

Stability analysis on plain journal bearing with effect of surface roughness

S.Udaya Bhaskar, Mohammed Manzoor Hussain, Md. Yousuf Ali

Abstract— The effect of surface roughness has a major role in steady state and dynamic characteristics of hydrodynamic journal bearings. To evaluate the surface roughness effect on bearing surface, it is necessary to concentrate upon the surface roughness profiles. Flow factor method is used in the analysis to evaluate the roughness terms and the finite difference method is used to find the pressure distribution over the bearing surface. The steady state characteristics are with transverse, isotropic and longitudinal surface pattern are compared with the bearing having smooth surface profile. The results showed a considerable effect on steady state and dynamic characteristics of the journal bearing system.

Index Terms— attitude angle, eccentric ratio, journal bearing, reynolds equation, sommerfeld number, surface roughness.

1 INTRODUCTION

Fluid film bearings are machine elements which produce smooth (low friction) motion between solid surfaces in relative motion and to generate a load support for mechanical components. The lubricant between the surfaces may be a liquid or gas. Fluid film bearings are designed to support static and dynamic loads, and consequently as their effects on the performance of rotating machinery are of great importance.

In the design of high speed rotary machinery, consideration of surface roughness is important to predict the steady state characteristics and stability analysis of the hydrodynamic journal bearing. It is observed by B.C Majumdar [5] that an increase in the load carrying capacity occurs when the effect of surface roughness is taken into consideration. Zhand and Qiu investigated the effect of two sided purely longitudinal, transverse and isotropic surface roughness on hydrodynamic lubrication of dynamically loaded journal bearing. Surface appreciably effects the lubrication of surfaces if the film thickness is same order as the roughness. By using the Christensen's stochastic model, the steady state curves were obtained.

2 Effect of surface roughness

The importance of roughness in predicting bearing performance has gained considerable attention in tribology. All previous developments were based on perfectly smooth bearing surfaces. In reality however engineering surfaces are covered with asperities. Even for a ground surface, asperities might reach $1.25 \mu\text{m}$ in height and ten times this value in lateral spacing; the lateral distance we equate with the in plane characteristic length L_{xz} . The minimum film thickness in a journal bearing, say of diameter $D=25\text{mm}$ operating eccentricity $\epsilon=0.5$ is $h_{\min}=12.5$ this minimum thickness is selected here to represent the across the film characteristic length L_y . Because the average asperity height is one order of magnitude smaller than the minimum film thickness, we might be tempted to ignore surface roughness. However the local characteristic lengths are of the same magnitude, $L_y=L_{xz}=1.25 \mu\text{m}$ violating the thin film assumption of lubrication analysis, and it becomes questionable whether the Reynolds equation is at all valid. In case when the lubrication approximation still holds even though the surfaces are rough, we said to be dealing with

Reynolds roughness. When there is significant pressure variation across the film due to surface roughness, to the extent that the lubrication approximation is no, longer valid.

Tzeng and Saibel were among the first to apply statistical methods to lubrication of rough surfaces. They investigated the inclined plane slider having one dimensional roughness transverse to the direction of relative motion. This method employed by the investigators is based on the statistical averaging of Reynolds equation. As remarked by Tripp, the Reynolds equation has the property, unusual among the equations of mathematical physics that the boundary conditions are incorporated into the equation – it is this feature of Reynolds equation that offers for ensemble averaging the equation itself.

Christen and Tonder make two fundamental assumptions in their analysis:

1. The magnitudes of pressure ripples due to surface roughness are small and the variance of the pressure gradient in the roughness direction is negligible.
2. The flow in the direction transverse to roughness direction has negligible variance.

Governing equations

The basic equation that gives pressure distribution in bearings is Reynolds equation. Steps involved in obtaining solution are discussed.

Surface roughness

Consider two real surfaces with normal film gap h in the sliding motion. Local film thickness of h_T for this friction to be of the form

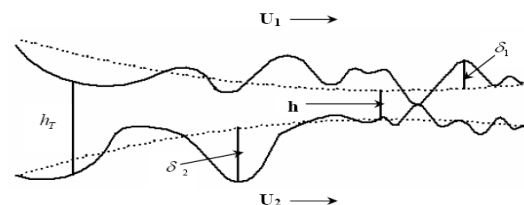


Fig. 1. Two rough surfaces in relative motion

$$h_T = h + \delta_1 + \delta_2 \quad (1)$$

The combined roughness $\delta = \delta_1 + \delta_2$ with variance

$$\sigma^2 = \sigma_1^2 + \sigma_2^2 \quad (2)$$

The equation for the average pressure

$$\frac{\partial}{\partial x} \left[\phi_{xx}^p \frac{h^3}{12\mu} \frac{\partial p_m}{\partial x} \right] + \frac{\partial}{\partial y} \left[\phi_{yy}^p \frac{h_T^3}{12\mu} \frac{\partial p_m}{\partial y} \right] = \frac{U_1 + U_2}{2} \frac{\partial \bar{h}_T}{\partial x} + \frac{U_1 - U_2}{2} \frac{\partial \phi_{xx}^S}{\partial x} + \frac{\partial h_T}{\partial t} \quad (3)$$

To study surfaces with directional properties the surface characteristic γ can be used. The parameter γ can be viewed as the length to width ratio of a representative asperity. There are mainly three sets of asperity patterns are identified purely

1. Transverse roughness pattern $\gamma < 1$
2. Isotropic roughness pattern $\gamma = 1$
3. Longitudinal roughness pattern $\gamma > 1$

Reynolds equation

A number of restrictive assumptions are introduced before starting with the present analysis of journal bearings. The assumptions are:

1. Inertia and body forces of the lubricant are neglected ;
2. Both bearing and the rotor are rigid;
3. The flow is laminar;
4. The no-slip condition exists at the fluid/solid interface;
5. Pressure variation across the film thickness is negligible;
6. The density of the lubricant is constant;
7. The operation of the bearing system is isothermal.

To investigate the effects of surface roughness on the steady state and dynamic characteristics of journal bearings, the modified Reynolds equation for hydrodynamic lubrication can be written as follows.

For steady loading and incompressible lubricants the generalized Reynolds equation becomes

$$\frac{\partial}{\partial x} \left(h^{n+2} \phi_{xx}^p \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left(h^{n+2} \phi_{yy}^p \frac{\partial p}{\partial y} \right) = 6\nu U^n \frac{\partial}{\partial x} (h + \sigma \phi_{xx}^S) \quad (4)$$

A developed view for half of the bearing is drawn in. the area is divided into a number of mesh sizes $(\Delta\theta \times \Delta y)$ and using central difference quotients, the equation can be written in the form as

$$\phi_{xx}^p \left[\frac{P_{i+1,j} - 2P_{i,j} + P_{i-1,j}}{\Delta\theta^2} \right] + \left(\frac{D}{L} \right)^2 \phi_{yy}^p \left[\frac{P_{i,j+1} - 2P_{i,j} + P_{i,j-1}}{\Delta y^2} \right] - \frac{3}{2} \varepsilon \left[\frac{P_{i+1,j} - P_{i-1,j}}{h_i(\Delta\theta)} \sin \theta_i \right] = -\varepsilon \frac{\sin \theta_i}{h_i^3} \quad (5)$$

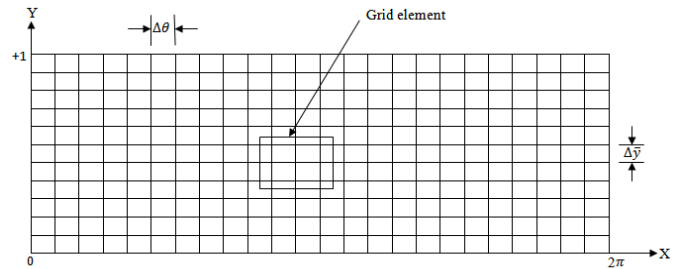


Fig.2: Finite difference scheme - Mesh

On simplifications the above equation becomes

$$P(i, j) = \frac{\phi_{xx}^p [P(i+1, j) + P(i-1, j)] + \left[\frac{D}{L} \right]^2 \left[\frac{\Delta\theta}{\Delta y} \right]^2 \phi_{yy}^p [P(i, j+1) + P(i, j-1)] + \left(\frac{-3\varepsilon}{2h} \right) [P(i+1, j) - P(i-1, j)] \Delta\theta \sin \theta + \left[\frac{\varepsilon \sin \theta}{h^3} \right] \Delta\theta^2}{2 \left(\phi_{xx}^p + \phi_{yy}^p \left[\frac{D}{L} \right]^2 \left[\frac{\Delta\theta}{\Delta y} \right]^2 \right)} \quad (6)$$

It is seen that the pressure at any mesh point (i, j) is expressed in terms of pressure at four adjacent points. To start the iteration process the pressure at all the mesh points are assumed and those at the boundaries are set.

Load carrying capacity of journal bearing:

The tangential and radial components of the load is written as

$$w_y = \frac{w_y c^2}{6\nu UR^2}; w_z = \frac{w_z c^2}{6\nu UR^2} \quad (7)$$

Total load carrying capacity is calculated as follows:

$$w_h = \sqrt{w_y^2 + w_z^2}; w_h = \frac{w_h c^2}{6\nu UR^2} \quad (8)$$

Attitude angle is calculated as

$$\phi = \tan^{-1} \left[\frac{w_z}{w_y} \right] \quad (9)$$

Sommerfeld number

$$S = \frac{\nu N_S DL}{w_h} \left(\frac{R}{C} \right)^2 \quad (10)$$

3 Results and Discussion

The steady state analysis of plain journal bearings has been carried out at various eccentric ratios with surface roughness for different L/D ratios. Attitude angle and sommerfeld number have been determined and finally compared with results in reference [2]. The results obtained have plotted for comparison for different surface roughness parameters.

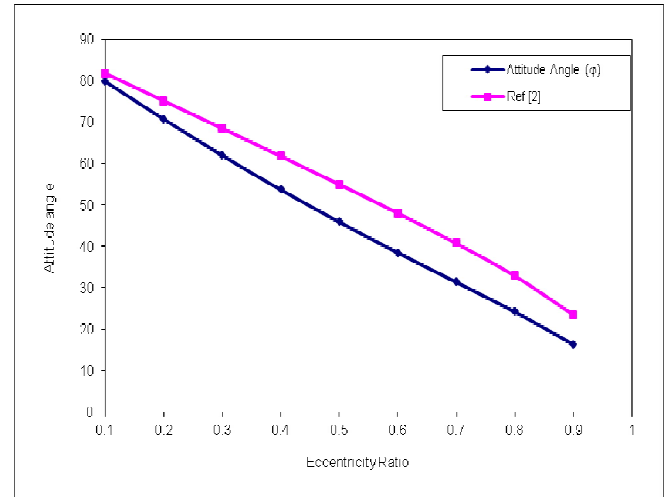


Fig.5: Variation of Attitude angle (ϕ) with Eccentricity ratio (ϵ) for isotropic surface roughness at L/D = 0.5

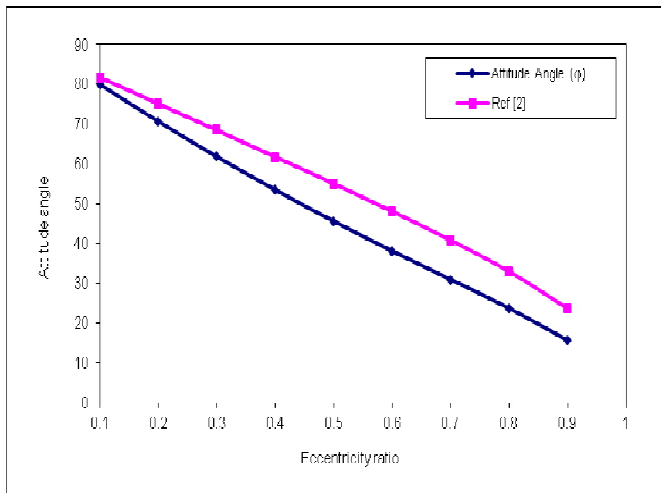


Fig.3: Variation of Attitude angle (ϕ) with Eccentricity ratio (ϵ) for transverse surface roughness at L/D = 0.5

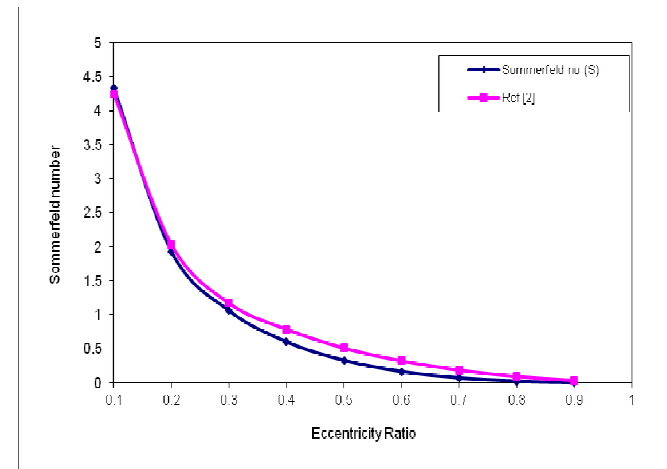


Fig.6: Variation of Sommerfeld no (S) with Eccentricity ratio (ϵ) for isotropic surface roughness at L/D = 0.5

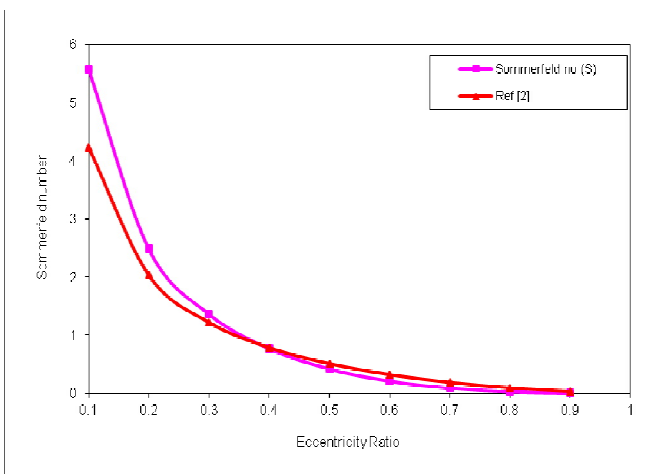


Fig.4: Variation of Sommerfeld no (S) with Eccentricity ratio (ϵ) for transverse surface roughness at L/D = 0.5

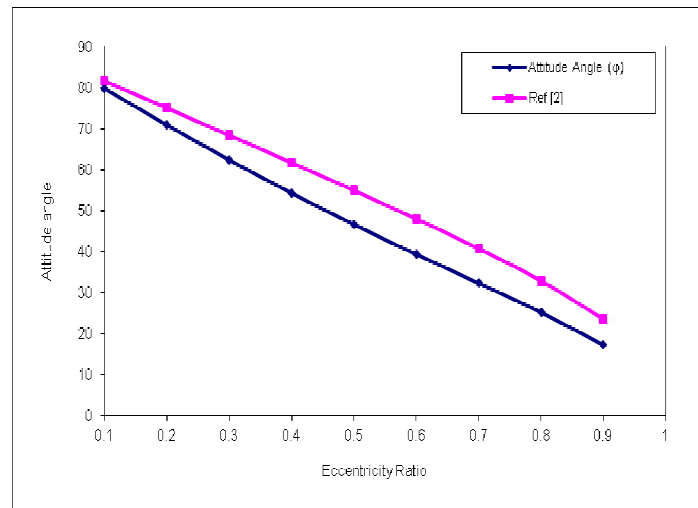


Fig.7: Variation of Attitude angle (ϕ) with Eccentricity ratio (ϵ) for longitudinal surface roughness at L/D = 0.5

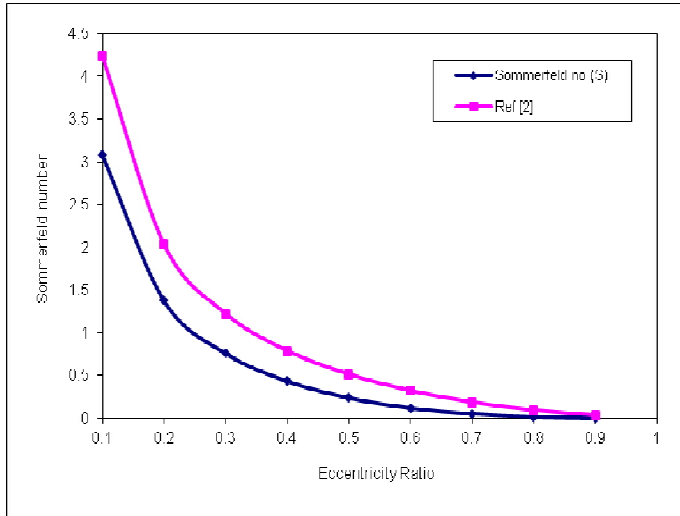


Fig.8: Variation of Sommerfeld no(S) with Eccentricity ratio(ϵ) for longitudinal surface roughness at $L/D = 0.5$

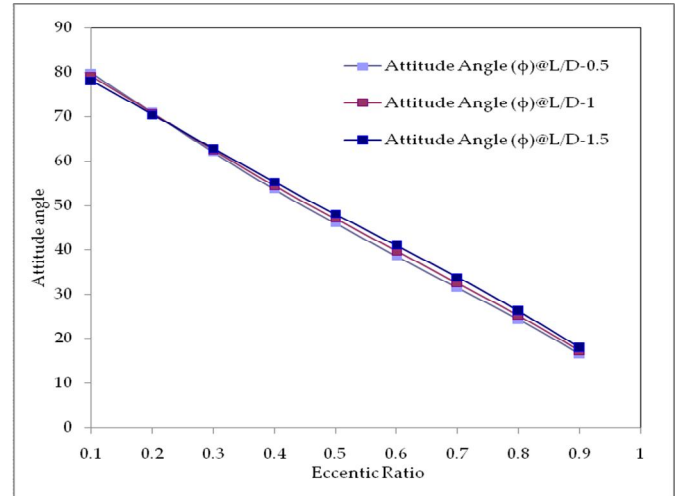


Fig.10: comparison of Attitude angle (ϕ) with Eccentricity ratio(ϵ) for isotropic surface roughness at all L/D ratio's

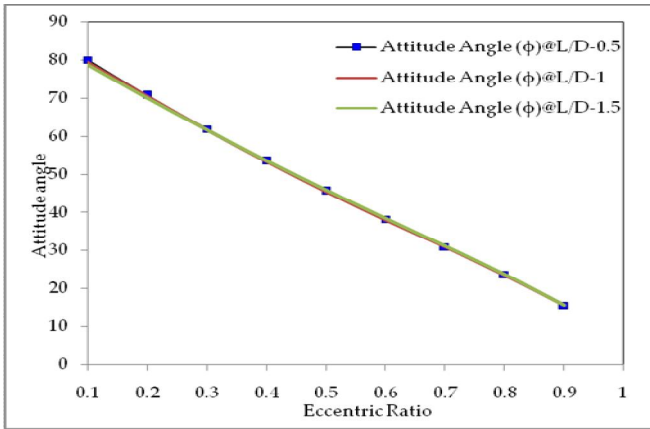


Fig.9: comparison of Attitude angle (ϕ) with Eccentricity ratio (ϵ) for transverse surface roughness at all L/D ratio's

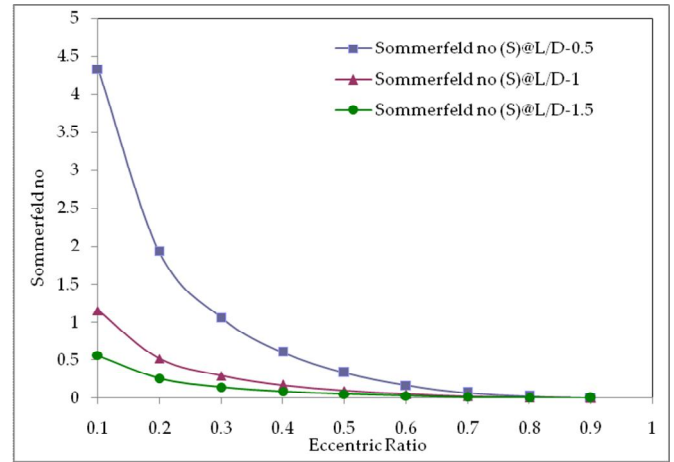


Fig.11: comparison of Sommerfeld no(S) with Eccentricity ratio(ϵ) for isotropic surface roughness at all L/D ratio's

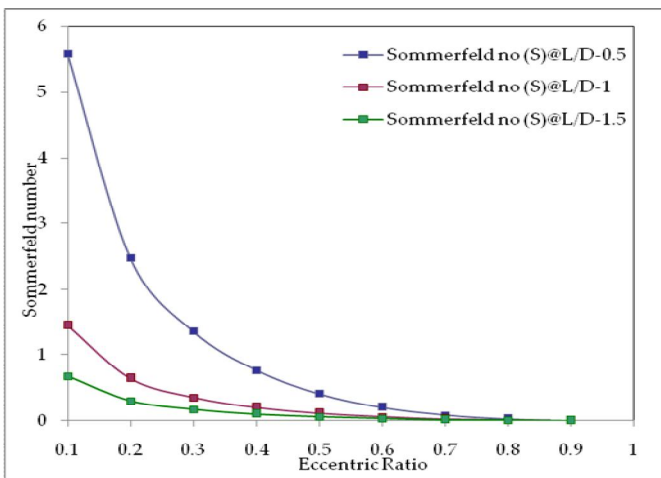


Fig.9: comparison of Sommerfeld no(S) with Eccentricity ratio(ϵ) for transverse surface roughness at all L/D ratio's

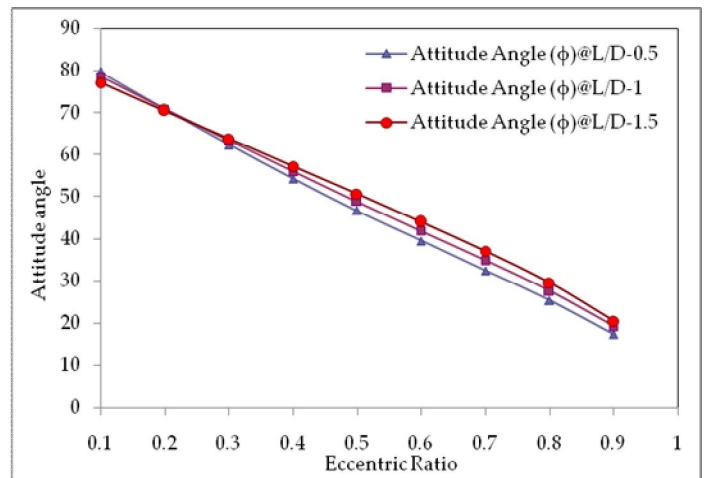


Fig.12: comparison of Attitude angle(ϕ) with Eccentricity ratio(ϵ) for longitudinal surface roughness at all L/D ratio's

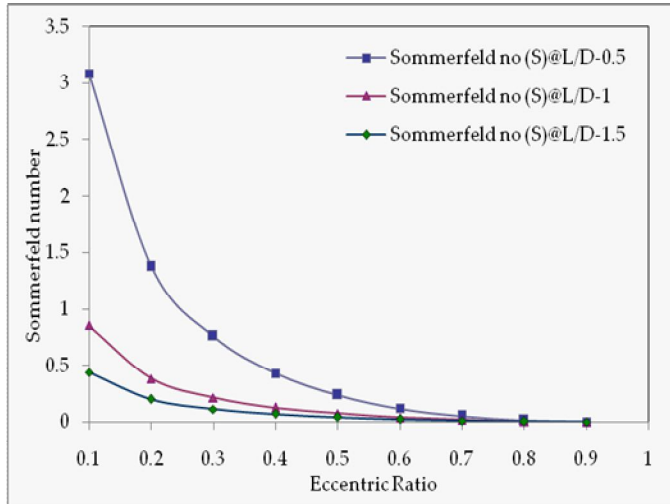


Fig.13: comparison of Sommerfeld no(s) with Eccentricity ratio(ϵ) for longitudinal surface roughness at all L/D ratio's

Analysis of plain journal bearing:

The work has been further extended in modeling the plain journal bearing using GAMBIT software. The meshed model is analysed using ANSYS-fluent. The isotropic roughness for the plain bearing is applied and analyzed for different eccentric ratios to obtain the pressure distribution contours of plain journal bearing with surface roughness. The variation of pressure contours in the plain journal bearing is obtained shown in figure 14 to 18. The sommerfeld number is determined from the pressure distribution curves for the different eccentric ratios. The variation of sommerfeld number with eccentric ratio has been compared with numerical method and fluent analysis results for L/D ratio=1 and with isotropic roughness. It has been observed that the pressure increased with increase in eccentric ratio and the sommerfeld number decreased with increase in eccentric ratio.

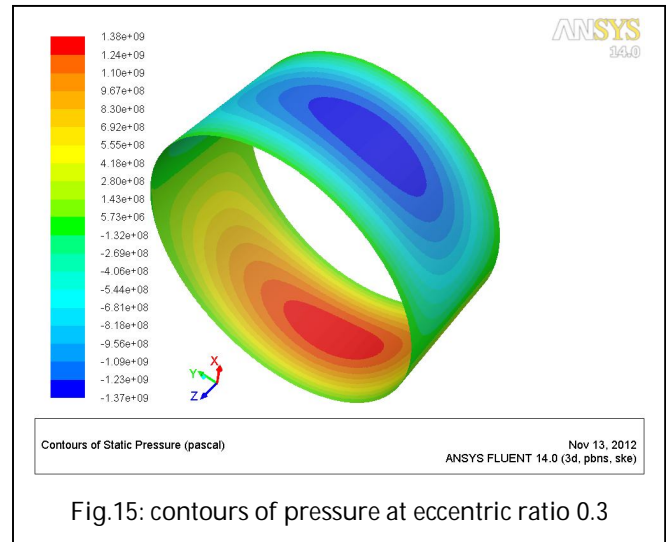


Fig.15: contours of pressure at eccentric ratio 0.3

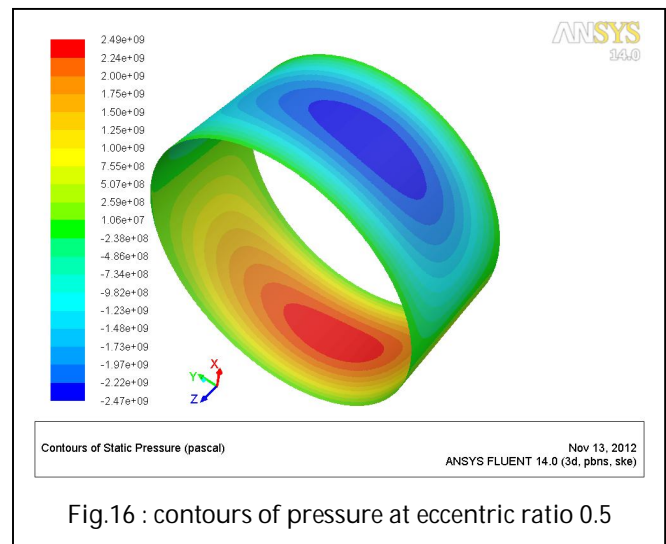


Fig.16 : contours of pressure at eccentric ratio 0.5

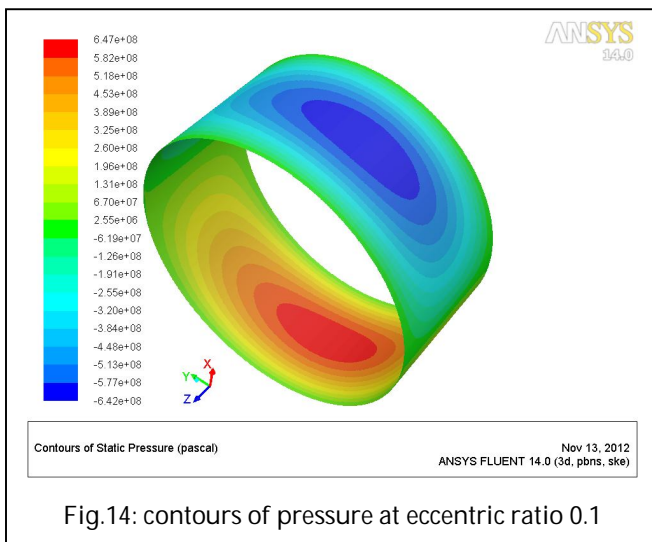


Fig.14: contours of pressure at eccentric ratio 0.1

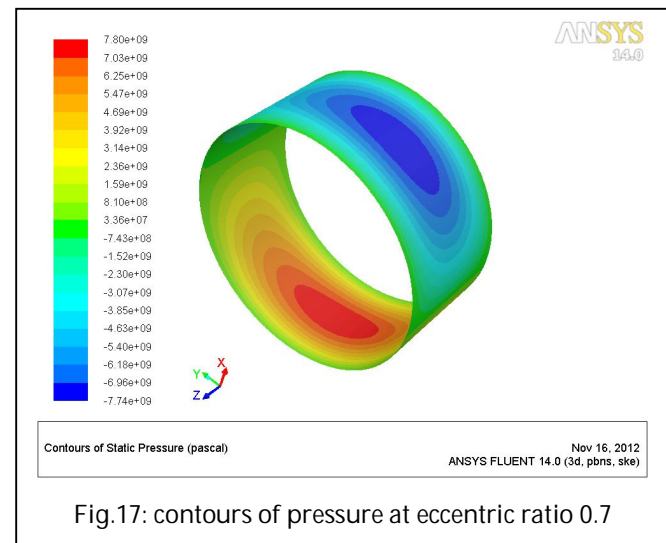


Fig.17: contours of pressure at eccentric ratio 0.7

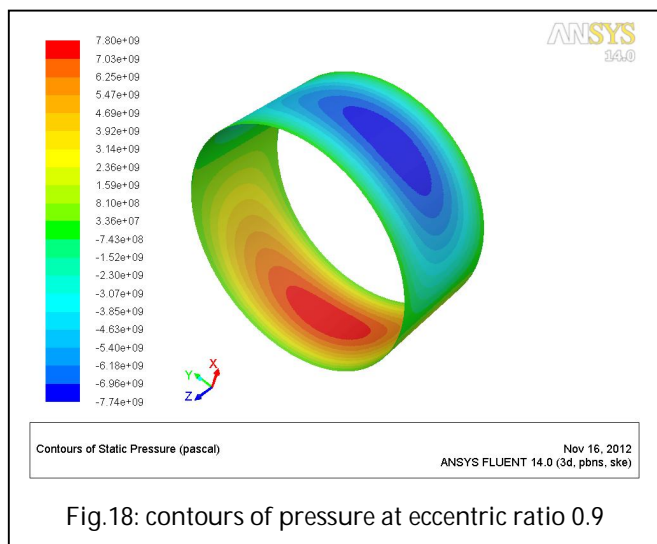


Fig.18: contours of pressure at eccentric ratio 0.9

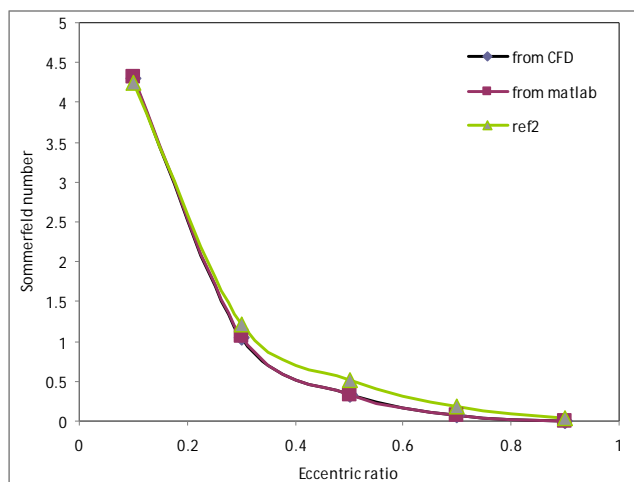


Fig.19: Comparison of sommerfeld number with eccentric ratio

4 CONCLUSION

The static characteristics of the finite bearings for various L/D ratios are determined for three types of the surface roughness orientations. Longitudinal surface roughness has greater effect on the Sommerfeld number irrespective of the L/D ratio. Sommerfeld number and eccentricity are plotted for three surface roughness orientations at different L/D ratios. The load carrying capacity is increased by 52.57% compared with the bearing having zero surface roughness. It is the clear evidence that the surface roughness has the effect on the performance of static characteristics of the finite bearings. From the graphs it is observed that the calculated values are deviated from the actual values.

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Nomenclature

Symbol	Parameter
h_T	nominal film thickness
h	distance between the mean levels of the two surfaces
δ_1, δ_2	random roughness amplitudes of the two surfaces
n	power-law index of the non-Newtonian fluid
p	Pressure
ν	constant of the power-law fluid
U	circumferential velocity of the journal surface
ϕ_{xx}^p, ϕ_{yy}^p	pressure flow factors in the x and y directions
ϕ_{xx}^s	shear flow factor in the x direction.
σ	standard deviation of the composite roughness surfaces with standard deviations σ_1 and σ_2
$p_{i,j}$	pressure at any point (i, j)
h_i	film thickness at any point (i, j)
N_s	speed of the journal
C	radial clearance
R	radius of the bearing
L	length of the bearing

