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RESEARCH

Influence of High Permeability Parameter on the Performance of Textured Porous Journal Bearings

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A B S T R A C T

The addition of different shapes and sizes of surface textures (rectangular, triangular, square, spherical, elliptical etc.) inside the bearing surface successfully improved its tribological properties for conventional bearings to a great extent as reported by various research groups. The present study involves the incorporation of spherical shaped textures to test the performance of porous journal bearings by taking into account the position of the textures at different eccentricity ratios (0.3 & 0.7). The originality of the work is to investigate the influence of permeability parameter on textured bearing. A finite difference method with central differencing scheme has been adopted for the solution of the governing equations i.e., Reynolds equation and Darcy's equation. The permeability factor plays a key role in computing the results. It has been observed from the outcomes that the location of the spherical textures and variation of permeability parameter showed significant influence on the bearing performance characteristics. Textures incorporated at the different locations (convergent zone & maximum pressure region) based on eccentricity ratio significantly helps to uplift the performance of the non-porous and porous journal bearings.

1. INTRODUCTION

Now-a-days, the introduction of micro surface texturing/ dimpling on tribological elements has successfully recognized for enhancing the performance of the elements which eventually improve the life of the machine elements. A lot many researches have been conducted to authenticate the advantages of texturing [1-6]. The texture patterns produced on the surface serves as innumerable micro-fluid film bearings. In one study, the influence of positive micro grooving at different locations on the journal bearing surface has been investigated by Kango and Sharma [7]. The authors concluded that the load carrying capacity was found considerably high in the minimum film thickness region for optimum eccentricity ratio. Similar research group reported a comparative study of sinusoidal, full and half wave positive micro grooving. The observations showed that longitudinal

sinusoidal roughness was found to be best among all. In another study, Kango et al. [8] reported in their studies that the half wave texture enhances the bearing performance more in comparison to full wave textures on the journal bearing surface.

In the study of hydrodynamic slider and journal bearings by Tala-Ighil et al. [9-10], the effect of surface texture was investigated. It was established from the results that the film thickness, pressure distribution, side leakage, frictional torque etc. are significantly influenced due to the presence of surface textures. It has also been reported that the attributes of dimples such as size, depth, density, and orientation etc. meaningfully affect the bearing characteristics. Cupillard et al. [11], Brizmer and Kligerman [12] have studied the roles of different surface dimpling in their respective studies under certain considerations for hydrodynamic journal bearings. The authors revealed that the partial surface texturing has a big influence on the performance characteristics of the journal bearings.

The combined effect of surface textures and non-Newtonian lubricants was also a very important concern which was reported by numerous researchers. Kango et al. [13] investigated the micro-textured journal bearing including non-Newtonian fluid effects and JFO (Jakobsson, Floberg and Olsson) boundary conditions. The thermal analysis (viscous heat dissipation) was also taken into consideration and found a noteworthy reduction in the average temperature of the lubricant in case of textured surface. In another study, Kango et al. [14] also gave a comparison of dimpled and grooved surface and investigated the influence on journal bearings. In their findings, they established that micro-grooved texture reduces the average temperature and friction coefficient in comparison with spherical texture. Sharma et al. [15] studied the combined effects of spherical textures and couple stress fluids for a finite journal bearing with JFO boundary conditions and concluded that load carrying capacity get increased with couple stresses at different eccentricity ratios. However, with textures the increment was found prominent only at low eccentricity ratios.

In a study, Meng et al. [16] used fluid structure interaction (FSI) technique to investigate the effect of compound shaped viz., rectangular-spherical micro-dimples on the performance characteristics of journal bearing. The authors recognized that the addition of compound shape dimples give noteworthy increment in the load carrying capacity and reduction in the friction. Authors also recommended that a suitable selection of geometric features and operating conditions are very important to achieve efficient performance of bearing. Gropper et al. [17] reviewed the global research efforts on surface texturing with different modelling techniques for fluid flow, cavitation and micro-hydrodynamic effects. The authors observed that for wide-ranging variety of applications, surface texturing with optimal texture parameters and operating conditions has been proven to be capable of enhancing the tribological performance.

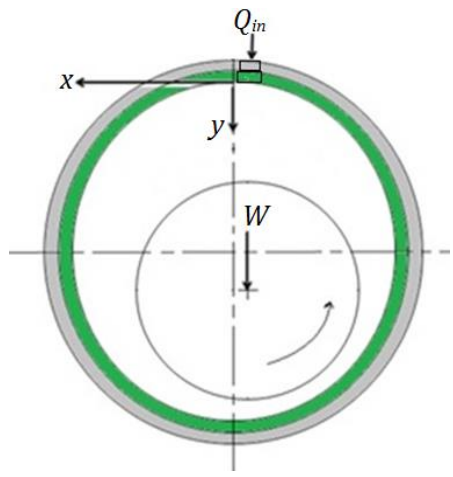
Permeability is a key parameter for testing the performance of porous bearings. Some studies on textured porous bearings are also available in literature. Sharma et al. [18-19] added sinusoidal wave textures with couple stress fluids as working lubricant at the full region as well as in the segmental zones of a porous journal bearing. The combined influence was surprisingly remarkable for the enhanced bearing performance characteristics. In another work, authors presented a comparison for two non-Newtonian fluid models for textured bearing. A numerical study was also given by Lee et al. [20] and investigated the lubrication features of a textured porous bearing where the type of textures viz., semi-spherical and semi ellipsoidal shapes were employed with variation of texture dimensions. Out of two, the semi-spherical dimple showed high rate of increase in load capacity than semi-ellipsoidal dimple and is opposite in case of equivalent friction coefficient. In recent studies by Some and Guha [21-22], the combined effects of shaft misalignment and non-Newtonian lubricant were investigated in case of a double-layered porous journal bearing. They found that high degree of misalignment contributes to higher film pressure and due to couple stress lubricants, these pressures enhances more. Recently, Taguchi based Grey relational analysis was adopted for the multi-

objective optimization for performance development by Shinde et al. [23]. A number of groove parameters viz., groove location, height, width; number of grooves and spacing between grooves were taken into consideration. It has been established that groove height and location are the most capable parameters for improving the load carrying capacity. For reduction of friction coefficient, groove location and numbers of grooves are the main contributors. It has also been noticed in a recent numerical study by Khatri et al. [24] that the textured non-recessed journal bearings improved the bearing performance appreciably. An experimental study was also reported by Galda et al. [25] for the investigation of surface textures on the performance of journal bearings. Due to the oil pockets on the mating surface, the textured journal bearings during shut down remain in hydrodynamic lubrication regime whilst compared with smooth one which comes in boundary lubrication. The friction torque was improved by 74% for the textured series as compared to smooth case. A numerical survey was reported by Manser et al. [26] on the performance of hydrodynamic journal bearing with twin effect of textures and misalignment. The authors clearly depicted that with right position of textures with geometries considerably enhance the performance. In some recent published studies, Bhattacharjee et al. [27] derived the modified Reynolds equation in the case of micropolar fluid with a numerical solution to investigate single layered porous short bearing characteristics. It has been established from the results that the micropolar fluids considerably improves lubrication and load capacity as compared with the Newtonian fluids. A comparison of surface texture patterns on the performance of gas seals has been explored by Shi et al. [28]. In their findings, texture shapes with a straight type edge perform better in maximizing the evaluating parameters in comparison with a round type edge. In another recent study, Syed and Sarangi [29] presented the lubrication performances of parallel sliding contacts with square shaped textures. It has been observed that the fluid-slip show progressive influence on the lubrication performance, however inertia reduces the performance. Moreover, the lower values of aspect ratio and texture height ratio showed considerable improvement in the hydrodynamic performance as compared to conventional and fluid inertia conditions.

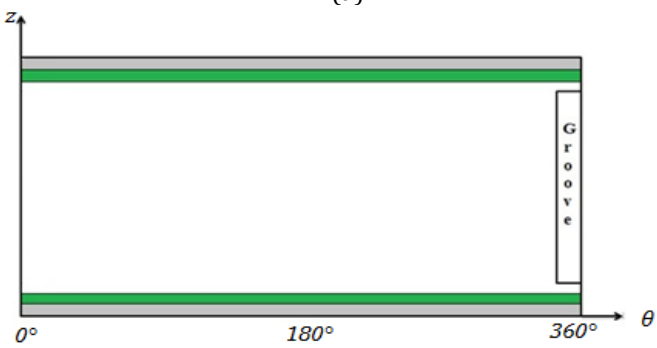
The main motivation for this study is to find the best position of texture at high eccentricity ratio taking high values of permeability parameter into account. Thus, the main objective of this work is to present a numerical model for the analysis of hydrodynamically lubricated porous journal bearing having spherical textures on the bearing surface by incorporating Reynolds boundary conditions at low & high values of eccentricity ratio and permeability parameter.

2. MATHEMATICAL MODELLING AND COMPUTATIONAL PROCEDURES

The Schematic illustration of a conventional porous journal bearing has been shown in Fig. 1a. The x-direction represents circumferential length of the bearing denoted by $R\theta$, y-represents film thickness direction denoted by H and z-represents length of the bearing denoted by L. Fig. 1b depicts the general arrangement of porous journal bearing in opened form.



(a)



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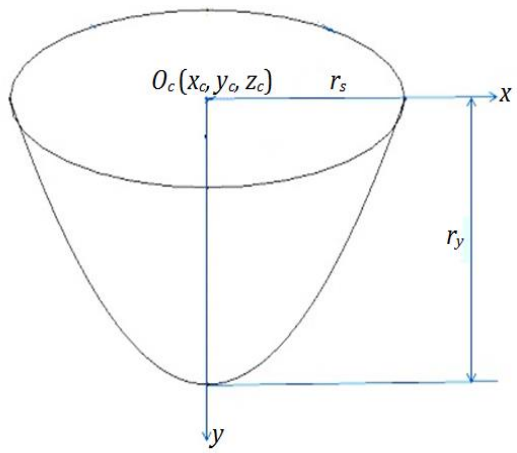


(c)

Fig. 1. (a) Schematic illustration of conventional porous journal bearing; (b) Unwrapped porous journal bearing, and (c) Schematic illustration of 3-D spherical shape of dimple.

From the literature review, it has been successfully proven that partial textures are successfully enhancing the journal bearing performance [7-8, 13-14]. Thus, referring to the work of Kango et al [13], location of textures has been taken in a restricted region (0° to 128°).

Interestingly, with the previous reported work related to porous journal bearing [18-20], it has been confirmed that the inclusion of texture helps in increasing the bearing performance. Porous bearings are also known as low load bearings. Due to the reason of their low eccentricity ratios and porous nature, the chances of cavitation are very less. So, in order to undervalue the pressures due to micro-cavitation mechanism inside the dimples, the JFO boundary conditions were not taken into account and Reynolds boundary conditions have been adopted in this study.

Further, the methodology and equations associated with texture work has also been taken with reference to the work of Kango et al [13-14].

For consideration of textures (Number of dimples) and size of texture, the film thickness equations are reframed as given below:

$$h = C_r(1+\varepsilon\cos\theta) \tag{1}$$

Where h, ε and C_r represents the film thickness of smooth bearing, eccentricity ratio, and radial clearance respectively.

$$H_T = C_r\,(1+\varepsilon\cos\theta) + \Delta h \tag{2}$$

$$\Delta h = \frac{r_y}{r_s}\sqrt{(r_s)^2 - (R\theta - x_c)^2 - (z - z_c)^2} \tag{3}$$

H_T is lubricant film thickness for textured bearing, while Δh denotes the film thickness to measure the spherical texture. r_y, r_s, R, x_c and z_c denote dimple depth, dimple radius, shaft radius, centre of dimple in x-direction and centre of dimple in z direction respectively as shown in Fig. 1(c). The expressions for x_c and z_c have been adopted from [13]. Dimple centre O_c (x_c, y_c, z_c) has also been indicated in Fig. 1(c). The centre of the dimple is located on the surface of the bearing (i.e. $y_c = 0$). The expressions for x_c, and z_c are written as:

$$x_c = na + \frac{(2n-1)}{2}L_x, \quad z_c = nb + \frac{(2n-1)}{2}L_z \tag{4}$$

Where: n is number of dimples i.e. 1, 2, 3…….

L_x and L_z are the length of unit cell in circumferential and axial direction respectively. The arrangement of spherical textures by using the location (0° to 128.5°) and the used terminology for unit cell is shown in Fig. 2 adopted from the work of [13].

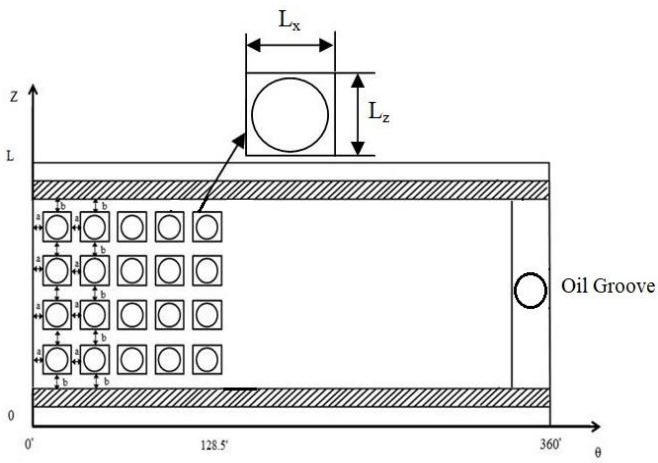


Fig. 2. Schematic illustration of an unwrapped textured porous journal bearing.

The generalized Reynolds equation (steady state incompressible laminar flow) for a textured porous journal bearing subsequent to the introduction of dimples can be written as [19]:

$$\frac{\partial}{\partial x}\left(\frac{H_T^{\,3}}{\mu}\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(\frac{H_T^{\,3}}{\mu}\frac{\partial p}{\partial z}\right) = 6U\frac{\partial h}{\partial x} - 12\varphi H\left(\frac{\partial}{\partial x}\left(\frac{1}{\mu}\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(\frac{1}{\mu}\frac{\partial p}{\partial z}\right)\right) \tag{5}$$

Where: p, H, U, φ and μ are hydrodynamic lubricant pressure, wall thickness for porous

region, velocity of journal, permeability and dynamic viscosity of the lubricant respectively. In the numerical solution, an equation (5) is simultaneously solved by the finite difference approach to yield the pressure profiles for smooth & textured bearings. The discretized equation is solved by the Gauss-Seidel iterative method combined with the over-relaxation factor to speed up the convergence for the values of lubricant pressure. Results of grid independence and convergence tests were not reported in the manuscript under the impression that these are pre-requisites of a numerical computation done by FDM. Furthermore, the convergence criteria used in these computational procedures are as follows:

For Pressure: $\sum\sum\left|\frac{(p_{i,k})_I-(p_{i,k})_{I-1}}{(p_{i,k})_I}\right|<10^{-7}$

The expressions for percentage change in load carrying capacity (W), attitude angle (ϕ), axial fluid flow (Q) and coefficient of friction (COF) can be given by the following equations:

$$\%W = \frac{W_{textured}-W_{smooth}}{W_{smooth}} \text{x}\,100$$

$$\%\phi = \frac{\phi_{textured}-\phi_{smooth}}{\phi_{smooth}} \text{x}\,100$$

$$\%COF = \frac{COF_{(textured)} - COF_{(smooth)}}{COF_{(smooth)}} \times 100$$

$$\%Q = \frac{Q_{textured}-Q_{smooth}}{Q_{smooth}} \text{x}\,100$$

As previously mentioned, the Reynolds boundary conditions have been taken for this proposed model for computation of pressures as adopted by [7-10].

(i) $p(0°, z)= 0 = p(360°, z)$ & $p(\theta, 0)= 0 = p(\theta, L)$

(ii) $\frac{\partial p}{\partial \theta}=\frac{\partial p}{\partial z}=0$ and p = 0 at the rupture limits of the film lubricant i.e. whenever the pressures become negative at any mesh point in the iteration, pressures as well as pressure gradients in both the directions (x, z) should be made equal to zero.

3. RESULTS AND DISCUSSION

A numerical analysis has been presented in this study. A MatLab code was developed based on the mathematical analysis which is described in section 2. The developed computational code is successfully verified and validated with the results obtained with the work of Reason and Dyer [30] and is shown in Fig. 3.

For the validation of textured model, authors used all the conditions (equations, input variables etc.) from the work of Tala-Ighil et al. [10] as shown in Table 1 for comparison purpose. The calculated results have shown clear evidence that the computational code prepared is obtaining a good agreement with the work of Tala-Ighil et al. [10]. After validating the model, numerical results have been calculated for various bearing performance parameters.

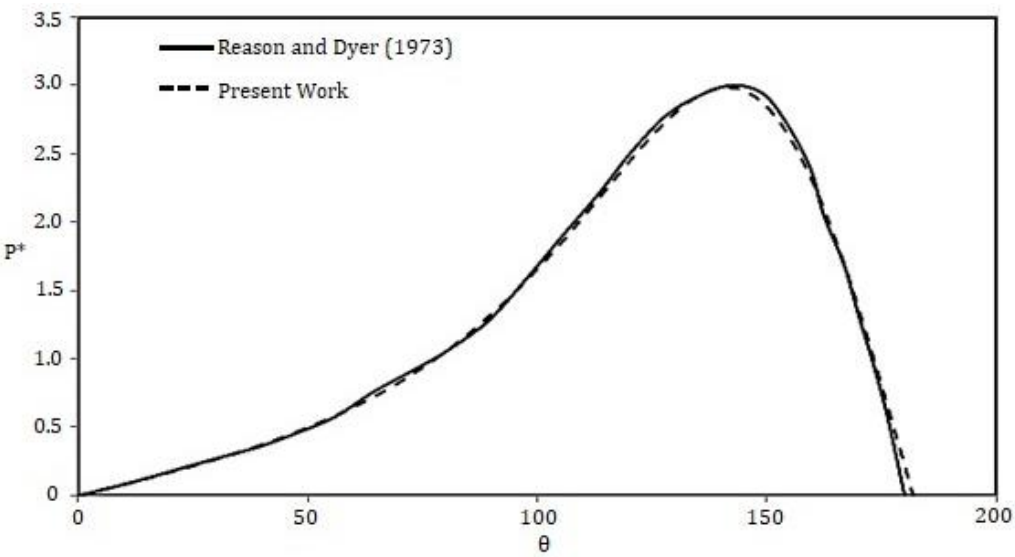


Fig. 3. Validation of dimensionless mid plane pressure P^* distribution as a function of (θ) with the work of Reason and dyer [30].

Table 1. Validation of the model.

Tala-Ighil et. al [10]			Present Work		
Case	p_{max} (MPa)	Friction Torque (Nm)	Case	p_{max} (MPa)	Friction Torque (Nm)
0	7.71	1.217	0	7.70	1.202
4	8.40	1.394	4	8.11	1.374
7	7.89	1.224	7	7.79	1.219

Table 2 presents the input data for the calculation of numerical results for textured and smooth porous journal bearings. An extensive range of permeability parameters are taken here into account (0 to 1). With reference to a number of research groups, in most studies, a lower permeability range has been adopted i.e., from 0 to 0.1 [18-19, 30-33]. However, in this work the permeability parameter range has been increased 10 times up to the value of 1. Two values of the eccentricity ratio have been taken, one is the lower value (0.3) and other is higher (0.7).

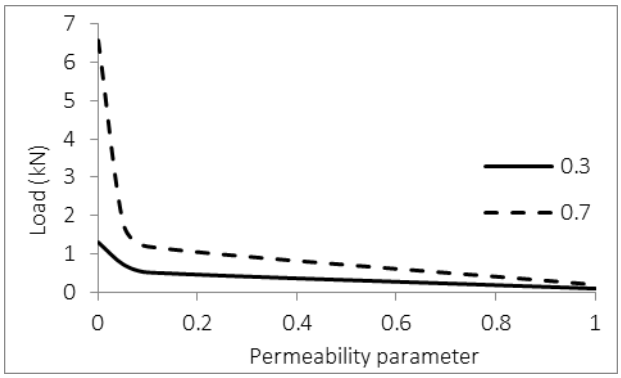
Table 2. Input Parameters.

Parameter	Value
Eccentricity ratio (ε)	0.3 and 0.7
Shaft speed(N), rpm	1500
Radial Clearance (C_r), m	50 x 10^{-6}
Shaft Radius(R), m	0.02
Bearing length(L), m	0.04
Lubricant dynamic viscosity Pa-s	0.08
Nodes in circumferential direction (N_θ)	223
Nodes in axial direction (N_z)	72
Dimple radius(r_s), m	0.003
Dimple depth(r_y), µm	20
Area density of dimple (S_p)	0.65
Length of unit cell in circumferential direction (L_x), m	0.0076
Length of unit cell in axial direction (L_z), m	0.0076
Distance between two dimples in circumferential direction (a), m	0.001132
Distance between two dimples in axial direction(b), m	0.001689
Theta (θ), °	0-360

The trends for load capacity (W), attitude angle, coefficient of friction (COF) and axial flow have been shown with respect to the variation in permeability parameters for smooth porous bearing case. It is well known fact that the permeability deprives the load capacity which causes the main reason for the increase in attitude angle as shown in Fig. 4 (a-b).

It is also well-known fact that with increase in eccentricity ratio, the load carrying capacity of the bearing increases and attitude angle decrease. So, in this case permeability plays a crucial role in depriving load capacity and increasing attitude angle. It is also observed here that the higher values of permeability parameter and high eccentricity ratio gives almost same values in the case of load and attitude angle i.e., at Ψ=0.8 to 1.0, the load capacity values for both low and high ε (0.3 and 0.7) is decreasing in the range of approximately 0.5 kN to 0.2 kN.

Similarly, in the case of attitude angle, at higher values of Ψ, the smaller range (77° to 80°) is observed. This can be concluded that very high values of Ψ does not contributing much in the evaluation of smooth porous journal bearing performance. The permeability parameter (Ψ) significantly influenced the coefficient of friction for both eccentricity ratios as shown in Fig. 4c. At low eccentricity ratio, the maximum value of COF is observed to be around 0.30 at Ψ=1. However, in case of high eccentricity ratio, the maximum value of COF is around 0.15, which is almost half. This might be happened due to the presence of porosity in the bearing which tends to decrease the lubricant pressure and thus decrease the load capacity. Further increase in Ψ may increase the coefficient of friction (COF) at a faster rate.



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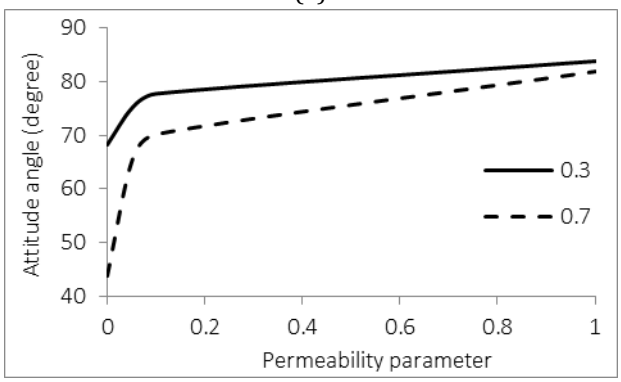

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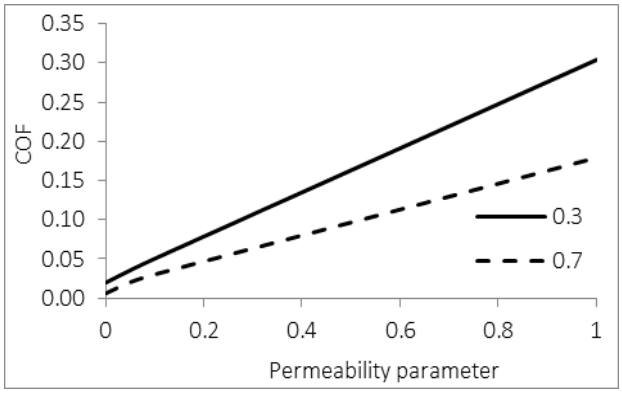

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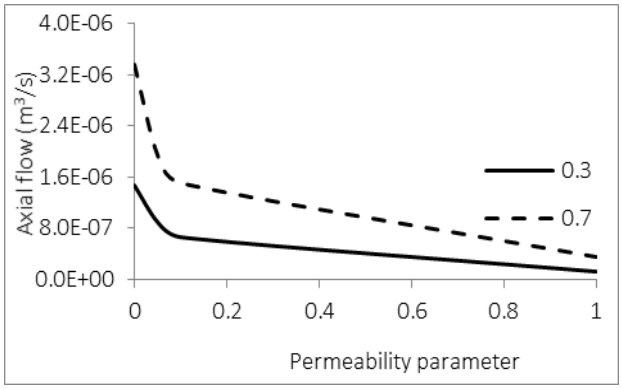
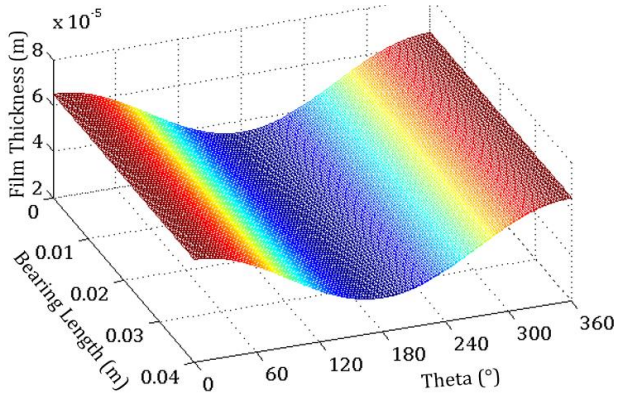


(d)

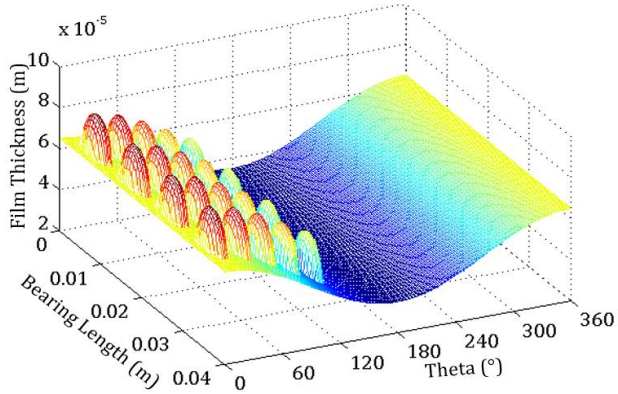
Fig. 4. Effect of permeability parameter on the performance characteristics viz., (a) Load, (b) Attitude angle, (c) Coefficient of friction and (d) Axial flow of smooth porous journal bearing at low (0.3) and high (0.7) eccentricity ratio.

Figure 4d shows the trends for axial fluid flow, as it is clear from the figure that the permeability parameter significantly reduced the axial flow. Moreover, the overall values of axial fluid flow have been found higher for high value of ε=0.7. It has been observed from the above discussion that the high value of permeability parameter always deteriorates static porous bearing performance characteristics.

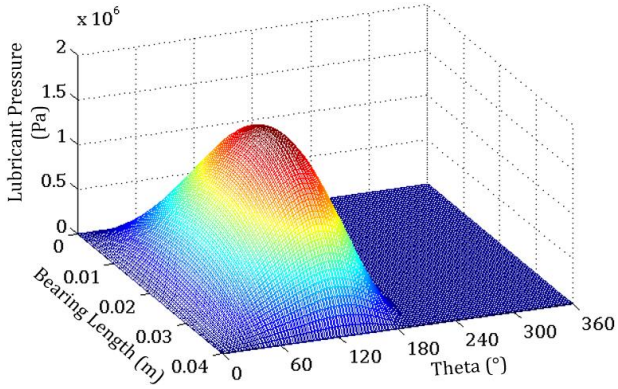
In order to know the effect of partial textures, an arrangement of (5×4) numbers of dimples in bearing surface has been considered in the convergent zone. This is considered due to the reason that the micro-textures at convergent zone initiate the variation in film thickness which gives escalation in film pressures and ultimately give a better hydrodynamic load capacity. In order to confirm the competence of the prepared computational code, some predicted results for the depiction of the evolution of lubricant film thickness and film pressure for smooth and partial textured surface is shown in Fig. 5 (a-d).



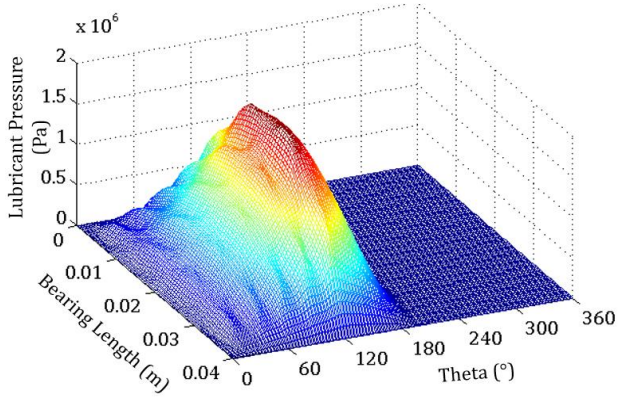
(a) Smooth case



(b) Partial spherical texture case



(c) Smooth case



(d) Partial spherical texture case

Fig. 5. Lubricant film thickness profiles (a-b) for smooth and textured journal bearing and lubricant pressure profiles (c-d) evolution for smooth and textured journal bearing

Figures 5a and 5b show the profile of smooth and spherical partial surface texture of the hydrodynamic journal bearing. As clearly depicted in the figures for film thickness that inclusions of textures clearly change the trends compared to smooth case. The dimple depth is kept constant throughout. The textures added in the convergent zone will give a supplementary enhancement to increase the lubricant pressure as shown in pressure profiles of Fig. 5d compared to profiles shown in Fig. 5c. However, in case of high eccentricity, texture has not shown beneficial results as confirmed by the work of numerous authors [9-10, 14-15].

Figure 6 shows the results in the form of percentage variation with reference to smooth bearing case for various bearing performance characteristics. The results are calculated for both low and high values of ε with variation in Ψ. It is well known fact that the partial textures enhance the bearing performance at low eccentricity ratio because the minimum film thickness is large which can support high load.

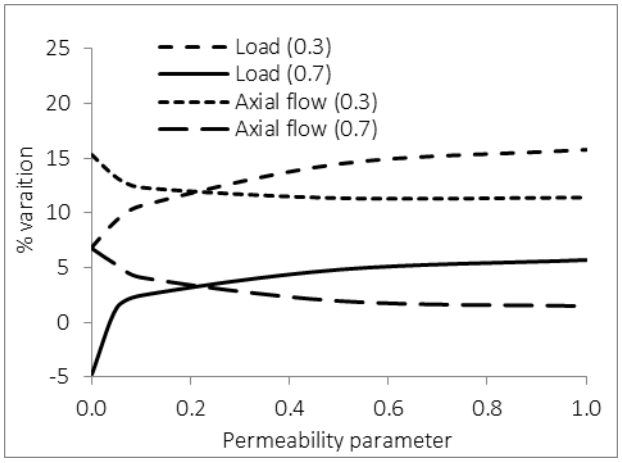


Fig. 6. Effect of permeability parameter on percentage change in Load (W) and Axial flow (Q) on spherical textured porous journal bearing at low and high eccentricity ratio.

It was interesting to observe from Fig. 6 that the increase in Ψ increases the percentage change of load carrying capacity (W) & reduces the percentage change of axial fluid flow for both values of eccentricity ratios. For low eccentricity ratio, positive percentage variation ranges from 7 % to 15 % (Load) & 15 % to 10 % (axial flow) approximately for Ψ=0 and 1, respectively. For high eccentricity ratio, the load carrying capacity was initially found a percentage decrease of about 5 % for conventional bearing. However, with the increase in Ψ, this percentage change shifted into incremental side from approximately 2.5 % to 5 %.

Moreover, for axial flow the positive percentage variation was noted 7 % in the case of conventional textured bearing & it get decreased at around 2 % for high permeability parameter.

Figure 7 shows the percentage variation of COF and attitude angle with permeability parameter for both values of eccentricity ratios. As it is well known that the COF is the ratio of friction force and load, which is the main reason for the decreasing trends of percentage variation in COF. The attitude angle is also calculated based on load carrying capacity. Therefore, the decreasing trends of variation of attitude angle have been observed from the Fig. 7. From the above discussion, the synergetic effects of surface texturing and permeability parameter on the performance characteristics for porous journal bearing have been observed. The most probable reason for these kinds of development might be due to increase in permeability parameters which leads to increase pores on the inner surface of journal bearings and further develop smaller fluid film pressures. However, because of the presence of spherical textures in the convergent location, this elevates the fluid pressures and ultimately improves the load carrying capacity not even at low eccentricity ratio but also at high eccentricity ratio. It has been observed from above discussion that surface texturing is also fruitful for porous journal bearing at high eccentricity ratios for this textured location.

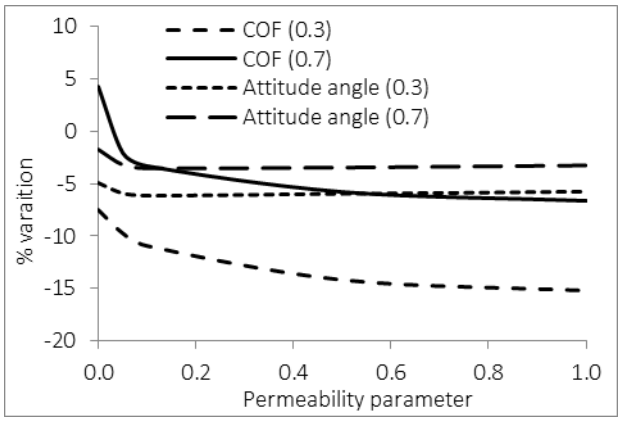


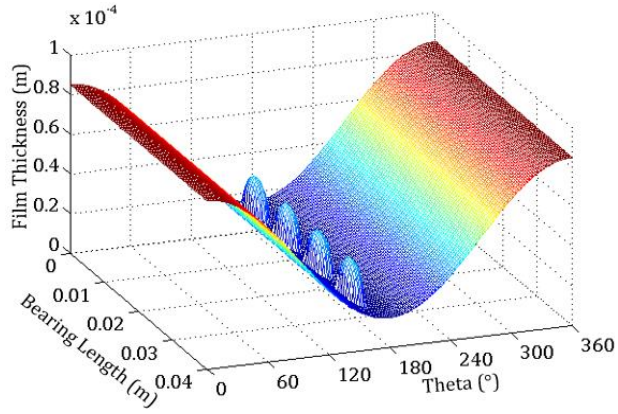
Fig. 7. Effect of permeability parameter on percent change in Load (W) and Axial flow (Q) on spherical textured porous journal bearing at low and high eccentricity ratio.

Furthermore, Cupillard et al. [11], Kango et al. [8,14] concluded that the bearing performance parameters for conventional journal bearings are also enhanced with texturing at high eccentricity ratios due to the change of location of textures. Authors observed that if the textures are placed at maximum pressure region for high eccentricity ratios, there is improvement in the bearing performance. However, the study related to this concept has not been found in the available literature in the case of textured porous bearing. Therefore, further study is based on this thought. Five different locations (Location 1 to 5) have been taken into consideration with the combination of different number of dimples. The lubricant film thickness evolution for all the considered locations has been depicted in Fig. 8. In relation to the porous journal bearing with textured surfaces at different locations and dimples, the most important bearing characteristics viz., load capacity and coefficient of friction are computed and listed in Table 3.

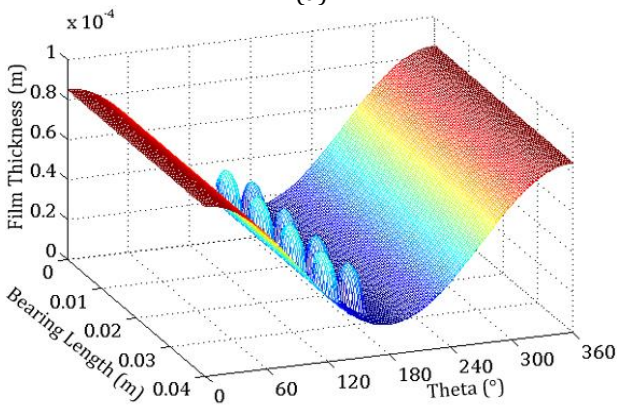
Table 3. Effect of different texture locations and permeability parameters (PP) on various bearing performance characteristics (ε =0.7).

Theta (degree)	Number of dimples in circum-ferential direction	Performance parameters	% variation	
			Ψ =0 (Non-porous)	*Ψ= 1 (Porous)*
172°-153°	1	Load	+7.74	-2.87
		COF	-8.04	+0.74
172°-134°	2	Load	+9.81	-5.01
		COF	-10.00	+1.73
172°-115°	3	Load	+7.37	-5.72
		COF	-8.24	+1.86
172°-96°	4	Load	+6.18	-4.95
		COF	-7.40	+0.69
172°-77°	5	Load	+5.90	-3.19
		COF	-7.24	-1.32

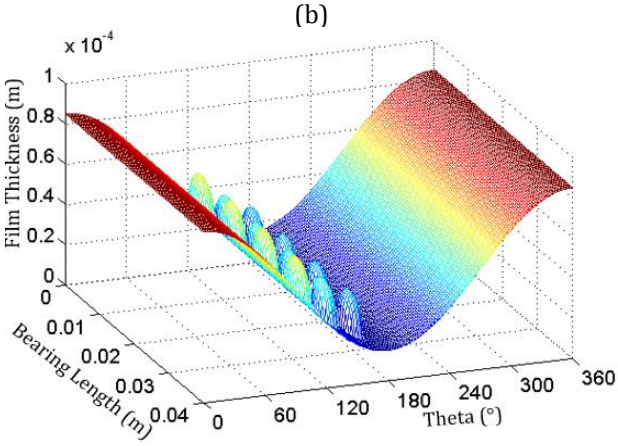
For each location, bearing characteristics have been calculated for non-porous conditions (Ψ =0) and high permeability conditions (Ψ =1). It has been observed that the positive percentage variation in load is noted even at high eccentricity ratio. And out of all considered locations, location 2 (172°-134°) is showing the maximum enhancement of 9.81 % approximately, and location 5 (172°-77°) shows minimum of about 5.9 % for non-porous bearing, Fig. 8. For the same locations, COF has the maximum decrement of 10 % and minimum of 7.24 %. These results are also in accordance with the work of Kango et al. [14].



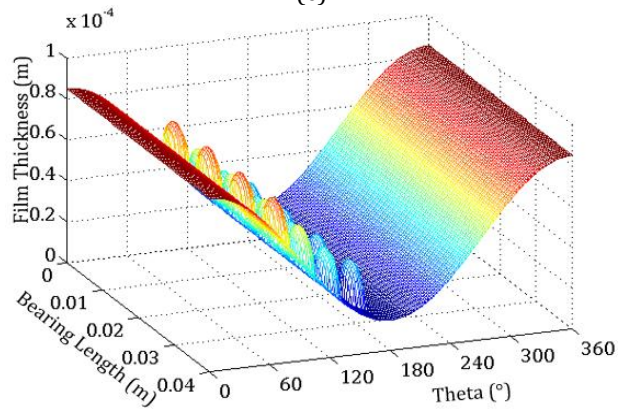
(a)



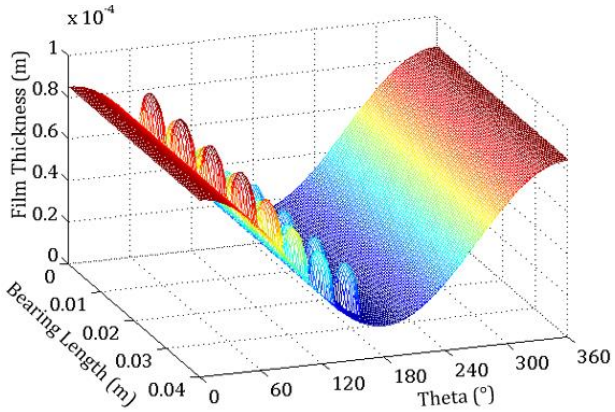
(b)



(c)



(d)



(e)

Fig. 8. Lubricant film thickness evolution for different texture locations: (a) 172°-153°, (b) 172°-134°, (c) 172°-115°, (d) 172°-96°, and (e) 172°-77°.

The textured porous bearing with high permeability parameter (Ψ=1) in the maximum pressure zone results in the deterioration of the bearing characteristics for all considered locations. For the above discussion it has been concluded that the textured porous bearing at high eccentricity ratio and at high permeability parameter does not show fruitful results even at best locations of textures. The reason behind the deterioration of the performance characteristics for porous bearing at maximum pressure region is the basic nature of these bearings. Porous bearings are suitable for low load and high-speed applications; therefore, high permeability parameter further leads to lowest load values. It has been observed from above discussion that the location of surface textures (maximum pressure region) is fruitful for non-porous journal bearings at high eccentricity ratio.

4. CONCLUSIONS

In the present study, the influence of spherical textures is investigated with variation in permeability parameter for the evaluation of performance of porous journal bearing.

The following conclusions have been made based on the computed results:

- Larger values of permeability parameter deteriorate the static performance characteristics of smooth porous journal bearings.
- Spherical surface textures at convergent zone significantly help in enhancing the performance characteristics of porous journal bearing at high eccentricity ratio.
- Spherical surface textures at maximum pressure region are only fruitful for non-porous journal bearing at high eccentricity ratio.

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