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Influence of Radial Clearance on the Static Performance of Hydrodynamic Journal Bearing System

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ABSTRACT

The present work aims to analytically study the clearance effect on hydrodynamic journal bearing system's static performance. The mathematical model of hydrodynamic lubrication i.e. the non-dimensional Reynolds equation is solved with the help of FEM formulation using Garlerkin's approach along with appropriate boundary conditions. A positive pressure zone, which is unknown priori is established using Reynolds boundary condition through iteratively. The study uses a non-dimensional clearance parameter and its influence on fluid-film pressure and static performance parameters of hydrodynamic journal bearing is studied. The static performance parameters include maximum pressure, load carrying capacity, attitude angle, and coefficient of fluid-film friction etc. The simulated results indicate that the low value of clearance is often a better option which however is limited by surface asperities. The results of the study are useful to the bearing designer from the view point of optimal selection of clearance parameter on the bases of maximizing bearing load and minimizing friction force.

Keywords: *Hydrodynamic Journal Bearing; Static Characteristics; Finite Element Method (FEM).*

1.0 Introduction

The high speed rotating machines are generally designed for low weight and low costs, which yields the use of hydrodynamic journal bearing in a large number of rotating machineries. In this bearing the position of journal is directly related to external load. When the bearing is supplied with lubricant and external load is zero, the journal will rotate concentrically within the bearing. As the load increases, the journal shifts eccentrically, and forming a wedge shaped oil film. This action causes load supporting pressure, commonly known as "hydrodynamic fluid-film pressure". The hydrodynamic fluid-film pressure and bearing performance is influenced by various operating and geometric design parameters. The clearance contact is a cause for relative motion between journal and bearing surface. The lubricant between the interfacing surfaces forms wedging action and thereby builds up fluid-film pressure. It is therefore apparent that the magnitude of clearance has an impact on the hydrodynamic performance and therefore an understanding on how clearance parameter influences journal bearing performance is the aim of the current study. The paper present to deals with the influence of radial clearance on the film pressure and static performance of bearing. The influence of clearance was studied by many investigators.

Ocvirk and Dubois [1] examined that load carrying capacity decreases with increase in bearing clearance and the peak pressure in the oil film rises rapidly with eccentricity ratio and with increasing bearing clearance. Mitsui et al [2] examined that the bearing surface temperature rise increases with decreasing the radial clearance. Prasad [3] studied the effect of clearance ratio on the maximum bearing temperature and its location, the surface temperature rise increases with decreasing of radial clearance. Simmsons and Dixon [4] showed that the results for large clearance configurations did not show dramatic variation in maximum bearing temperature associated with a transition from laminar to turbulent cooling that was found for standard clearance cases. Papadopoulos et al.[5] theoretically presented the identification of clearances and stability analysis for a

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rotor journal bearing system using response measurements of the rotor at a particular point i.e. the midpoint of the rotor. The measurements should be taken at two different speeds and from different wear effects.

This present work also verified experimentally from the previous work. Sharma et al [6] studied on the journal bearing performances and metrology issues. In this experimental study out-of-roundness and radial clearance of journal bearings were measured with high precision and the impact of their metrology was examined on the specific oil film thickness of the bearing.

Results showed that the radial clearance measurements can vary from one measuring device to another and the specified clearance may not necessarily meet the design criteria of specific oil film thickness. Tian et al [.7] examined the effect of bearing outer clearance on dynamic behavior of full floating ring and observed that the stability could be enhanced by increasing clearance. Fargere and Velex [8] examined that the alignment of the shaft can be modified by changing the bearing clearance.

The studies conducted by the various investigators indicate the radial clearance or clearance ratio is invariably an essential design parameter which influences performance of hydrodynamic journal bearing.

The research on clearance although is not new but being the prime design parameter, the exclusive study and investigation on its influence on bearing performance is still lacking. An insight on the effect of clearance on fluid-film pressure variation, maximum pressure, fluid-film friction etc. for different loading range is lacking in the tribology books and various publications [9-14]. The current theoretical study is conducted to fill the gap in the existing literature.

The study aims to examine the influence of clearance ratios on the non-dimensional values of pressure distribution and static performance parameters for a bearing operating under different loading influence.

The study considers a wide loading range by choosing different values of eccentricity ratios. The results of the study from the view point of static performance are useful to the researchers for selecting appropriate value of clearance parameter.

A non-dimensional clearance parameter ($\bar{\square}$) is described relative to a reference value of clearance. Based on the value of $\bar{\square}$, the expression for fluid film thickness is expressed, which is used in non-dimensional Reynolds equation as described in the following section.

2.0 Analysis

In order to conduct the theoretical study, the governing two-dimensional Reynolds equation is solved using finite element method (FEM) for evaluating pressure distribution.

2.1 Reynolds equation

For the analysis purpose, a mathematical model in terms of two dimensional Cartesian co-ordinating static Reynolds equation is expressed as (11)

$$\frac{\partial}{\partial x}\left(\frac{h^3}{12\mu}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{h^3}{12\mu}\frac{\partial p}{\partial y}\right)=\frac{u}{2}\frac{\partial h}{\partial x} \qquad (1)$$

Where the fluid-film thickness, h is expressed as

$$h = c - x_j \cos\alpha - z_j \sin\alpha \qquad (2)$$

The equation (1) reduces to the following non-dimensional form which is expressed as

$$\frac{\partial}{\partial \alpha}\left(\frac{\bar{h}^3}{12\mu}\frac{\partial \bar{p}}{\partial \alpha}\right)+\frac{\partial}{\partial \beta}\left(\frac{\bar{h}^3}{12\mu}\frac{\partial \bar{p}}{\partial \beta}\right)=\frac{\Omega}{2}\frac{\partial \bar{h}}{\partial \alpha} \qquad (3)$$

Where from equation (2), a non-dimensional value of $\bar{h}$ is expressed as

$$\bar{h} = \bar{c} - \bar{x}_j \cos\alpha - \bar{z}_j \sin\alpha \qquad (4)$$

The clearance parameter $\bar{c}$ is expressed as the ratio of actual clearance to the reference clearance and is used as a variable term in the current study. Thus the approach used in the current work is quite different and moreover the study has been performed using FEM, which to the best of author's knowledge is quite significant in the context of clearance related study.

2.2 FEM components

The fluid flow in the clearance space of finite journal bearing is discretized using four noded quadrilateral isoparametric elements. Using lagrangian linear interpolation function, the pressure at a point in the element is bilinearly distributed as

$$\left.\begin{aligned} \bar{p} &= \textstyle\sum_{j=1}^{4} \bar{p}_j \bar{N}_j \\ \alpha &= \textstyle\sum_{j=1}^{4} N_j \alpha_j \\ \beta &= \textstyle\sum_{j=1}^{4} N_j \beta_j \\ \frac{\partial p}{\partial \alpha} &= \textstyle\sum_{j=1}^{4} \frac{\partial N_j}{\partial \alpha} \cdot \bar{p}_j \\ \frac{\partial p}{\partial \beta} &= \textstyle\sum_{j=1}^{4} \frac{\partial N_j}{\partial \beta} \cdot \bar{p}_j \end{aligned}\right\} \quad (5)$$

Using Garkelin's orthogonally criteria and approximate value of □as in eq (5), the equation (3) can be expressed as

$$\frac{\partial}{\partial x}\left[\frac{\bar{h}^3}{12\bar{\mu}}\frac{\partial}{\partial \alpha}\left(\textstyle\sum_{j=1}^{4} P_j N_j\right)\right] + \frac{\partial}{\partial \beta}\left[\frac{\bar{h}^3}{12\bar{\mu}}\frac{\partial}{\partial \beta}\left(\textstyle\sum_{j=1}^{4} P_j N_j\right)\right] - \Omega\left[\frac{1}{2}\frac{\partial \bar{h}}{\partial \alpha}\right] = R^e \quad (6)$$

Where Re is known as residue for a element. As per this technique, minimization of residue is obtained by ortogonalising the residue with interpolation function, i.e.

$$\iint_{\Omega e} N_j R^e \quad d\alpha . d\beta \quad = \quad 0 \quad (7)$$

Integrating the 2nd order term in equation (7) by parts, so as to obtain co continuity, the resultants equation for a typical element is obtained in matrix form as follows

$$[\bar{F}]^e \{\bar{p}\}^e = \{\bar{Q}\}^e + \Omega\{\bar{R}_H\}^e \quad (8)$$

For eth element, the matrix and coulumn-vectors are expressed as

$$\bar{F}^e{}_{ij} = \iint_{\bar{A}^e} \left\{\frac{\bar{h}^3}{12\bar{\mu}}\left(\frac{\partial N_i}{\partial \alpha}\frac{\partial N_j}{\partial \alpha} + \frac{\partial N_i}{\partial \beta}\frac{\partial N_j}{\partial \beta}\right)\right\} d\alpha\, d\beta \quad (9a)$$

$$\bar{Q}^e{}_i = \int_{\tau^e} \left\{\left(\frac{\bar{h}^3}{12\bar{\mu}}\frac{\partial \bar{p}}{\partial \alpha} - \frac{1}{2}\bar{\Omega}\bar{h}\right) l1 + \left(\frac{\bar{h}^3}{12\bar{\mu}}\frac{\partial \bar{p}}{\partial \bar{\beta}}\right) l2\right\} Ni\, d\bar{A}^e \quad (9b)$$

$$\bar{R}^e{}_{Hi} = \iint_{\bar{A}^e} \frac{1}{2}\bar{h}\frac{\partial N_i}{\partial \alpha}\, d\alpha\, d\beta \quad (9c)$$

Where l1 and l2 are direction cosines and i,j = 1,2…… n_j^e (number of nodes per element) are node numbers. Aerefers to the domain and τe is the boundary of the eth element.

2.3 Boundary conditions

For small disturbances, the extent of positive pressure film may be assumed to remain unchanged and the following boundary conditions may be stipulated

1. At the external boundary, nodal pressure are zero,

$$\bar{p}(\alpha,\beta = \pm 1.0) = 0.0 \quad (10a)$$

2. At the trailing edge of positive pressure region using Reynolds boundary condition,

$$\bar{p}(\alpha_2,\beta) = 0\,,\ \frac{\partial \bar{p}}{\partial \alpha} = \frac{\partial \bar{p}}{\partial \beta} = 0. \quad (10b)$$

It may be observed that the positive pressure zone is unknown and is iteratively established by considering the activated pressure and its gradient to be constant and equals zero.

2.4 Static performance parameters

Fluid-film forces and friction coefficient

The dimensionless fluid-film force on the journal surface is given by

$$\left.\begin{aligned} \bar{F}_x &= \int_{-1}^{1}\int_{0}^{2\pi} \bar{p}\, cos\alpha . d\alpha . d\beta \\ \bar{F}_z &= \int_{-1}^{1}\int_{0}^{2\pi} \bar{p}\, sin\alpha . d\alpha . d\beta \end{aligned}\right\} \quad (11)$$

and the resultant fluid-film force F, is equivalent to

$$\bar{F} = \sqrt{F_x{}^2 + F_z{}^2}$$

2.4.1 Friction force

The non-dimensional hydrodynamic friction force in a journal bearing is calculated as

$$\bar{F}_f = \textstyle\sum_{e=1}^{n_e} \int_{A^e} \left[\frac{1}{\bar{h}} + \left(\frac{\bar{h}}{2}\frac{\partial \bar{P}}{\partial \alpha}\right)\right] d\bar{A}_e] \quad (12)$$

Where, ne are total number of elements in fluid-flow domain. The coefficient of friction of fluid-film is given by

$$\bar{F}\left(\frac{R}{c_o}\right) = \frac{\bar{F}_f}{\bar{F}}$$

2.4.2 Attitude angle

The location of journal centre is measured by attitude angle which is the angle formed between the vertical and a line that crosses through the centre of the journal and the centre of the bearing. The attitude angle is calculated

$$\text{as } \phi = \tan^{-1}\left(\frac{\bar{x}_j}{\bar{z}_j}\right)$$

3.0 Solution Procedure

The solution of hydrodynamic journal bearing requires an iterative scheme at two stages. In order to establish the positive pressure zone, which is unknown priori, iteration is needed. Further the pressure area is computed for a specific value of attitude angle at a specified eccentricity. The solution scheme uses various modules as shown in (figure 1), the iteration is continues until a journal center equilibrium is attained for a specified eccentricity ratio.

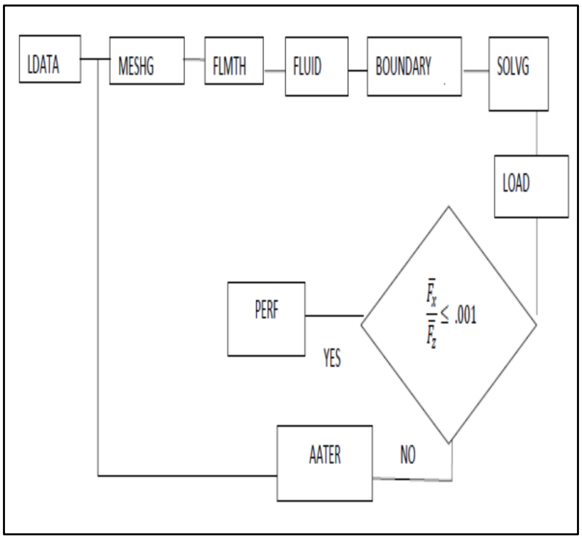
The unit LDATA reads program controlling parameters, which is followed by MESHG, which generates 2-D lubrication mesh by specifying elements in circumferential and axial length direction. The unit FLMTH computes fluid film- thickness given by equation (3), utilizing tentative value of eccentricity ratio and attitude angle.

The element matrices of equation (9), are generated using numerical integration and assembled in unit FLUID. Unit BOUNDARY modifies the system equation, which is solved in unit SOLVG using MATLAB/ operator, which will give nodal pressure. Integrating nodal pressure in LOAD module will give load Fx, and Fz components. The journal center equilibrium for a specified eccentricity ratio is attained in unit AATER.

The detail of unit AATER has been specified in a separate flowchart as shown in (figure 2), which is established to attain attitude angle. In order to achieve the convergence, the tolerance has been specified to be 0.001 in the current study which have been taken from the reference (11, 12). The algorithm uses the convergence criteria as $\frac{F_x}{F} < 0.1\%$. When attitude angle is established, the static performances are computed.

A MATLAB code is developed based on the above algorithm and has been used in the current study.

Fig. 1 Overall Solution Scheme



4.0 Results and Discussion

The current study uses an optimal grid (36×8) to describe the fluid-film domain. The selection of grid size is based on the accuracy of simulated data and minimum computational time.

The simulated results from the present study is compared with the published results of Raimondi and Boyd10, Chandrawat and Sinhasan11 and Jain et al.12For clearance parameter □̄ = 1.0, ε = 0.6 and 0.7. The comparisons of computed data are presented in Tables 1and 2.

It may be observed from these tables that the results from the current study is in good agreement from the published data [10-12].The simulated results given in Tables 1 and 2, establishes the accuracy of developed code.

The numerically simulated results from the developed model is presented for fluid-film pressure distribution along circumferential and axial direction for the commonly used operating parameters ε = 0.5. The bearing configuration in the current study is considered to be finite one.

Table 1 Analysis of Static Characteristics of Plain Journal Bearing (Eccentricity Ratio $\varepsilon = 0.6$, L/D =1)

Performance Parameters	Current Work	Raimondi and Boyd Ref. (10)	Chandrawat and Sinhasan Ref. (11)
Load Capacity, $\overline{W}_0$	5.095	5.2613	5.1662
Attitude Angle, ϕ	52.044	50.58	51.99
Friction coefficient	3.319	3.22	3.41

Table 2 Comparison of Static Characteristics of Plain Journal Bearing (Eccentricity Ratio $\varepsilon = 0.7$, L/D=1)

Performance Parameters	Current Work	Jain et.al Ref. [2]	Chandrawat and Sinhasan Ref. [11]
Minimum Fluid-Film Thickness, $\overline{h}_{min}$	0.303	0.300	0.300
Maximum Fluid-Film Press. $\overline{p}_{max}$	5.478	5.360	5.516
Load Capacity, $\overline{W}_0$	7.889	7.66	7.98
Attitude Angle, ϕ	45.104	43.66	44.97
Friction coefficient	2.478	2.5457	2.4524

The clearance parameters for the non-dimensional pressure and fluid-film thickness are taken to be □= 0.9, 1.0, 1.1. Figure 3 represents the relation between the maximum pressure and circumferential direction. Figure represents the pressure profile at different clearance parameter resulting the fluid film journal bearings achieve a higher value at small clearance parameter.

From the figure it is clear that from α = 0 to 120 degree oil film pressure is zero. From α = 120 degree pressure starts rising and becomes maximum near to α = 290 degree and then falls at α = 300 degree.

A decrease in clearance parameter yields the pressure profile rise, which is tending to increase in hydrodynamic action. Figure 4 represents as the clearance parameter increases, the overall fluid film thickness also increases. It shows that the bearing is unwrapped, the fluid film.

Thickness starts increasing up to 150 degree, which is top segment of bearing whereas it starts decreasing at the bottom segment and then slightly increasing up to 360 degree.

Fig. 2 Flow Chart of the Program (AATER)

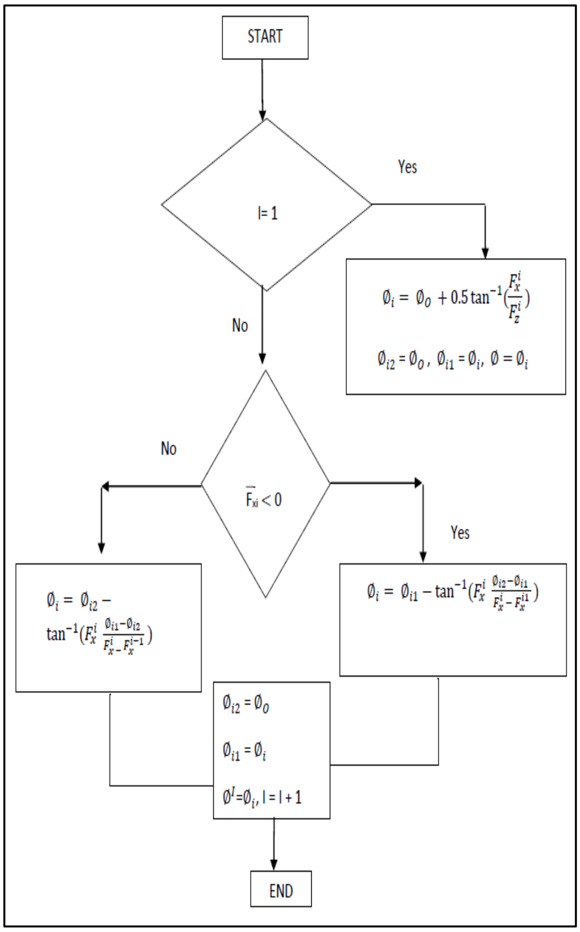
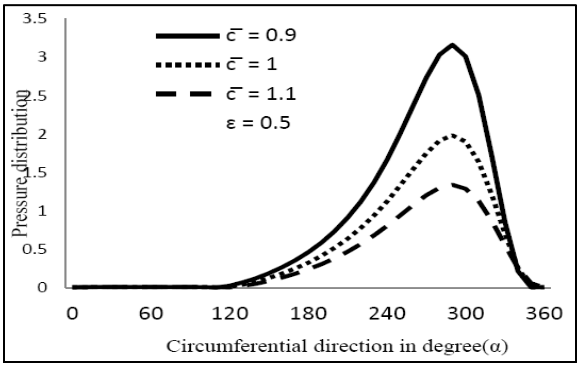


Fig.3 Influence of Clearance Parameters in Pressure Distributions Along Circumferential Direction

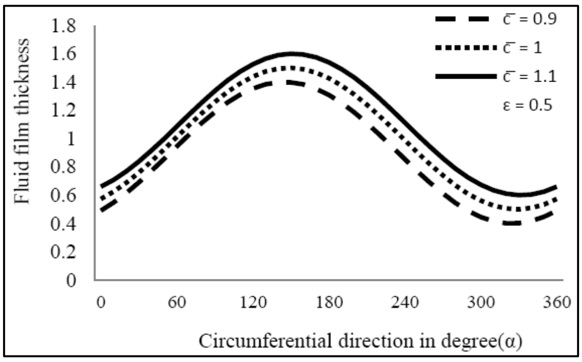


4.1 Static performance parameters

The influence of clearance parameter on static performance of fluid-film bearing has been carried covering low, medium and high operating range of eccentricities. Therefore the simulated results have been generated for eccentricity ratios ε = 0.3, 0.6 and 0.9 for a wide range of clearance parameter □̄ = 1.0-2.0. From the generated data, the bearing performance characteristics have been plotted as a function of clearance parameter □̄.

Circumferential direction in degree(α)

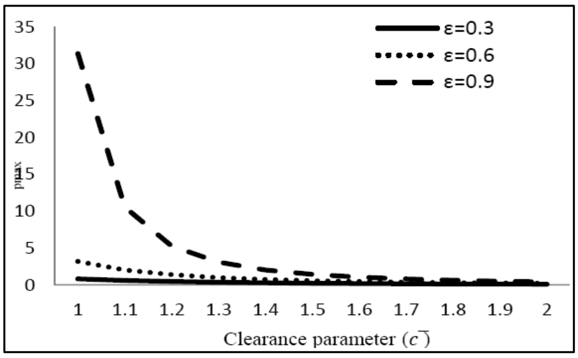
Fig. 4.Influence of Clearance Parameter on Fluid Film Thickness Along Circumferential Direction



4.2 Influence on maximum pressure (□̄max)

With the increase in clearance parameter the non-dimensional value of maximum fluid-film pressure decreases as may be seen in Figure 5. The hydrodynamic fluid-film pressure is carried due to wedge shape fluid-film.

Fig. 5 .Maximum Pressure Verses Clearance at Different Eccentricity Ratio



4.3 Influence on load carrying capacity (□̄)

It is clear from the Figure 6 that as the clearance parameter increases there is decreases in the value of load carrying capacity. When the □̄ increases, the film thickness also increase, which causes the fluid-film to behave as soft spring and thereby load carrying capacity is also dropped. A similar trend is observed for the different value of ε.

4.4 Influence on attitude angle (ø)

At a specified value of eccentricity ratio, attitude angle is indicator of journal centre position in fluid film domain. With the increase in the value of clearance parameter there is a slight increase of attitude angle as in Figure 7. Similarly if the value of ε increases to 0.6 and 0.9, there is appreciable rise in the value of attitude angle at higher value of □̄. Higher the attitude angle is an indicator of minimum fluid-film thickness.

Fig. 6 Load Carrying Capacity Verses Clearance at Various Eccentricity Ratios

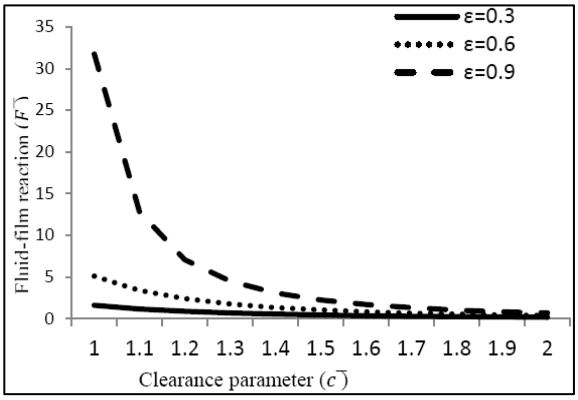


Fig. 7 Attitude Angle Verses Clearance at Various Eccentricity Ratios

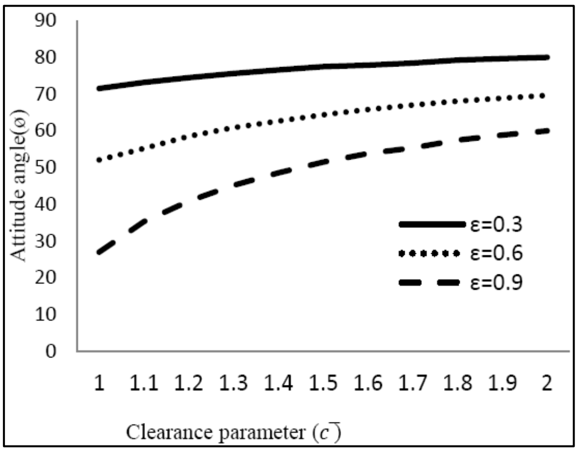


Fig. 8 Force of Friction Verses Clearance at Different Eccentricity Ratio

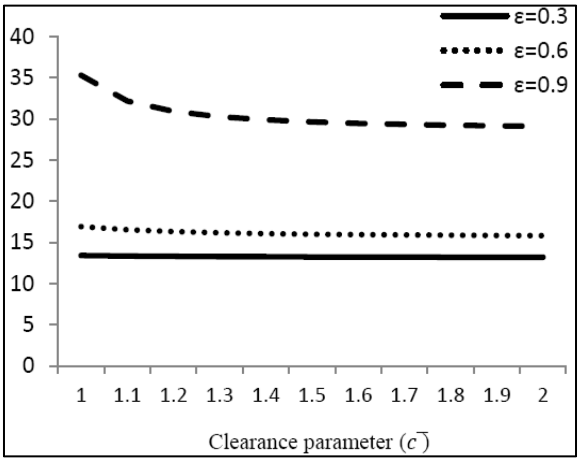
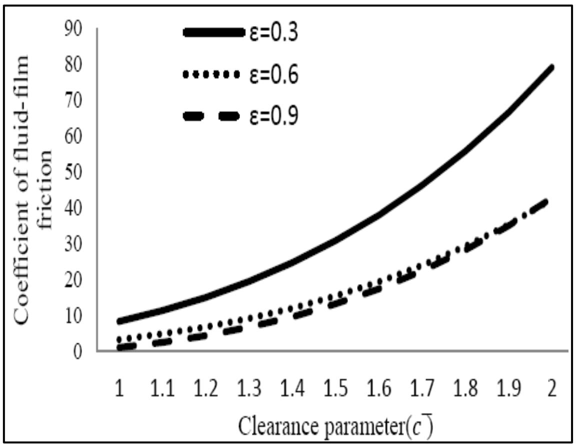


Fig. 9 Coefficient of Friction Verses Clearance at Different Eccentricity Ratio



4.5 Effect on fluid film friction $(\bar{F}_f)$ and coefficient of fluid-film friction

From the Figure 8 the fluid-film friction remains invariant with clearance parameter, however at higher eccentricity ratio a slight reduction is seen when clearance parameter is increased up to 20%. Figure 9 illustrates that if the clearance parameter increases, than the coefficient of fluid-film friction is also increases. As this parameter is a ratio of fluid film friction to fluid-film load carrying capacity. As load carrying capacity decreases with the increase in the τ parameter hence COF value causes an increase due to reduction in the denominator term, i.e. load carrying capacity.

5.0 Conclusions

This work presents the theoretical study concerning the effect of clearance on the pressure, film thickness and static performance parameters of fluid film journal bearing system. Based on the numerically simulated results following conclusions are drawn.

1. The maximum fluid film pressure, load carrying capacity and fluid-film friction decreases with the increase in clearance parameters.
2. The attitude angle, coefficient of fluid-film friction increases with the increases in the clearance parameters.

From the above discussion, it is clear that low value of clearance parameter is preferable choice as it enhances load carrying capacity of the fluid-film bearing without any appreciable increase in the fluid-film friction force. The optimal value of clearance parameter from the selected data $\bar{c}$ is ≤ 1.0. The lowest value however will be dependent on the surface asperities of the contacting surfaces.

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Nomenclature

Dimensional parameters

x,y,z = co-ordinate axes with origin at geometric centre of bearing

x_j, z_j= journal centre position from geometric centre of bearing

h = fluid-film thickness

μ = viscosity of lubricant

μ_r = reference viscosity

ω = angular speed of journal (rad/s)

c = radial clearance

c_0= reference clearance

U = tangential journal speed (ωR)

R = journal radius

Non-Dimensional Parameters

$\boldsymbol{\alpha}$ = circumferential co-ordinate $\left(\frac{x}{R}\right)$

$\boldsymbol{\beta}$= axial xo-ordinate $\left(\frac{y}{R}\right)$

$\boldsymbol{\Omega}$ = speed parameter, $\frac{\omega}{\omega_r}$

$\bar{\mu}= \frac{\mu}{\mu_r}$

$\bar{p} = \frac{p}{\mu\omega\left(\frac{R}{c_o}\right)^2}$

$\bar{h} = \frac{h}{c_o}$

$\bar{c}$= clearance parameter, $\frac{c}{c_o}$

$\boldsymbol{\varepsilon}$ = eccentricity ratio, $\frac{e}{c_o}$

$\boldsymbol{\alpha_2}$ = extent of positive pressure film

e = journal eccentricity

ø = attitude angle, radian

$$\bar{F}, \bar{F}_x, \bar{F}_z = \frac{F, F_x, F_z}{\mu\omega R^2\left(\frac{R}{c_o}\right)^2}$$