

Steady State Characteristics of Finite Hydrodynamic Porous Oil Journal Bearing Under Turbulent Regime

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Abstract: In this paper theoretical investigation of steady state characteristics of finite hydrodynamic porous oil journal bearing under turbulent regime has been conducted. Governing equation is solved numerically by using finite difference technique. Constantinescu's turbulent lubrication theory is used for this analysis. The steady state characteristics such as load carrying capacity, friction variable, end flow rate and attitude angle have been investigated over a broad span of variables like Reynolds's number, eccentricity ratio, slenderness ratio and bearing speed parameter. These characteristics are numerically evaluated using Borland C++ software for windows 7 operating system. The results are denoted by graphs which can be convenient for the analysis of such bearings.

Keywords: Porous bearing, hydrodynamic, turbulent, lubrication.

I. INTRODUCTIUON

The expeditious creative development of modern science and technology in the fields of electronics, nuclear engineering, computer science and space technology demands high speed bearing with high refinement working characteristics even in unfavorable working condition. Porous bearing are deliberated to provide hydrodynamic lubrication without the requirement of lubricant to supply externally. Porous bearings are widely used in industrial applications where other solid bearings are not practical to use due to the lack of space or unreachable lubrication, like domestic appliances, medical apparatus, computers, automobiles, machine tools, food and textile industry etc.

Turbulent has become practical importance in hydrodynamic lubrication, especially for high speed machinery. In an effort to determine the performance characteristics of finite porous oil journal bearing at high speed, turbulent Reynolds's equation for lubrication has been developed from the fundamental equation. In this paper this is done by assuming that porous journal bearing is full and running steady. Petroff [1] was first to consider the viscous friction in the fluid film bearings. The important contribution in the science of lubrication is the experiments of Tower [2] in 1883 and the work of Reynolds in 1886, in which he made a review of detailed experimental [3] results of Tower. Taylor [4,5] studied theoretically flow between two concentric rotating cylinders and found the condition under which the laminar flow broke down to turbulent. An exhaustive study on the theory of turbulence in the lubrication has been given by Constantinescu's [6,7]. The effect of turbulence has been taken into account in the design of such bearings. A. Kumar and N.S Rao [8] originated the hydrodynamic theory of turbulent porous bearings. B.C Majumdar [9], described an analytical solution of hydrostatic porous journal bearing. K. Kalita [10], theoretically investigated steady state and dynamic characteristics of hybrid porous

oil journal bearing under turbulent regime. B.C Majumdar and A.K Chattopadhyay [11] have presented a theoretical analysis and experimental investigation of steady state characteristics of externally pressurized porous oil journal bearings. The purpose of the present analysis is to study the steady state behavior in terms of load carrying capacity, friction variable, and attitude angle and oil flow rate.

II. MATHEMATICAL FORMULATION

A schematic diagram of hydrodynamic porous journal bearing is shown in fig 1. The journal of radius R operating with steady state eccentricity ratio ϵ_0 rotates with a uniform tangential velocity U . The wall thickness H is fitted in a solid housing. The pores of the bearings are impregnated with incompressible Newtonian fluid of constant viscosity. Pressurized oil with a supply pressure P_s is fed to the outer surface of the bearing. To obtain the governing equation of flow through porous matrix, Darcy's law is implemented. The Cartesian co-ordinate system is used for the analysis of the fluid film in the bearing clearance space and it is considered that oil film in the bearing clearance space is turbulent.

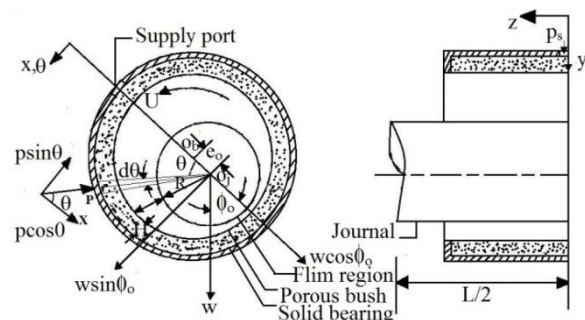


Fig 1 Hydrodynamic porous journal bearing under steady state condition

The generalized steady state equation of flow in the theory of hydrodynamic lubrication and the turbulent Reynolds's equation for porous matrix can be written as

$$\phi_x \left(\frac{\partial^2 p'_o}{\partial x^2} \right) + \phi_y \left(\frac{\partial^2 p'_o}{\partial y^2} \right) + \phi_z \left(\frac{\partial^2 p'_o}{\partial z^2} \right) = 0 \quad (1)$$

$$\frac{\partial}{\partial x} \left(\frac{h_o^3}{\mu k_x} \frac{\partial p_o}{\partial x} \right) + \frac{\partial}{\partial z} \left(\frac{h_o^3}{\mu k_z} \frac{\partial p_o}{\partial z} \right) = \frac{U}{2} \frac{\partial h_o}{\partial x} + \frac{\partial h}{\partial t} + \frac{\phi_y}{\mu} \left(\frac{\partial p'_o}{\partial y} \right)_{y=0} \quad (2)$$

k_x and k_z , the turbulence co-efficients. Inserting the dimensionless parameters given in the following in Equations (1) and (2), we get,

$$\overline{p'_o} = \frac{p'_o}{p_s}, \overline{y} = \frac{y}{H}, \overline{p_o} = \frac{p_o}{p_s}, \overline{h_o} = \frac{h_o}{C}, \overline{z} = \frac{2z}{L}, \theta = \frac{x}{R}, \overline{p'} = \frac{p'C^2}{\mu UR}, \overline{p} = \frac{pC^2}{\mu UR}, \overline{\phi_x} = \frac{\phi_x}{\phi_y}, \overline{\phi_z} = \frac{\phi_z}{\phi_y}, \lambda = \frac{\mu UR}{C^2 p_s}, \beta = \frac{12 \phi_y R^2}{C^3 H}$$

$$\frac{\phi_x}{\partial \theta^2} \frac{\partial^2 \overline{p'_o}}{\partial \theta^2} + \frac{\partial^2 \overline{p'_o}}{\partial \overline{y}^2} \left(\frac{R^2}{H^2} \right) + \frac{\phi_z}{\partial \overline{z}^2} \frac{\partial^2 \overline{p'_o}}{\partial \overline{z}^2} \left(\frac{D}{L} \right)^2 = 0 \quad (3)$$

$$\frac{\partial^2 \overline{p_o}}{\partial \theta^2} \left(\frac{h_o^3}{k_x} \right) + \left(\frac{D}{L} \right)^2 \frac{\partial^2 \overline{p_o}}{\partial \overline{z}} \left(\frac{h_o^3}{k_z} \right) = \frac{\lambda}{12} \left(\frac{\partial \overline{h_o}}{\partial \theta} \right) + \frac{\beta}{12} \overline{p_o} - \frac{\beta}{12} \quad (4)$$

Where the turbulence co-efficients k_x and k_z are is given by,

$$k_x = 12 + 0.026 (Re *)^{0.8265}$$

$$k_z = 12 + 0.0198 (Re *)^{0.741}$$

Here $\overline{h_o}$ and k_x are functions of θ only and k_z independent of z .

III. BOUNDARY CONDITION

The boundary conditions in the film region are given below:

The pressure is symmetrical about the origin in z directions,

$$\frac{\partial \overline{p_o}}{\partial \overline{z}} (\theta, 0) = 0$$

The bearing ends are considered to be open to the atmosphere,

$$\overline{p_o} (\theta, \pm 1) = 0$$

The Swift-Stieber condition or Reynolds's boundary condition is,

$$\left(\frac{\partial \overline{p_o}}{\partial \theta} \right)_{\theta=\theta_2} = 0 \text{ for } (\theta_2 \leq \theta \leq 2\pi)$$

Where θ_2 be the angle at which the film cavitates.

Again,

$$\overline{p_o} (\theta, \overline{z}) = \overline{p_o} (\theta + 2\pi, \overline{z})$$

Whenever the pressures become negative at any mesh point in the iteration, it should be made equal to zero or the cavitations' pressure, i.e. $\overline{p_o} = 0$ If $\overline{p_o} < 0$, in the cavitations zone, $\theta \geq \theta_2$.

IV. ANALYSIS OF STEADY STATE CHARACTERISTICS

A. Load Carrying Capacity:

The load carrying capacity can be evaluated by integrating the following pressure along and perpendicular to the line of center, i.e.

$$\overline{w}_x = -\frac{1}{2} \int_0^1 \int_0^{\theta_2} \overline{p_o} \cos \theta R d\theta d\overline{z} \quad (5)$$

$$\overline{w}_z = \frac{1}{2} \int_0^1 \int_0^{\theta_2} \overline{p_o} \sin \theta R d\theta d\overline{z} \quad (6)$$

Where w_x and w_z are the components of load carrying capacity along and perpendicular to the line of centers respectively.

The load parameter is given by,

$$\overline{w} = (\overline{w}_x^2 + \overline{w}_z^2)^{\frac{1}{2}}$$

And attitude angle is given by,

$$\phi_o = \tan^{-1} \left(\frac{\overline{w}_z}{\overline{w}_x} \right)$$

B. Friction Variable:

Shearing stress acting on the surface of the journal can be represented by [13]

$$\tau = \frac{\mu U}{h} \overline{\tau}_c + \frac{h}{2} \left(\frac{\partial p}{\partial x} \right) \quad (7)$$

Where $\overline{\tau}_c = 1 + 0.0023 \{Re \cdot \overline{(h_o)}\}^{0.855}$

The equation (7) is valid for non-cavitated zone and total friction force in the non-cavitated zone is given by,

$$\overline{F}_{s1} = 2 \int_0^1 \int_0^{\theta_2} \left\{ \frac{\mu U}{h} \overline{\tau}_c + \frac{h}{2} \left(\frac{\partial p}{\partial x} \right) \right\} R d\theta d\overline{z}$$

Now frictional force for the cavitated zone can be calculated as suggested by [12] and obtained as

$$\overline{F}_{s2} = \overline{h}_2 \int_0^1 \int_{\theta_2}^{2\pi} \frac{\lambda \overline{\tau}_c}{6 \overline{h_o}} d\theta d\overline{z}$$

Where $\overline{h}_2 = 1 + \epsilon_0 \cos \theta_2$

Thus the total frictional force is

$$\overline{F}_s = \overline{F}_{s1} + \overline{F}_{s2}$$

And friction variable is given by,

$$f \left(\frac{R}{C} \right) = \frac{\overline{F}_s}{2 \overline{w}}, f \text{ is co-efficient of friction.}$$

The equation (7) is valid for non-cavitated zone and total friction force in the non-cavitated zone is given by,

$$\overline{F}_{s1} = 2 \int_0^1 \int_0^{\theta_2} \left\{ \frac{\mu U}{h} \overline{\tau}_c + \frac{h}{2} \left(\frac{\partial p}{\partial x} \right) \right\} R d\theta d\overline{z}$$

Now frictional force for the cavitated zone can be calculated as suggested by [12] and obtained as

$$\overline{F}_{s2} = \overline{h}_2 \int_0^1 \int_{\theta_2}^{2\pi} \frac{\lambda \overline{\tau}_c}{6 \overline{h_o}} d\theta d\overline{z}$$

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C. Oil Flow Rate:

The oil can be flowed from the bearing by the following two ways

- (a) Flow from the open ends of the bearing
- (b) Flow from the clearance space of the bearing

At the open ends of the bearing bush, flow is negligible as compared to clearance space. Hence only flow from the clearance space is calculated.

Flow through the clearance space is given by,

$$\overline{Q} = -\frac{1}{3} \int_0^{2\pi} \frac{\partial \overline{p_o}}{\partial z} \overline{h_o^3} d\theta$$

$\overline{w}, \overline{F}_s$ and \overline{Q} Values are calculated adopting the Trapezoidal rule of numerical integration.

V. RESULTS AND DISCUSSION

A computer program in C using the relaxation iterative method to solve the governing equation is developed. The program could consider the laminar flow condition by letting $\beta = 0$ and by allowing $k_x = k_z = 12$ with $Re = 0$ in the generalized program and also for laminar flow the

value of $\bar{\tau}_c$ was put equal to unity. The results obtained by computer program operating in turbulent flow regime were generated. The steady state performance or characteristics of bearings depends on the parameters viz. $\beta, Re, L/D, \lambda$ and ϵ_0 . In the present analysis the effect of various parameters on load carrying capacity, friction variable, oil flow rate and attitude angle are analyzed by varying all the above mentioned parameters.

A. Effect of β and Re

The effect of the bearing feeding parameter shows that the load carrying capacity decreases (Fig 2) and friction coefficient increases (Fig 3). For lower values of β , \bar{w} remains unaffected by a change of β . However lower values of β depends upon Re . It is seen that for various values of Re friction variable remains constants for lower values of feed parameter ($\beta \leq 0.1$) and for higher values of β friction variable increases. Again Re increases oil flow rate increases (Fig 4). For all values of Re , oil flow rate remains constant up to certain values of β ($\beta \leq 0.1$) and then increases for higher values of β . The attitude angle is more prominent for lower values of β .

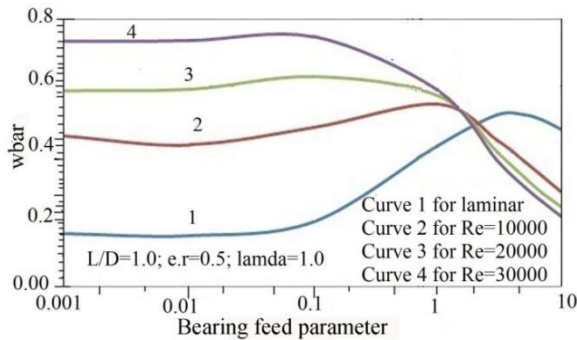


Fig 2 Non-dimensional load capacity VS beta for different Re

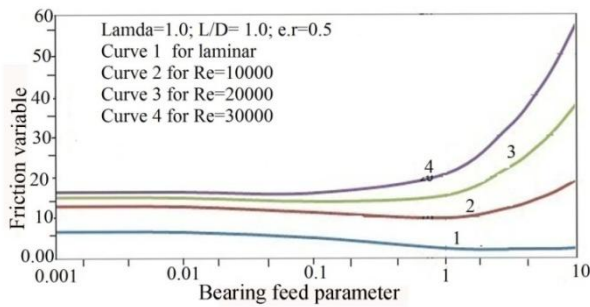


Fig 3 Friction variable VS beta for different Re

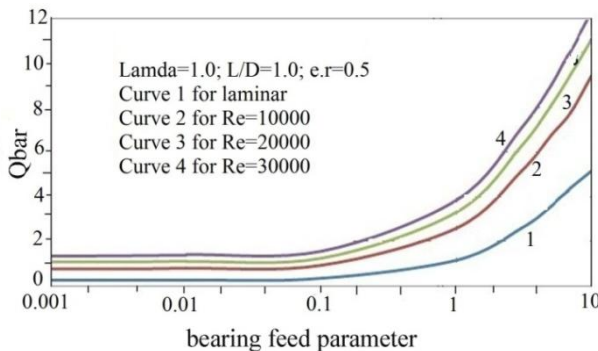


Fig 4 Oil flow rate VS beta for different Re

B. Effect of L/D

The effect of L/D ratio is to increase the load carrying capacity (Fig 5) and decrease the friction variable (Fig 6). However for particular higher values of L/D ($L/D \geq 2$) \bar{w} decreases. The oil flow rate and attitude angle also increases with the increasing L/D ratio (Fig 7 and 8).

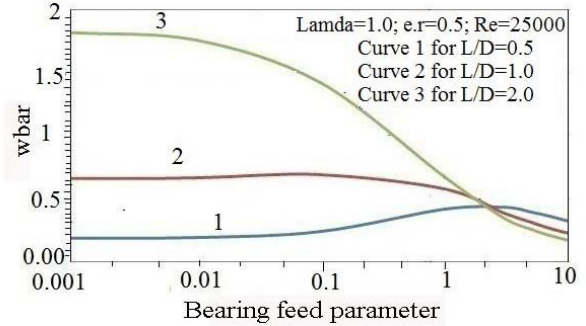


Fig 5 Load carrying capacity vs beta for different L/D ratio

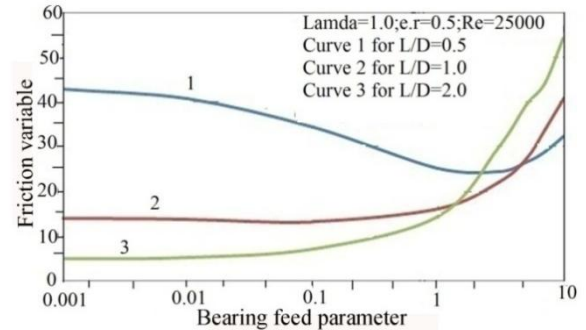


Fig 6 Friction variable VS beta for different L/D ratio

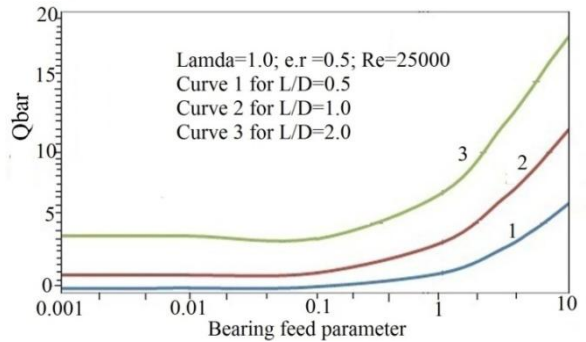


Fig 7 Oil flow rate VS beta for different L/D ratio

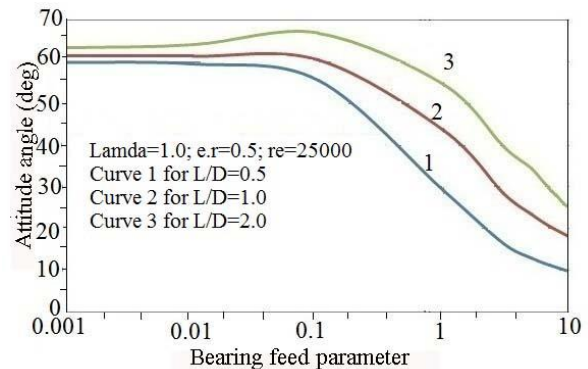


Fig 8 Attitude angle VS beta for different L/D ratio

C. Effect of λ

Load carrying capacity increases as speed parameter increases for a certain range of β (Fig 9). However at

higher values of λ ($\cong 10$), \bar{w} decreases at a faster rate with β . Friction variable increases as λ increases (fig 10). Oil flow rate increases as λ increases up to a certain values of β (fig 11). Further increase of β , they have found to be merged with each other.

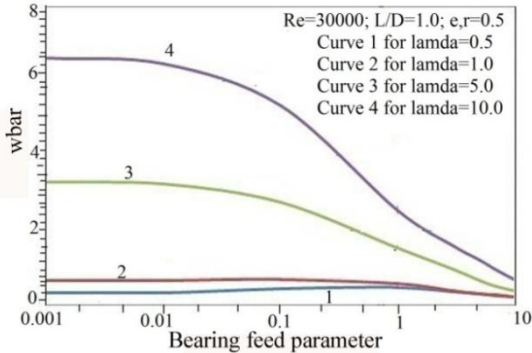


Fig 9 Load carrying capacity VS beta for different lambda

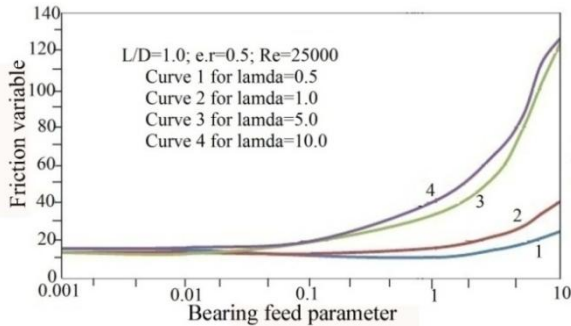


Fig 10 Friction Variable VS beta for different lambda

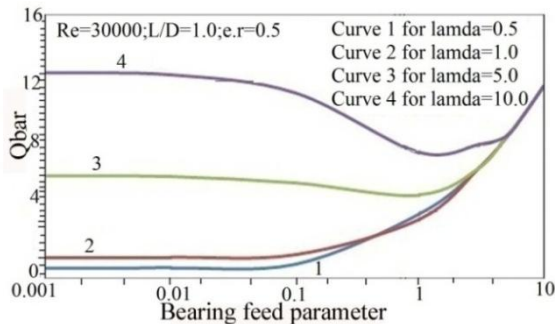


Fig 11 Oil flow rate VS beta for different lambda

D. Effect of ϵ_0 :

The effect of eccentricity ratio is to increase the load capacity and to decrease the friction variable (Fig 12 and 13).

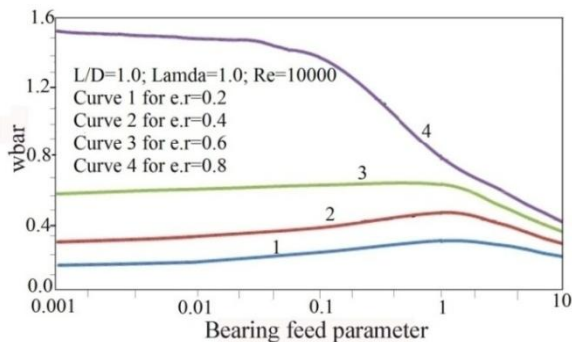


Fig 12 Load carrying capacity VS beta for different eccentricity ratio

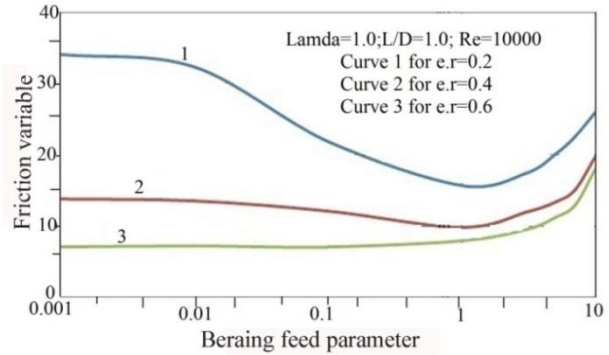


Fig 13 Friction variable Vs beta for different eccentricity ratio

With the increase in eccentricity ratio oil flow rate increases up to a certain value of β (Fig 14), beyond which flow rate is almost same. Attitude angle decreases as eccentricity ratio increases (Fig 15).

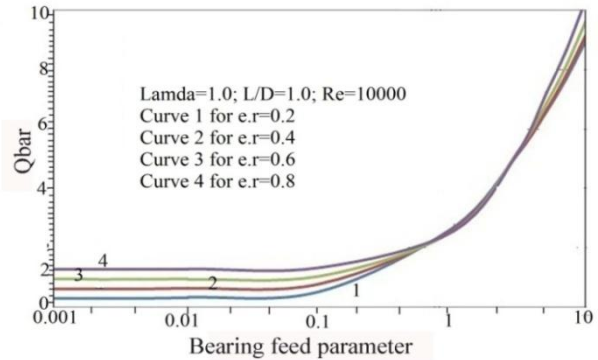


Fig 14 Oil flow rate VS beta for different eccentricity ratio

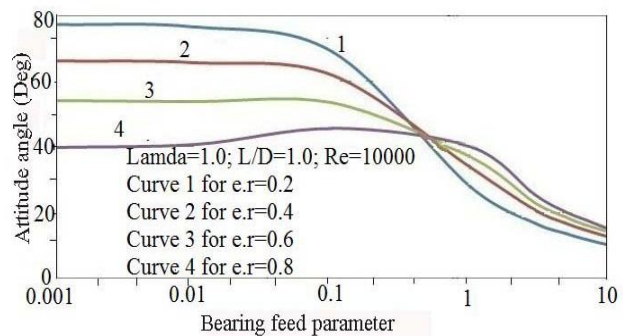


Fig 15 Attitude angle VS beta for different eccentricity ratio

VI. CONCLUSION

The effect of various parameters on steady state parameters have been investigated theoretically in a finite porous oil journal bearing on the basis of Constantinescu's theory of turbulent lubrication. The pressure distribution equation is solved numerically with the help of finite difference technique with a grid having 44 divisions in θ direction and 7 divisions in z direction.

On the basis of the results presented in fig 2-11, the following conclusion can be found.

1. The effect turbulence as Re increases is to increase the load carrying capacity, friction variable, and oil flow rate and attitude angle. But at a higher values of β , ($\beta > 2$), load carrying capacity decreases sharply.

2. Oil flow rate increases as Reynolds's number, speed parameter or eccentricity ratio increases. But at higher values of the curves tend to merge with each other.

3. The increase in non dimensional load carrying capacity is more accentuated for lager values of eccentricity ratio parameters.

Thus the bearing feed parameter value should be maintained as low as possible, ideally around 0.1, for better a pursuance of turbulent bearings.

NOMENCLATURE

C	Radial clearance (m)
e	Eccentricity (m)
D	Diameter of journal (m)
μ	Co-efficient of viscosity of the fluid (pas)
\emptyset	Attitude angle (rad. or degree)
f	Coefficient of friction
$f(\frac{R}{C})$	Friction variable.
F_s	Friction force (N)
F_{s1}	Friction force in non-cavitated region (N)
F_{s2}	Friction force in cavitated region (N)
\overline{F}_{s1}	Non-dimensional friction force in non-cavitated region (N)
\overline{F}_{s2}	Non-dimensional friction force in cavitated region (N)
h	Local fluid film thickness (m)
\overline{h}	Non-dimensional fluid film thickness
h_o	Steady State local fluid film thickness (m)
\overline{h}_o	Non-dimensional local steady state fluid film thickness
h_2	Fluid film thickness where cavitations starts (m)
\overline{h}_o	Non- dimensional form of h_2
H	Thickness of the wall of the porous bush (m)
w	Load carrying capacity of the bearing (N)
\overline{w}	Non-dimensional load capacity of the bearing
w_x, w_z	Components of load carrying capacity along and perpendicular to the line of centers
$\overline{w}_x, \overline{w}_z$	Non-dimensional form of W_x and W_z
x, y, z	Circumferential, radial and axial co-ordinates
ρ	Density of the lubricating film (kg/m ³)
τ	Shear stress on journal surface (N/m ²)
φ_0	Steady state attitude angle (Rad. or degree)
$\varphi_x, \varphi_y, \varphi_z$	Permeability coefficients of porous medium along x, y and z direction (m ²)
ϵ	Eccentricity ratio
ϵ_0	Steady state eccentricity ratio
λ	speed parameter which is equal to $\frac{\mu UR}{C^2 p_s}$
$C\beta$	Bearing feed parameter which is equal to $\frac{12\varphi_y R^2}{C^3 H}$
θ_2	Angle at which the fluid film starts to cavitates
$\Delta\theta, \Delta z$	Finite difference meshes size along circumference and length of the bearing. (m)
k	Constants
k_x, k_z	Constants in turbulent Reynolds equation
l	Length of the bearing (m)
L/D	Slenderness ration or Aspects ratio
n	Number of iterations

p	Pressure in the clearance space (N/m ²)
\overline{p}	Non-dimensional form of p
p_o	Steady state pressure in the clearance space (N/m ²)
\overline{p}_o	Non dimensional form of p_o
p'	Pressure in the porous medium (pa)
p'	Non-dimensional form of p'
p'_o	Steady state pressure in the porous medium (pa)
p'_o	Non-dimensional form of p'_o
p_s	Oil supply pressure to the bearing (pa)
u, v, w	Velocity component along x, y and z directions respectively in the fluid film region (m/s)
u', v', w'	Velocity component along x, y and z directions respectively in the porous medium (m/s)
U	Surface velocity of journal (m/s)
Q	End flow of oil (m ³ /s)
\overline{Q}	Non-dimensional form of Q
R	Radius of the journal (m)
Re	Average Reynolds number which is equal to $\frac{\rho UC}{\mu}$
Re^*	Instantaneous Reynolds number which is equal to $\frac{\rho U h}{\mu}$
R'	Modified Reynolds number which is equal to $k^2 Re^*$
τ_c	Coquette surface shear stress (N/m ²)
$\overline{\tau}_c$	Non dimensional form of τ_c which is equal to $\frac{\tau_c h}{\mu U}$
\overline{y}	Dimensional Radial co-ordinate which is equal to $\frac{y}{H}$
\overline{z}	Dimensional axial coordinate which is equal to $\frac{z}{L}$

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