at } (r^\wedge) = 1 \\ P^\wedge(r^\wedge, \theta, 1/2) &= P_s \\ \hat{P} = \frac{\partial P^\wedge}{\partial \theta} &= 0 \text{ at } \theta = \pi + \alpha_c \end{aligned} \right\} \quad (8)$$

Where α_c the angle at which cavitation starts.

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 as :

$$\left. \begin{aligned} \left(W_R^\wedge \right) &= - \int_0^1 \int_{\theta_1}^{\theta_2} (P^\wedge(\theta, \xi) \cos \theta) d\theta d\xi \\ \left(W_T^\wedge \right) &= \int_0^1 \int_{\theta_1}^{\theta_2} (P^\wedge(\theta, \xi) \sin \theta) d\theta d\xi \end{aligned} \right\} \quad (9)$$

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

$$\left. \begin{aligned} \left(\hat{W} \right) &= \sqrt{\left(\hat{W}_R \right)^2 + \left(\hat{W}_T \right)^2} \\ \left(\Psi \right) &= \tan^{-1} \left(\hat{W}_T / \hat{W}_R \right) \end{aligned} \right\} \quad (10)$$

The friction force can be evaluated as;

$$F_r^{\wedge} = \int_0^1 \int_0^{\pi+\alpha_c} \frac{h^{\wedge}}{2} \frac{\partial P^{\wedge}}{\partial \theta} d\theta dz^{\wedge} + \int_0^1 \int_0^{\pi+\alpha_c} \frac{h^{\wedge}}{2} \frac{\zeta_{1\theta}}{3} \frac{\partial P^{\wedge}}{\partial \theta} d\theta dz^{\wedge} + \int_0^1 \int_0^{\pi+\alpha_c} \frac{(1-\zeta_{0\theta})}{h^{\wedge}} d\theta dz^{\wedge} \quad (11)$$

The coefficient of friction can be evaluated as:

$$f\left(\frac{r}{c}\right) = \frac{F_r^{\wedge}}{W^{\wedge}} \quad (12)$$

METHOD OF SOLUTION:

To study the effect of journal misalignment on the performance of porous bearings, equation (1), (4) and (7) have been solved simultaneously satisfying the given boundary conditions (8) 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 trailing boundary for the oil – film region, iterations are continued until the following inequalities are satisfied simultaneously.

$$\left(\frac{\sum \sum \sum \left| P_{i,j,k}^{\wedge*(n+1)} - P_{i,j,k}^{\wedge*(n)} \right|}{\sum \sum \sum \left| P_{i,j,k}^{\wedge*(n)} \right|} < 10^{-5} \right) \quad (12)$$

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

In the above equations, (n) and (n+1) denote two consecutive iterations and $P_{i,j,k}^{\wedge*}$ 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. The computer program can be explained through the flow chart shown in fig(2).

RESULTS AND DISCUSSION:

The results obtained in this work were presented in figure (3) through figure (14). The computer program presented to solve the governing equations of the

problem has been tested by comparing the results obtained by this program with that published by Kaneko et. al. (1994) as shown in figure(3).

It is clear that the obtained results from the present work is reasonably agree with the published results. The maximum deviation between the results was found to be (7%). This is due to the different numerical approaches used in both works.

Atypical pressure curves to the oil film have been shown in figures (4) and (5). It can be deduced from these figures that a slight increase in oil film pressure was obtained when twisting misalignment of the journal was taken into consideration. The maximum pressure was seen to be moved toward the outlet of the oil film . This indicate that the oil film thickness decreases in this case. A drastic reduction in oil pressure was seen when combined axial and twisting journal misalignment were taken into consideration for a certain value of supply pressure. The pressure seen to have a higher value for a higher values of eccentricity ratio.

The effect of misalignment angle variations axial (vertical) and twisting (horizontal) on Sommerfeld number can be shown in figures (6 and 7). The general misalignment effect can be deduced from these figures. Increasing the degree of tilt in axial misalignment causes nearly linear decreases in Sommerfeld number. Drastic drop in Sommerfeld number was shown for higher values of bearing eccentricity ratio.

The oil supply pressure seems to have a little effect on the performance of the bearing as can be shown from figure (8) and (9). It is obvious that the effect of supply pressure became more clear as the axial and twisting misalignment become greater as shown in figure (8). A slight increase in coefficient of friction was seen when axial misalignment of the journal was taken into considerations as shown in figure (10), while a decreases in coefficient of friction was seen when twisting misalignment of the journal was taken into consideration. The decreases in coefficient of friction become higher when the combined effect of axial and twisting journal misalignment are taken into consideration.

Figure(11) shows that the attitude angle increases as the axial misalignment of the journal taken into consideration, while it decreases when the twisting and combined journal misalignment taken into consideration. All the results are compared with that of aligned journal.

The effect of non uniform permeability in both axial and radial directions has been shown in figures(12 to 14). A decrease in Sommerfeld number (bearing parameter) has been shown which indicates an increase in load carrying capacity of the bearing when the permeability of the porous bearing has been changed in axial or in radial direction. It can also be noticed that there is slight difference in results when the permeability was changed in axial or in radial directions.

Slight increase has been noticed in coefficient of friction when variable permeability has been considered. An increase in attitude angle has been noticed for low eccentricities while, a decrease in attitude angle has been noticed for higher values of eccentricity ratio as shown in figure (14).

CONCLUSIONS:

From the above discussion the following concluding remarks can be traced.

- 1- An increase in load carrying capacity of the bearing when the permeability of the porous matrix has been changed in axial and radial direction.
- 2- Decrease in attitude angle for higher values of eccentricity ratio when the permeability of the porous matrix has been changed in axial and radial direction.

- 3- Slight increase in attitude angle has been noticed for the bearing with non uniform permeability in comparison with that of uniform permeability.
- 4- It seen that the misalignment has a marked effect on the static characteristic of the bearing, slight increase in oil film pressure, 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. s

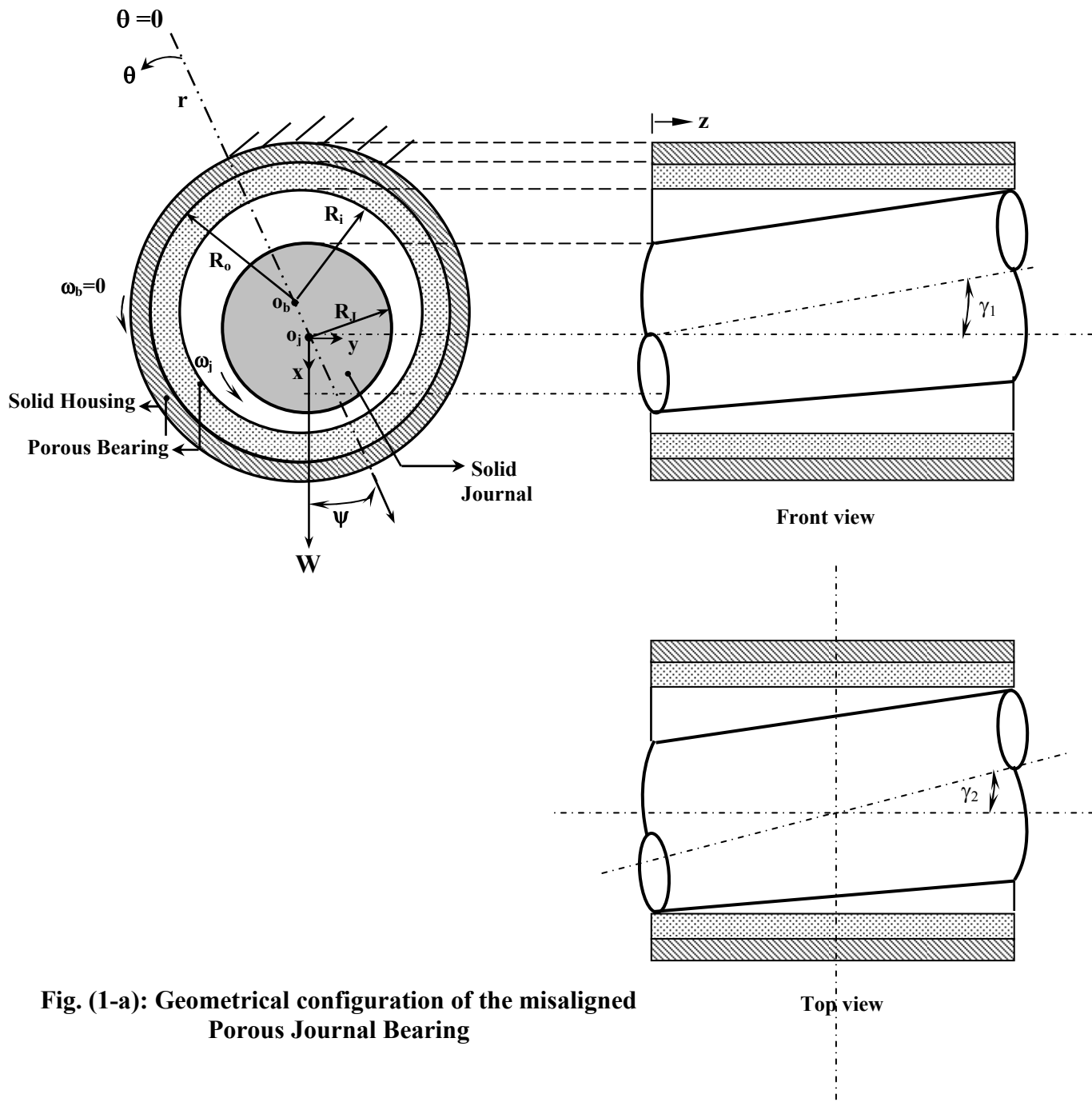
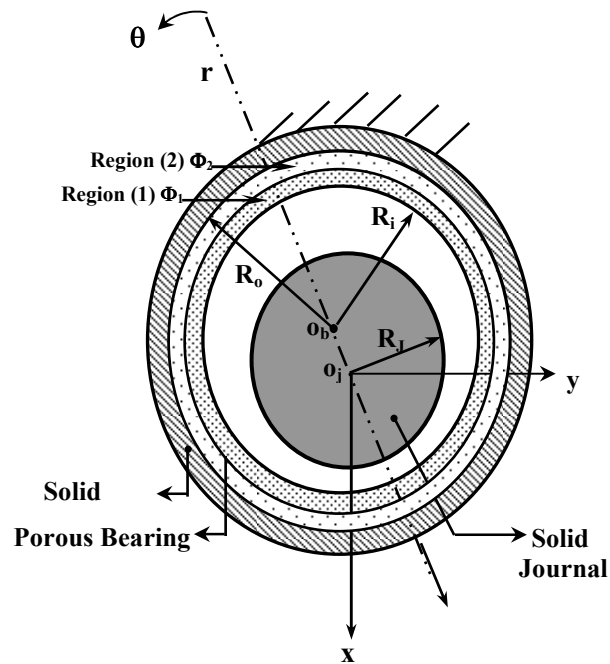
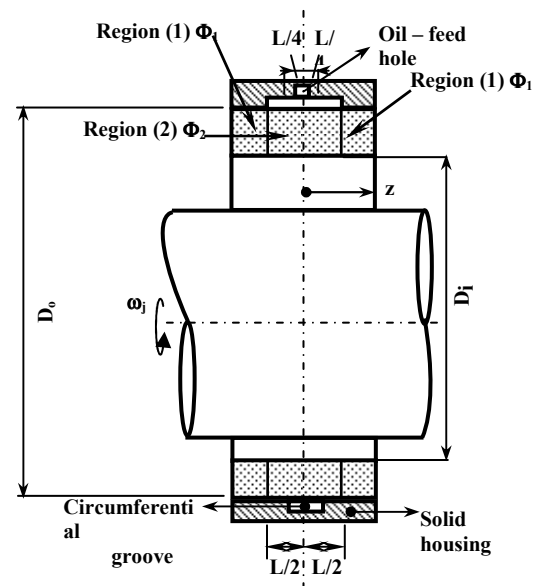


Fig. (1-a): Geometrical configuration of the misaligned Porous Journal Bearing

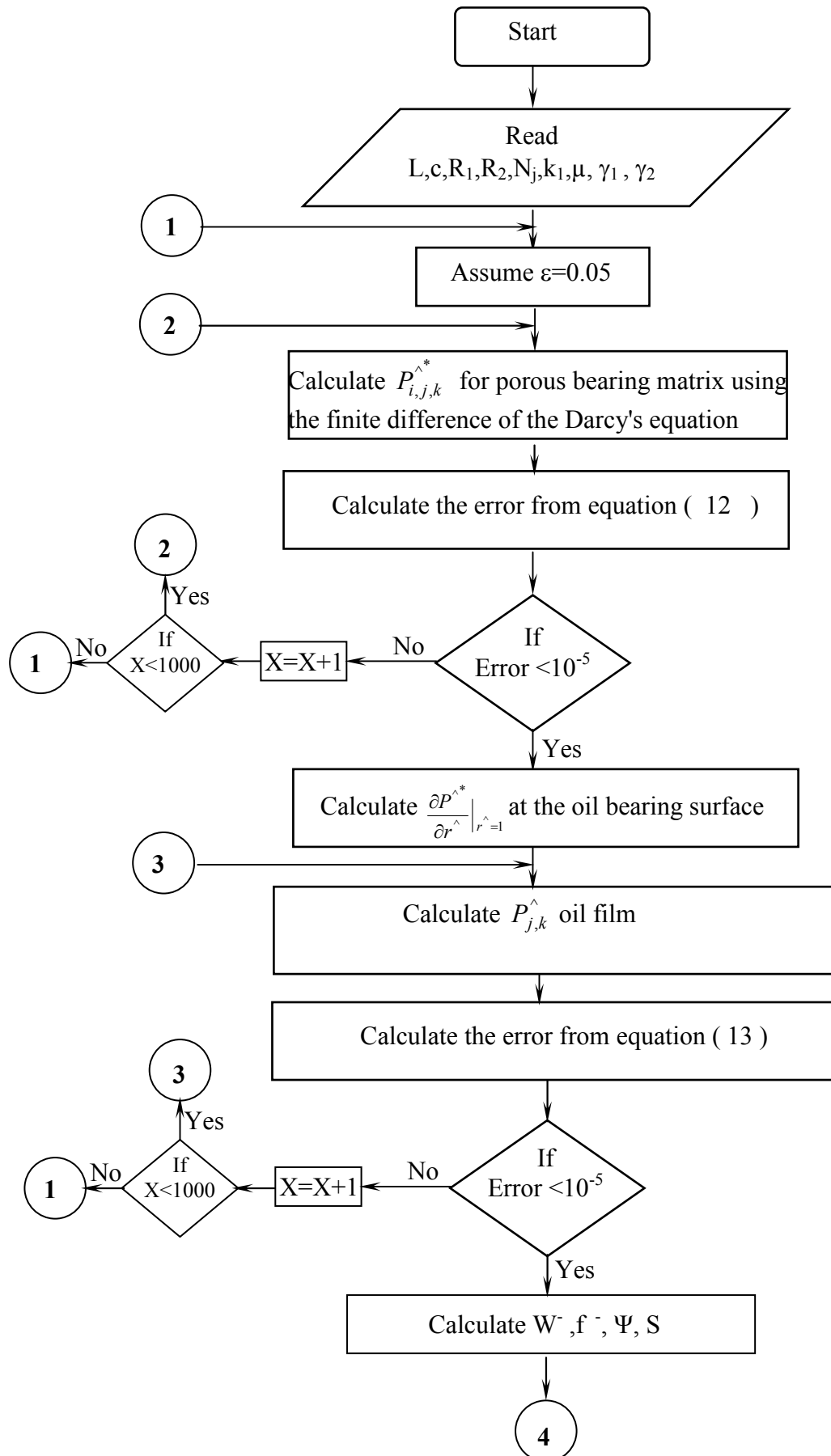


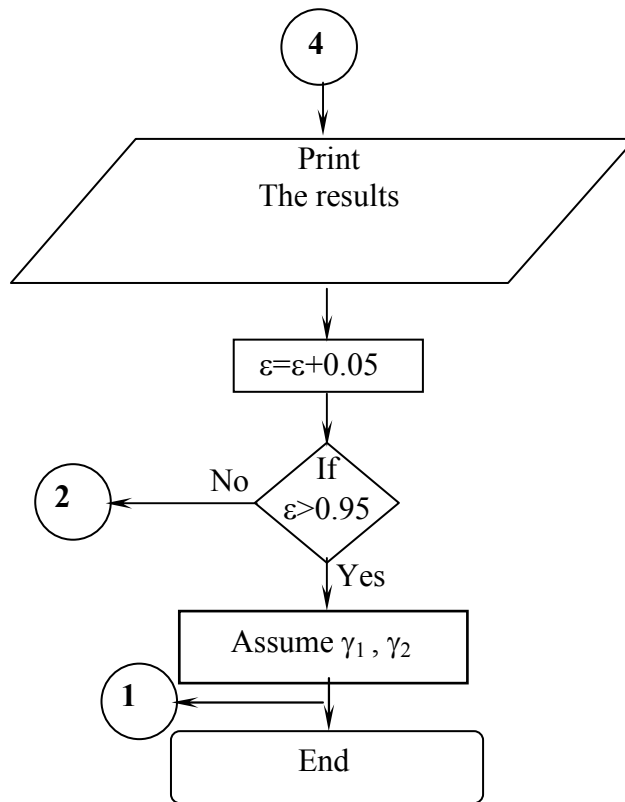
b, radial nonuniform permeability



c, axial nonuniform permeability

Fig. (1-b): Permeability Distribution in Porous Under Study.





Figure(2): flow chart of the computer program

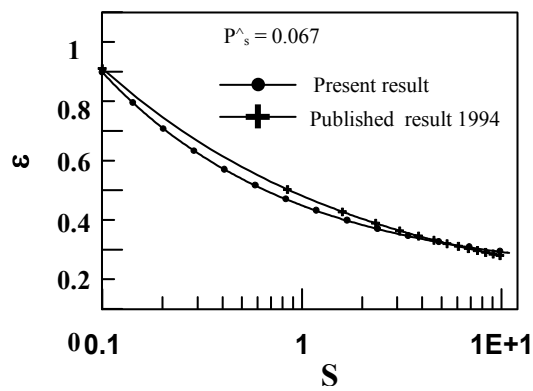


Fig. (3): Comparison between present and published result Kaneko et. al. (1994)

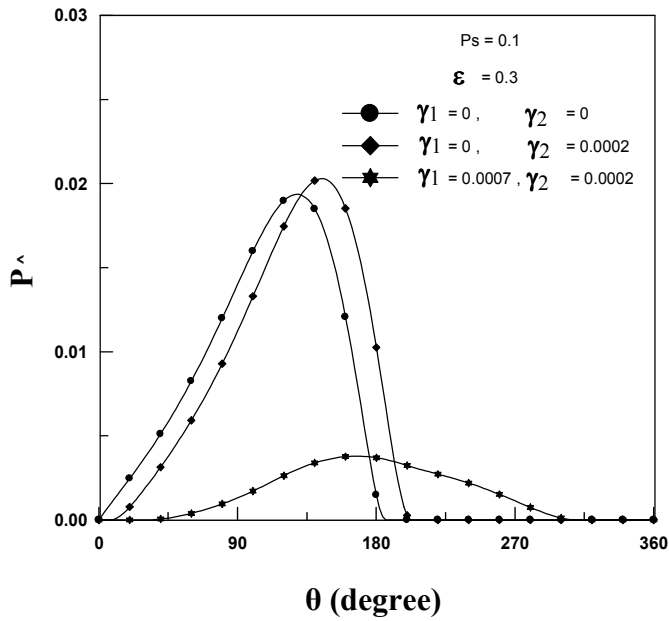


Fig. (4): Pressure distribution for different misaligned ratio at $P_s = 0.1$

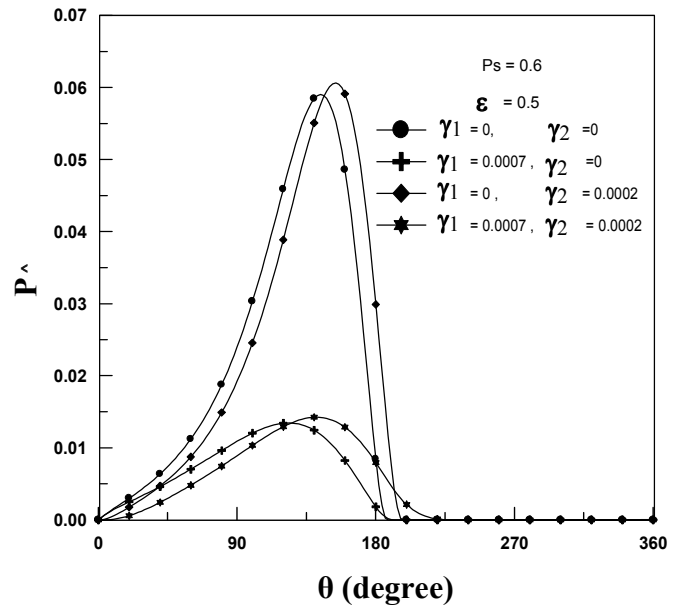


Fig. (5): Pressure distribution for different misaligned ratio at $P_s = 0.6$

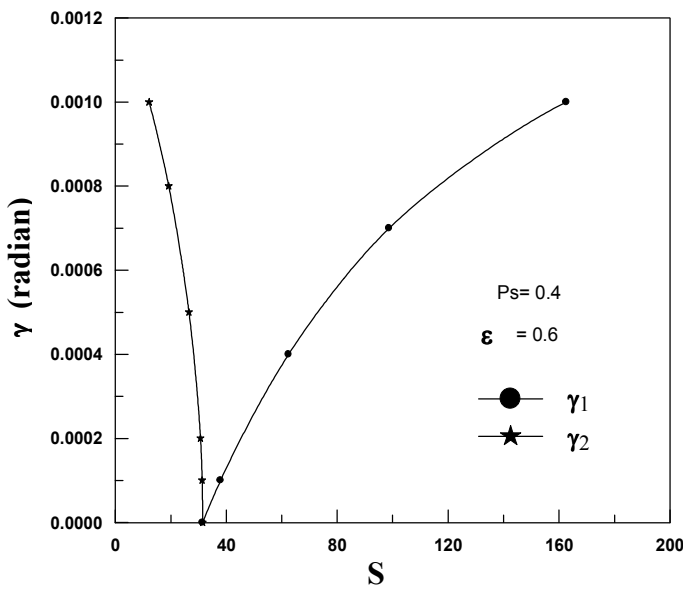


Fig. (6): Variation of Sommerfeld number with tilt angle at $\epsilon = 0.6$.

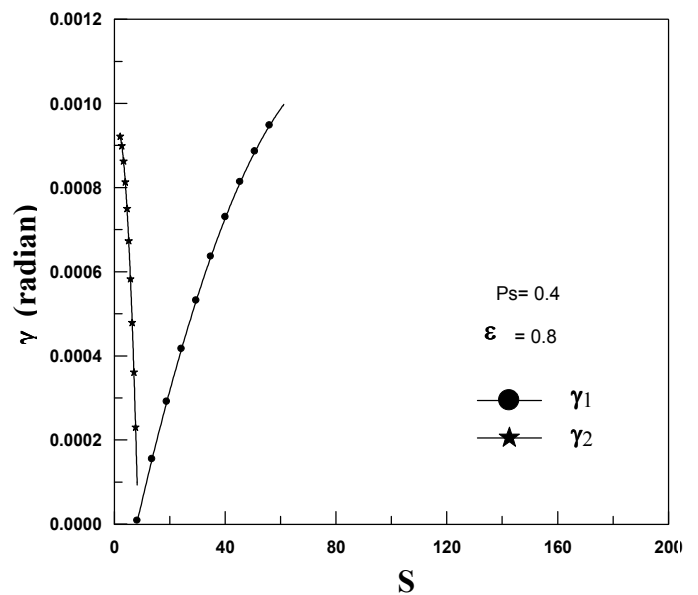


Fig. (7): Variation of Sommerfeld number With tilt angle at $\epsilon = 0.8$.

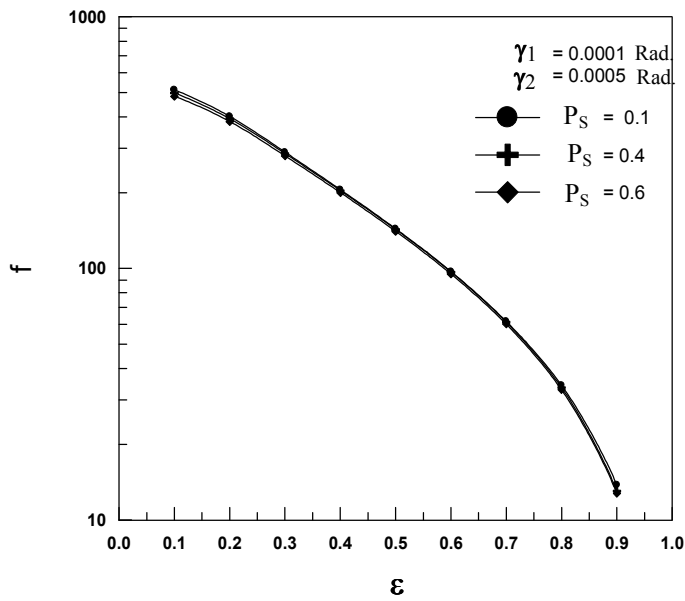


Fig. (8): Coefficient of friction versus eccentricity ratio for different values of supply pressure.

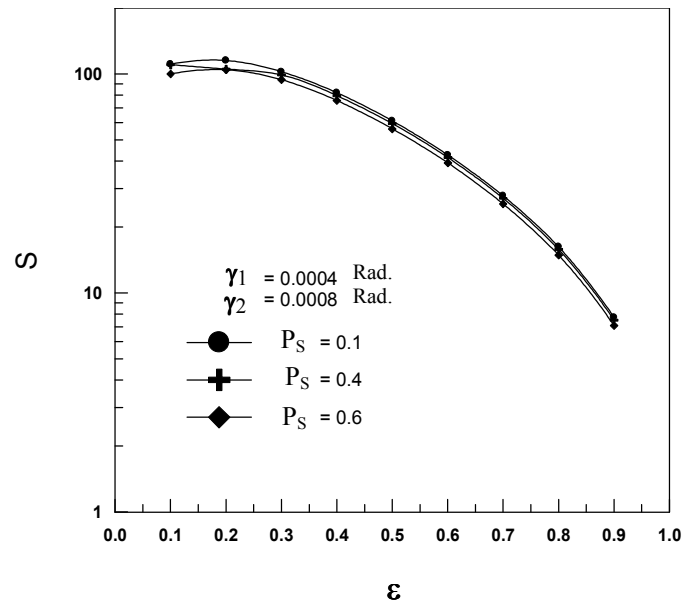


Fig. (9): Sommerfeld number versus eccentricity ratio for different values of supply pressure.

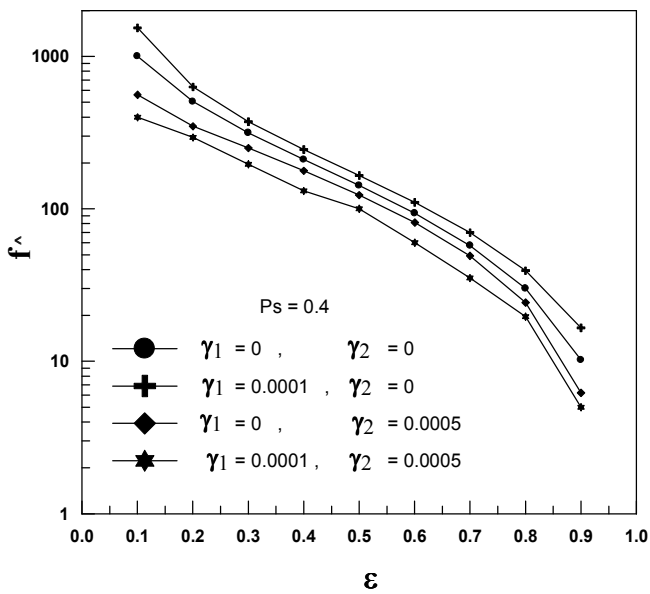


Fig. (10): Coefficient of friction versus eccentricity ratio for different misaligned ratios

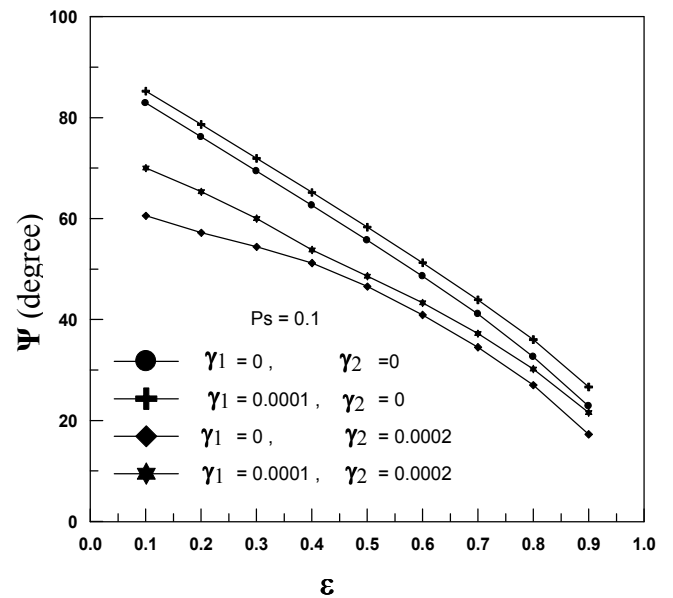


Fig. (11): Attitude angle versus eccentricity ratio for different misaligned ratios

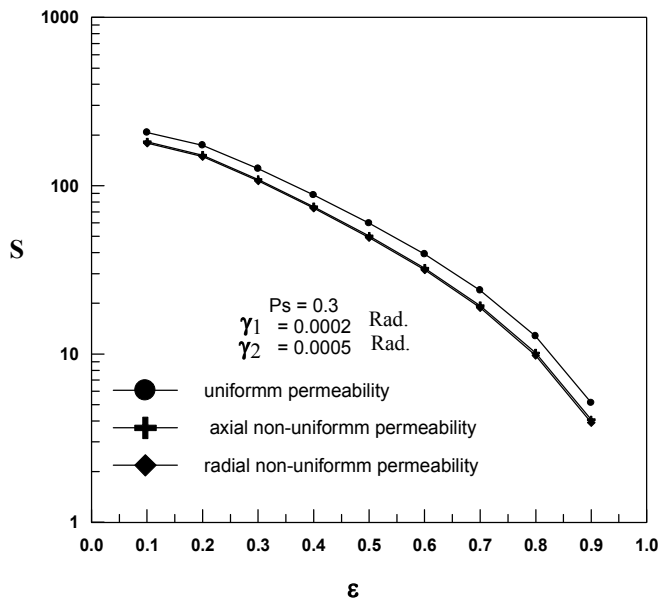


Fig. (12): Sommerfeld number versus eccentricity ratio for uniform permeability $\Phi=0.178$ and non-uniform permeability $\Phi_1=0.01$ and $\Phi_2=0.1$

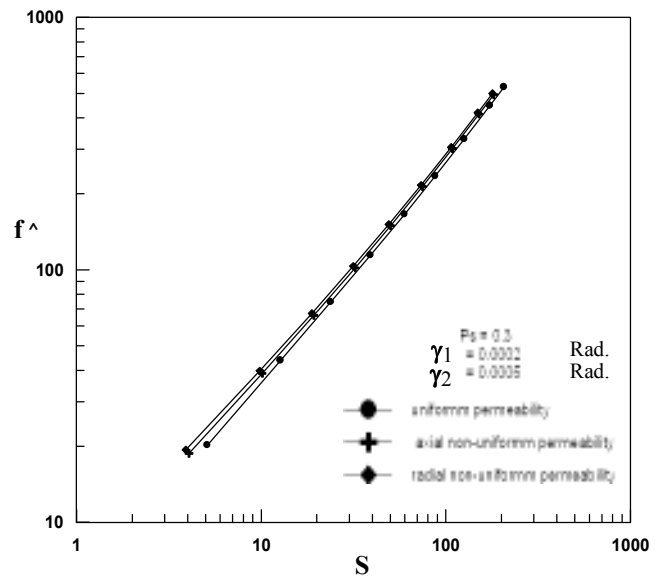


Fig. (13): Coefficient of friction versus Sommerfeld number for uniform permeability $\Phi=0.178$ and non-uniform permeability $\Phi_1=0.01$ and $\Phi_2=0.1$

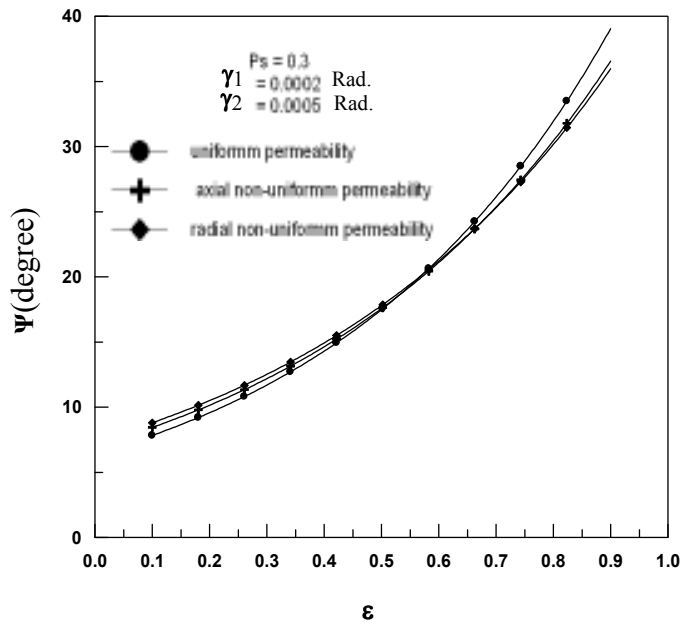


Fig. (14): Attitude angle versus eccentricity ratio for uniform permeability $\Phi=0.178$ and non-uniform permeability $\Phi_1=0.01$ and $\Phi_2=0.1$

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NOMENCLATURE:

c	journal bearing clearance	m	P_s	supply pressure (N/m ²)
D	diameter of bearing	(m)	\hat{r}	normalized radial coordinate, $\hat{r} = r/r_i$
F	friction force	(N)	R_j	journal radius(m)
\hat{f}	dimensionless friction coefficient $\hat{f} = (R/c)f$		r_i	inner radius(m)
\hat{h}	dimensionless film thickness, ($\hat{h} = h/c$)		r_o	outer radius(m)
$k_{\theta, r,z}$	permeability of the porous matrix in circumferential,		S	Sommerfeld Number , $S = (R\eta\omega_j L / W) * (r_i / c)^2$

	radial and axial direction respectively (m^2)		
L	length of the bearing (m)	s	slip parameter
N_j	journal rotational speed r.p.m	U_j	journal velocity (m/s)
P^\wedge	dimensionless oil-film pressure, $P^\wedge = c^2 P / (r^2 \eta \omega)$	W^\wedge	dimensionless load carrying capacity, $W^\wedge = W c^2 / \eta \omega_j r_i^3 L$
$P^{*\wedge}$	dimensionless oil – film pressure inside the porous matrix, $P^{*\wedge} = c^2 P^* / (r^2 \eta \omega)$	W_r^\wedge	dimensionless component of oil – film force along the line of centers
		W_T^\wedge	dimensionless component of oil – film force perpendicular to the line of centers

Greek Symbols

ϵ	Eccentricity Ratio	ψ	Attitude Angle (degrees)
η	Absolute Viscosity of Oil (Pa.s)	r, θ, z	Bearing coordinates in radial, circumferential and axial directions.
θ	Angular Coordinate from Maximum Film Thickness Position (Degree)	γ_1 & γ_2	tilt angles(rad)
ρ	Density of oil (kg/m^3)	σ_1 & σ_2	two independent misalignment parameters
Φ	Permeability parameter (m^2)	ξ	Normalized axial coordinate (z/L)

Subscript

b	Referring to Bearing
j	Referring to Journal

Superscript

\wedge	Dimensionless Quantity
*	Porous Parameter