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CFD Analysis of Static Characteristics of Hydrodynamic Journal Bearing With Textured Bearing Surface

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Abstract— ***Performance of a Journal bearing depends on two aspects. First one is the improvement of inherent properties of lubricant by rigorous research. Second one is the improvement of bearing performance by doing some research on the design aspects of bearing surface. Improvement of bearing performance does mean that its load carrying capacity will be increased and at the same time the frictional loss due to the shearing phenomenon between bearing surface and the lubricant layer should be decreased.***

It has been proved that if artificial roughness can be created on the bearing surface perfectly it can increase the performance of a journal bearing. Now it is the matter of research that which shape, dimensions, position and orientation will increase the performance of a given journal bearing most. In the present work an attempt has been made to study the influence of a specific type of artificial roughness on the performance of a journal bearing.

Keywords— Journal bearing, textured surface, friction, cavitation, load carrying capacity.

I. INTRODUCTION

Working principle of a hydrodynamic sliding contact bearing like journal bearing depends on the formation of wedge shaped lubricant film thickness and pressure distribution in this oil film. Depending upon the pressure distribution in the oil film load carrying capacity of a journal bearing is determined. An efficient pressure distribution is possible if the inherent properties of a lubricant can be improved. But it has been noticed that only improving lubricant properties performance of a journal bearing cannot be improved much.

Many scientists investigated the alternate way to improve bearing performance by doing research on the design aspects of the journal bearing. Floberg[6] investigated the influence of cavitation in journal bearings. Cavitation has a strong influence on the stability of a journal bearing. Rao and Swaick[10] did the research on the stability considering stability as the prime aspect.

Etison[13] made a beautiful attempt to increase the load carrying capacity of a journal bearing.

In his paper [13] he showed that bearing surface texturing has a positive influence on the bearing performance. He did remarkable works on laser texturing on bearing surface.

Rao and Vencel[11] did a research on about different tribological and design parameters of a lubricated sliding bearing in detail.

From the work of Fredric Sahlin et all[13] we come to know how texture of different shapes influence tribological and design parameters of the two mating surfaces. They used CFD for their research work.

In 2007 a research paper was published by Cuppilard et all [2] in the journal named ‘Proceedings of Institution of Mechanical Engineers’ where findings of Shalin et all [13] were used along with the cavitaion model of Floberg [6] on the journal bearing. Cupplilard et all [2] used CFD for the simulation of the textured journal bearing.

Later S. Mishra et al [19] extended the work of Cupillard et al [2] to implement reverse dimple on the bearing surface and studied the influence on the bearing performance.

Though from the work of S. Mishra et al [19] it is quite evident that reverse dimple can further improve the bearing performance but these are quite hard to manufacture.

So in the present work few wedge shaped textures which are easy to be manufactured have been introduced on the earing surface to study their influence on bearing performance.

II. NUMERICAL MODEL

A. Equations

In the present work the problem has been investigated numerically using a CFD software named Fluent. The Computational Fluid Dynamics is based on the theory of Navier-Stokes which can be expressed as following.

$$\frac{\partial}{\partial x_i}(\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right] \qquad (1)$$

$$\frac{\partial}{\partial x_i}(\rho u_i) = 0 \qquad (2)$$

B. Cavitation Model

When there is a mixed flow in any flow domain then cavitation model has to be followed if the flow is being analysed by a CFD software like fluent. Here in the present work Rayleigh–Plesset[17] cavitation model has been used which has been described and tested successfully in reference [19]. In the present work mix flow or multi-phase flow is occurred as lubricant vapour is produced when pressure falls under the saturation pressure psat. The quantity of vapour bubbles is governed by the equations of growth and collapse of bubbles described by Plesset [20]. This multi-phase model is chosen to be homogeneous, i.e. all fluids share the same velocity and pressure field.

C. Geometries Used and Parameters Studied

In the present simulation a two dimensional flow domain has been used. Use of two-dimensional geometry accelerates computational time. Reference [2] has been referred for the lubricant properties. Dimensions of the flooded bearing have been taken from reference [2] and the dimensions are as follows

Length l = 0.133 m,

Shaft radius Rs = 0.05 m,

Radial clearance c = 0.145 mm,

Eccentricity ratio 1 = 0.61,

Angular velocity v = 48.1 rad/s.

Properties of lubricant are as follows.

Density is 840 kg/m3 and a dynamic viscosity is 0.0127 Pa s.

Here no-slip boundary condition has been assumed at the bearing walls. Fluid layer attached to the shaft surface moves with same velocity of the shaft. Fluid layer attached to the bearing surface is static.

Cupillard considered a series of ten dimples on the surface of a two-dimensional bearing in his work.

The distance between dimples does not exceed 10 per cent of their width. A two-dimensional texture model can be seen in Fig. 1(a).

Cupillard et al showed in their work [2] that how position of dimple cluster influence the performance parameter of a journal bearing by changing the dimple position keeping other parameters, that is, the number of bearings, start angle of placement of bearing, the ratio of dimple width to minimum thickness of fluid film, angle of span and inter dimple angle unchanged.

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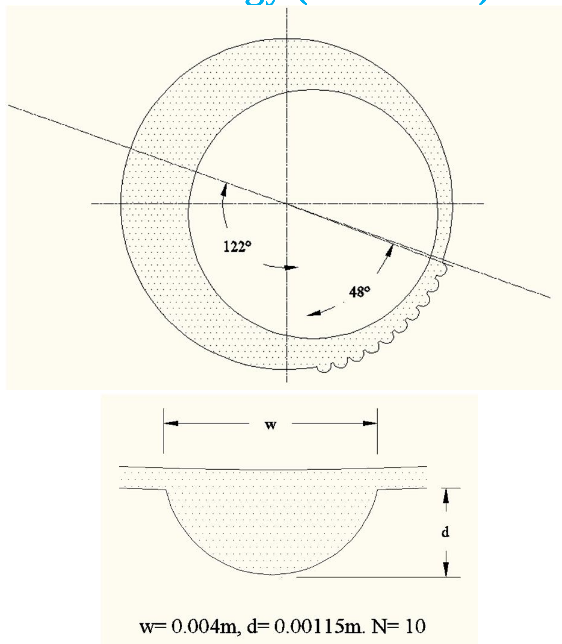

Fig. 1(a)

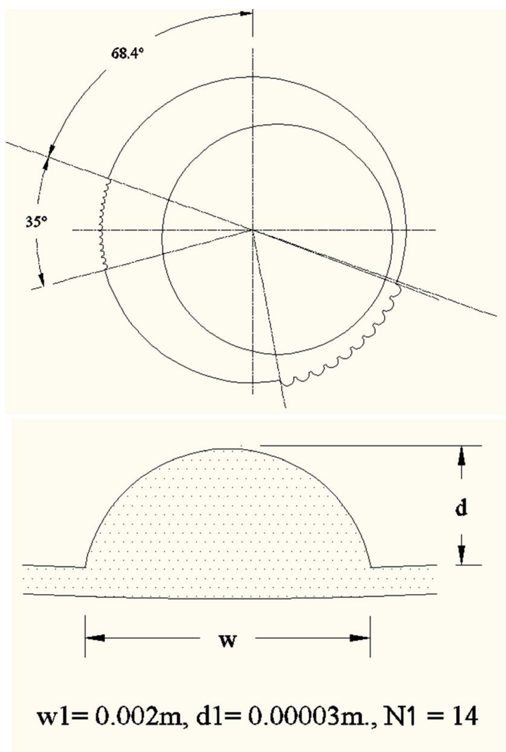

Fig. 1(b)

Fig. 1 Two-dimensional model: (a) textured bearing model and (b) dimple geometry

Dimples have been generated with parametric dimensions of its width and depth. The ratio of width (d) and minimum thickness of the lubricant film (h_{min}), which is a dimensionless number, has been considered as the reference. For the above bearing this $\frac{d}{h_{min}}$ is greater than unity

Load carrying capacity (W), the friction force (Fr), and the friction coefficient (f) have been calculated for a fixed eccentricity of the bearing. The load carrying capacity is calculated from the integration of the pressure acting on the shaft

$$W = L\sqrt{\left(\int_0^{2\pi} p\cos\theta\; R\,d\theta\right)^2 + \left(\int_0^{2\pi} p\sin\theta\; R\,d\theta\right)^2} \quad -(3)$$

The friction force is calculated from the following equation-

$$F_r = L\int_0^{2\pi} \tau R^2\, d\theta \quad -(4)$$

The friction coefficient is the ratio of friction force and load carrying capacity. It is expressed as-

$$f = \frac{F_r}{W} \quad -(5)$$

Performance of any bearing is judged by calculating the friction coefficient and comparing the value among the considered bearings.

D. Validation Of The Mathematical Model

In this paper, work of Cupillard et al [2] has been reproduced. In his work Cupillard et al [2] considered cylindrical dimple on the bearing surface and this topology has been used tested withANSYS CFX software but the his work here has been reproduced using Fluent software.

In this validation process all the settings regarding topological property of flow volume, physical properties of the flowing fluid and numerical properties of simulation have been verified.

Cupillard considered three types of dimple configurations in his work. Two of them have been verified with Fluent 6.3.26. These two configurations are dimples started at 57° and 122°. These two configurations have been shown in fig.2.

Here other parameters like the number of bearings, start angle of placement of bearing, the ratio of dimple width to minimum thickness of fluid film, angle of span and inter dimple angle have been kept unchanged in accordance with reference [2]

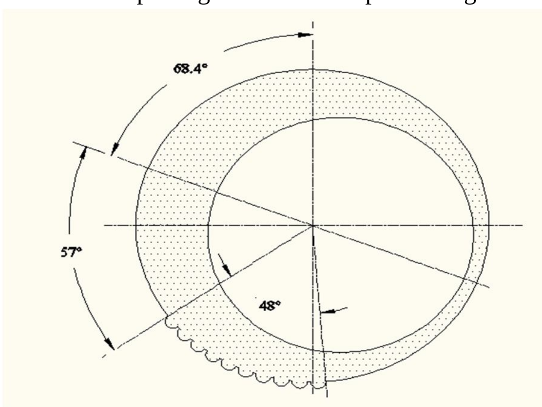

Fig 2(a)

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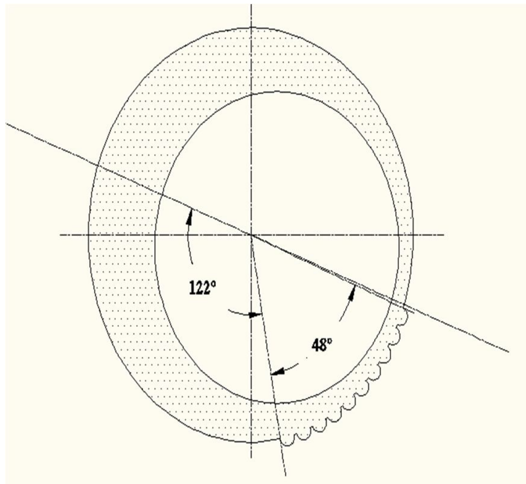

Fig2(b)

Fig. 2 Two-dimensional model: (a) dimple starting angle 57° (b) dimple starting angle 122°.

The results those have been obtained from the CFD analysis have been mentioned below.

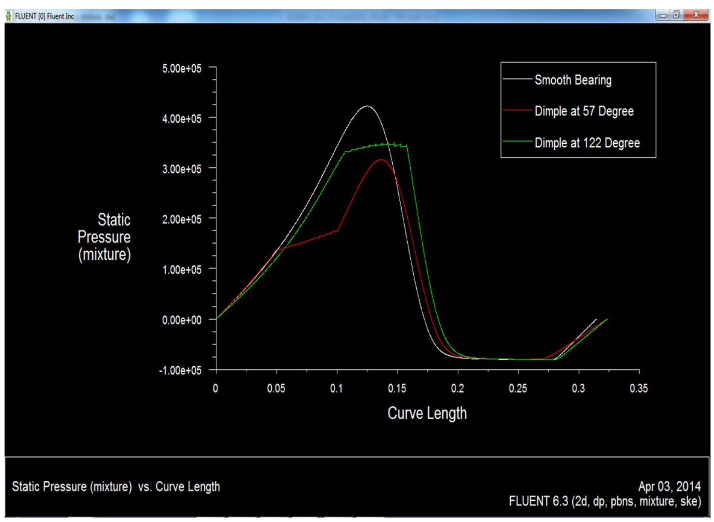

Fig. 3 Pressure distribution of Smooth bearing and bearings with dimple starting angle 57° and 122°.

Data regarding pressure at every node have been stored in form of a text file and then have been used in a MATLAB program for numerical integration to calculate carrying capacity of the bearing and frictional force between bearing surface and fluid film. . After calculating the load carrying capacity and friction force we have calculated the friction coefficient and examined with the result calculated by Cupillard et all [2].

The above mentioned comparisons have been mention in the table below.

TABLE I: LOAD CAPACITY AND FRICTION COEFFICIENT

Sl no.	Bearing Detail	Load carrying capacity (W) in newton	Friction Force (Fr) in Newton	Coefficient of friction	%age change
1	Smooth bearing	4664.100	13.7205	0.002941725	
2	start angle 57°	3581.287	12.47	0.003481988	18.3655287
3	start angle 122°	4527.330	12.275	0.002711311	-7.8326153

The above results are completely in complying with the reference [2].

E. Validation of Few Previous Modifications

In the year of 2014 a work on texturing of journal bearing surface was done by S. Mishra et al [21]. In their modification a set of another reversed dimples were introduced. The position of this dimple array and dimensions of each reverse dimples have been shown in figure below.

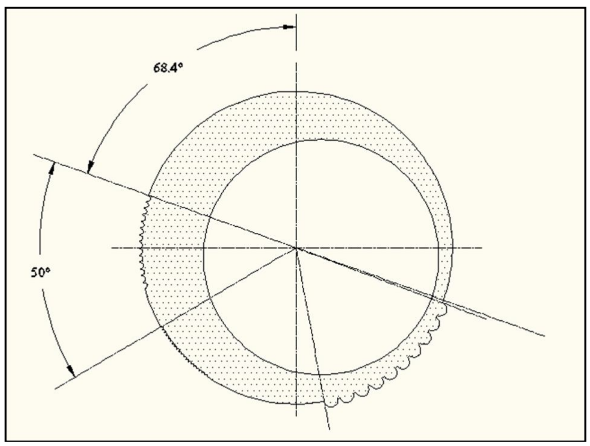

Fig: Overall position of all the dimples in Modification incurred by S. Mishra et al [21]

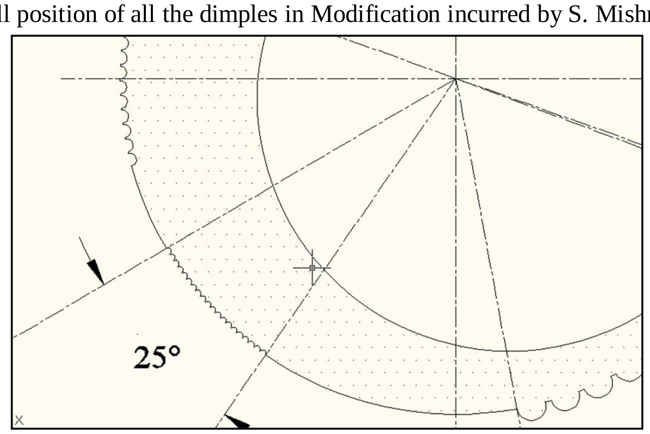

Fig: Enlarged view to show the second set of reverse dimples in Modification incurred by S. Mishra et al [21]

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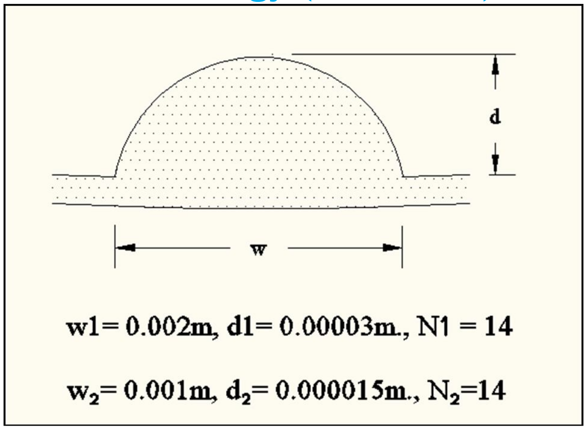
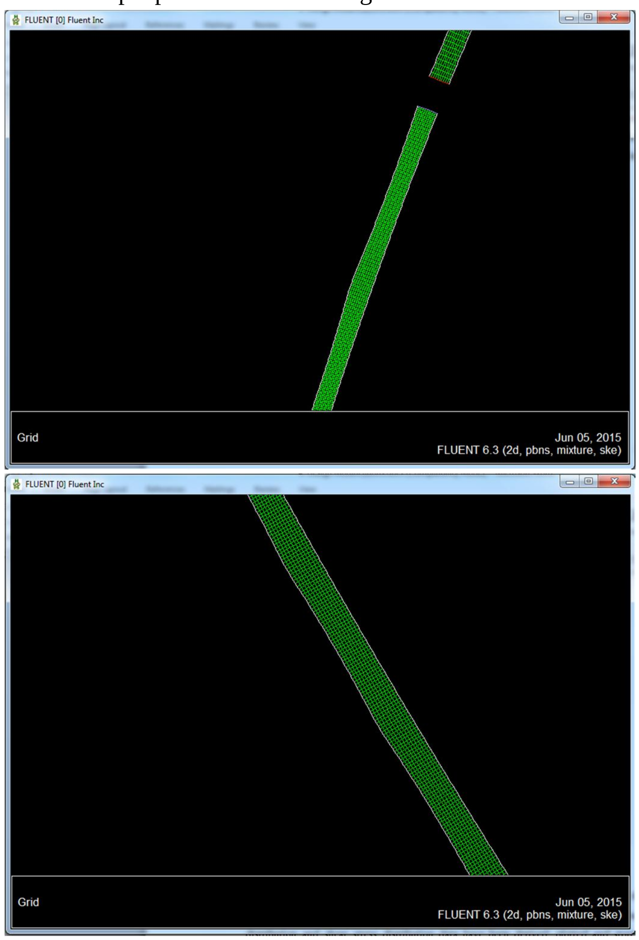


Fig: Geometrical dimensions of reverse dimples introduced in Modification incurred by S. Mishra et al [21]

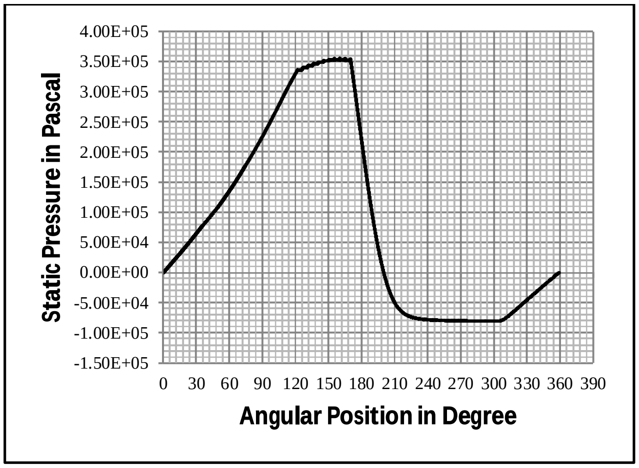
Below are the figures depicting meshed dimple portion of the bearing.



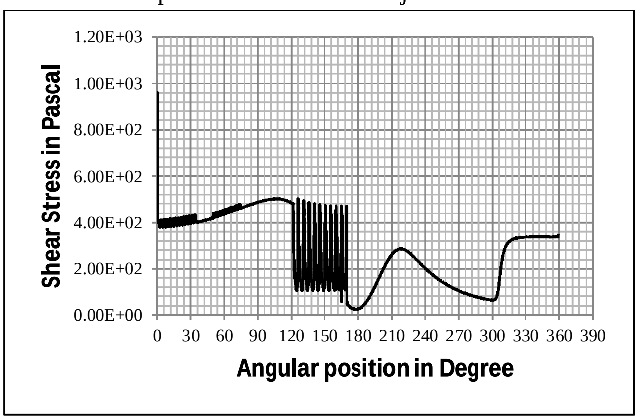
Enlarged view of meshing of two types of reverse dimples in modification incurred by S. Mishra et al [21]

On completion of simulation that is after getting the convergence, pressure distribution and shear stress distribution data have been derived, plotted and stored. Below are the graphs showing variation of pressure and shear stress at different nodal position on the journal surface.

With help of this stored data 'Load Carrying Capacity' and 'Friction Force' have calculated by integrating as per equation 1.14 and 1.15 in Matlab. The 'Load Carrying Capacity' has been calculated as 4563.492 N and the 'Friction Force' has been calculated as 12.315 N. To find out the 'Friction Coefficient' ratio of 'Friction Force' and 'Load Carrying Capacity' has been calculated it has been found out as 0.002698591.



Plot of pressure distribution on journal surface.



Plot of shear stress distribution on journal surface

Though the result achieved in this modification is same as result achieved by S. Mishra et al [21] but the configuration is not feasible to be achieved due to difficulties in manufacturing. So few more modifications have been tried.

F. DESIGN MODIFICATION ADOPTED

In the present work it has been investigated that whether any other shape of dimple increase the performance of the journal bearing more than the bearing dimple designed by Cupillard et al [2]. To do this, wedge shaped dimple has been introduced in place of spherical shaped dimple at the most efficient place on the bearing surface as found out by Cupillard et al in their work mentioned in reference [2]. Below is the topological details of the wedge shaped dimples.

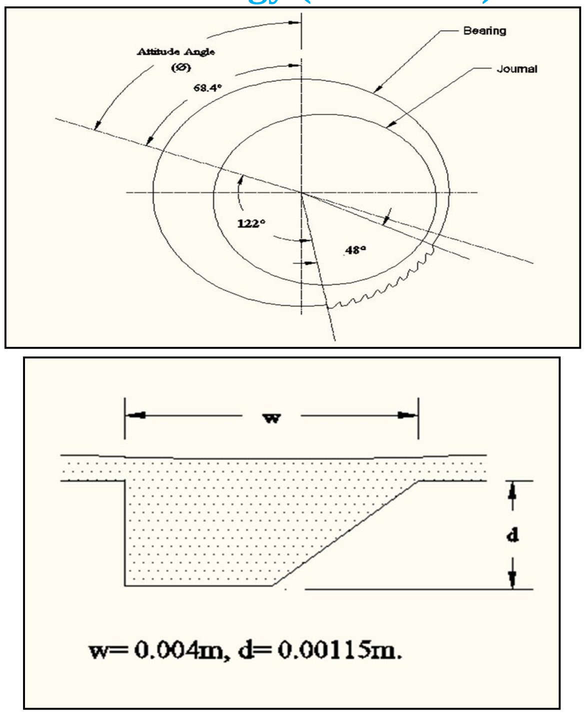


Fig. 4: Topological details of the wedge shape dimples

The geometry of the flow region has been created in GAMBIT software and meshing along with face zone and cell zone definition have been completed in the same software. After doing the mentioned job in GAMBIT the file has been saved as a '.msh' file to import in Fluent. In Fluent software the .msh file is read first to get the meshed flow region. This has been shown below.

The geometry of the flow region has been created in GAMBIT software and meshing along with face zone and cell zone definition have been completed in the same software. After doing the mentioned job in GAMBIT the file has been saved as a '.msh' file to import in Fluent. In Fluent software the .msh file is read first to get the meshed flow region. This has been shown below.

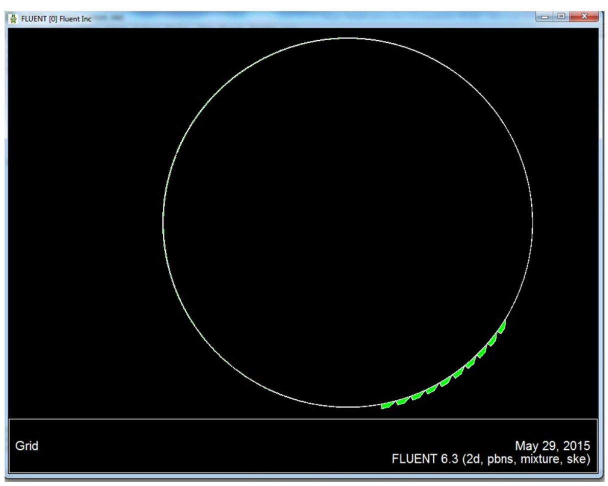


Fig 5: Full view of meshed journal bearing with wedge shaped bearing

In the figure below an enlarged view of meshed dimple has been shown. It is very clear from the figure that two types of element have been used to mesh the dimple region of the flow area. These elements are quadrilateral and triangular elements. To mesh the dimple regions first the regions have been divided in two parts in such a way that one part is a rectangular area and other part is a triangular area. In rectangular area quadrilateral elements have been imposed and in triangular area triangular elements have been imposed. This type of planed meshing improves the result of simulation and give better output.

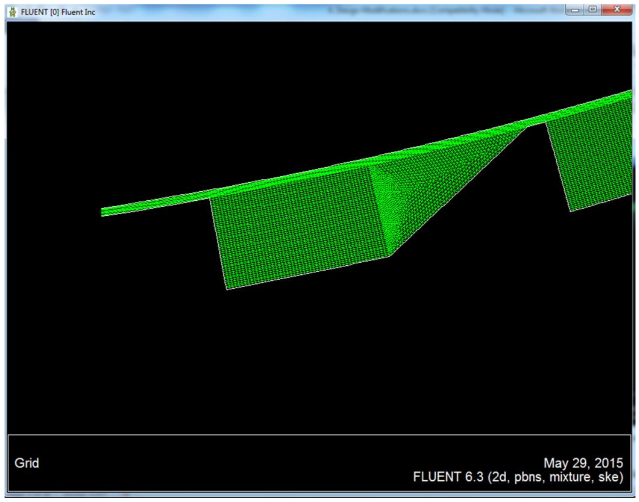


Fig 6: Enlarged view to show meshing strategy on a single wedge shape dimple region

Different physical and mathematical parameters those have been used for the simulation of flow inside a journal bearing with wedge shape dimples have been kept same as those used for the simulation of journal bearing with cylindrical dimples as mentioned in above sections. The parameters which have been set for CFD simulation have been mentioned below:

Face zone type of all the boundaries of the 'Flow Region'.

Properties of fluid.

Turbulence parameters.

Parameters to define physical state of fluid.

Parameters to define cell zone conditions.

Parameters to define different boundary conditions.

Convergence parameters.

Details about each above mentioned parameters have been discussed previously. After setting of all the above mentioned parameters the problem has been simulated and the following outputs have been derived after convergence is arrived. The outputs are-

Contour plotting of pressure distribution.

Graph plotting of pressure distribution versus linear length along the periphery of bearing where length measured from attitude angle line.

Graph plotting frictional shear stress of journal surface and fluid film versus peripheral length of journal starting from attitude angle line.

Writing of pressure data and shear-stress data at all the nodal position on bearing surface in a text file.

Below is the figure of a contour plotting depicting the pressure distribution at the region of inlet-outlet.

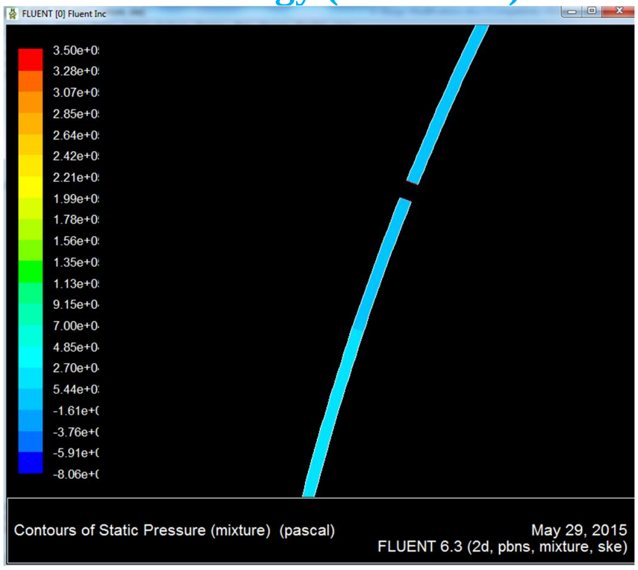


Fig 7: Pressure distribution at maximum film thickness region

Figure above shows that at the inlet-outlet region pressure is within atmospheric level. But as angular distances increases starting from the attitude line along the bearing periphery, pressure is increased. A figure has been shown below to depict pressure distribution at the dimple region.

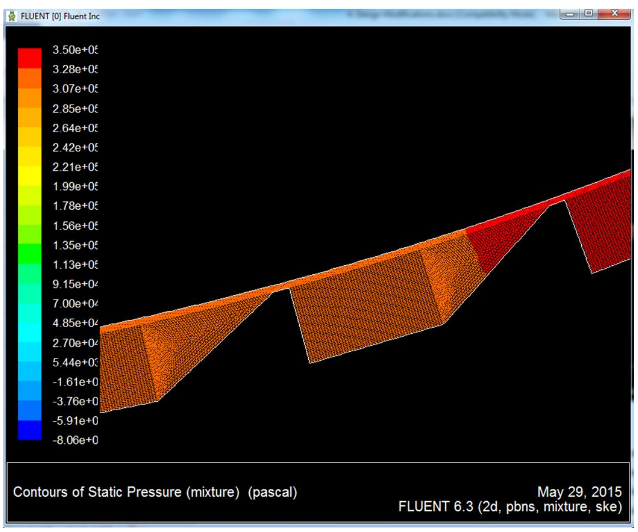


Fig 8: Pressure distribution at dimple region

With increase in angular distance along the periphery cavitation condition arrives beyond 180°. In cavitation region pressure is dropped below atmospheric pressure and when pressure reaches up to vapor pressure level of the lubricant the pressure variation stopped. A figure has been shown below to present the contour plotting of pressure variation at cavitation region.
In figure 3.25 graph plotting has been presented to show the variation of pressure with respect to the angular position of nodes on bearing periphery measuring from the attitude angle line.

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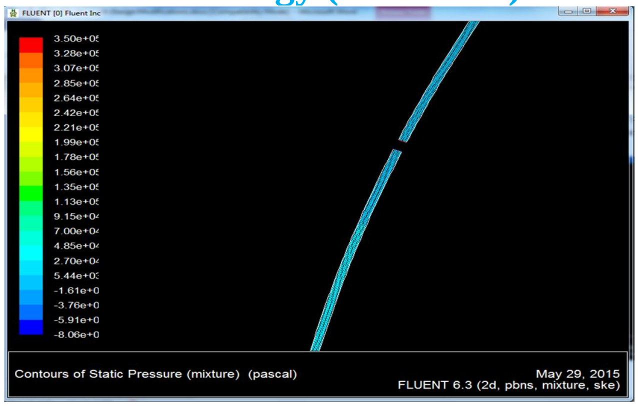

Fig 9: Pressure distribution at cavitation region

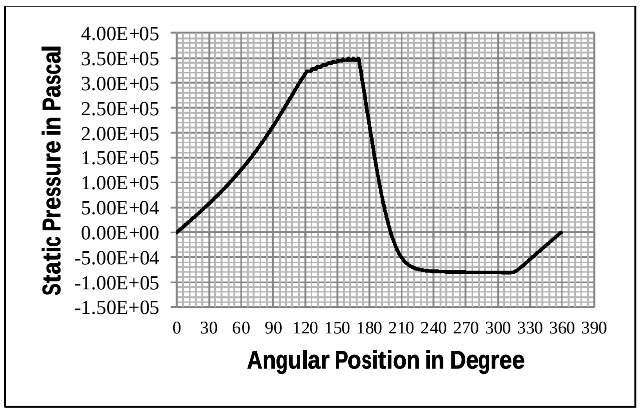

Fig 11: Plot of pressure distribution on journal surface

It is very much understood that if a shaft rotates on a lubricant film then there will be a friction in between rotating journal surface and the adjacent fluid film. This shear stress varies at point to point on the journal surface. A graph plotting has been shown below to depict the variation of shear stress at different nodes on journal surface measuring the angular distance from attitude line along the journal periphery.

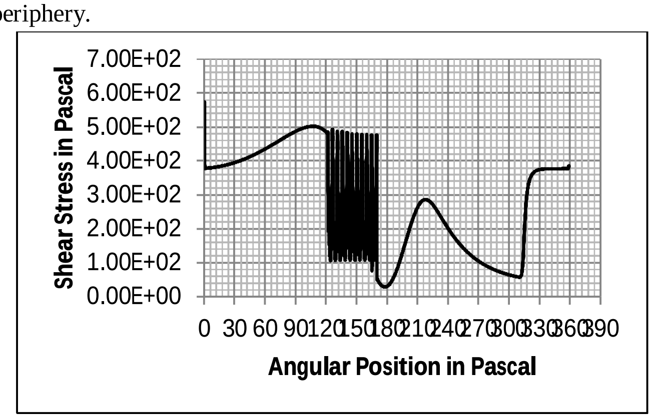

Fig 10: Plot of shear stress distribution on journal surface

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The data which have been generated from the simulation to predict the pressure distribution and shear-stress at different nodal point are stored in a text file for further work. Actually these data are having immense importance. With help of these data, 'Load Carrying Capacity' and 'Friction Force' are calculated with help of Matlab. As mentioned in chapter 1.0 'Load Carrying Capacity', 'Friction Force' and 'Friction Coefficient' are calculated from equation 1.14, 1.15 and 1.16 respectively, these mathematical operation have been done in Matlab on the stored pressure distribution data and shear stress data generated from CFD simulation in Fluent. After doing the integrations in Matlab with help of stored pressure distribution and shear stress distribution data following results have been gatten.

Load Carrying Capacity (W)= 4384.300 newton

Friction Force= 11.973 newton

Friction Coefficient= 0.002730881

III. RESULT AND DISCUSSION

All the modifications availed by different authors and also the idea of modification which have been incurred in the present work have been influenced the bearing performance in different way. All the results have been presented in this section in tabular form to compare al together.

Table 3.4 Comparisons of results of dimpled bearing as per reff. [2], Reff. [35] and journal bearing with modification incurred in the present work

Sl no.	Details of roughness	Load carrying capacity (W) in newton	Friction Force (Fr) in Newton	Coefficient of friction
1	Roughness parameter as per reference [2] with initial angle 122º	4527.330	12.275	0.002711311
2	Roughness parameter as per the modification adopted by S. Mishra et al. [19].	4563.492	12.315	0.002698591
3	Dimpled Journal Bearing with modification of the present work	4384.300	11.973	0.002730881

From the above table it is quite clear that modification adopted by S. Mishra et al [19] is most fruitful but very hard to manufacture so other comparatively simply manufacturable roughness may be incorporated and studied to see how performance parameter is getting influenced.

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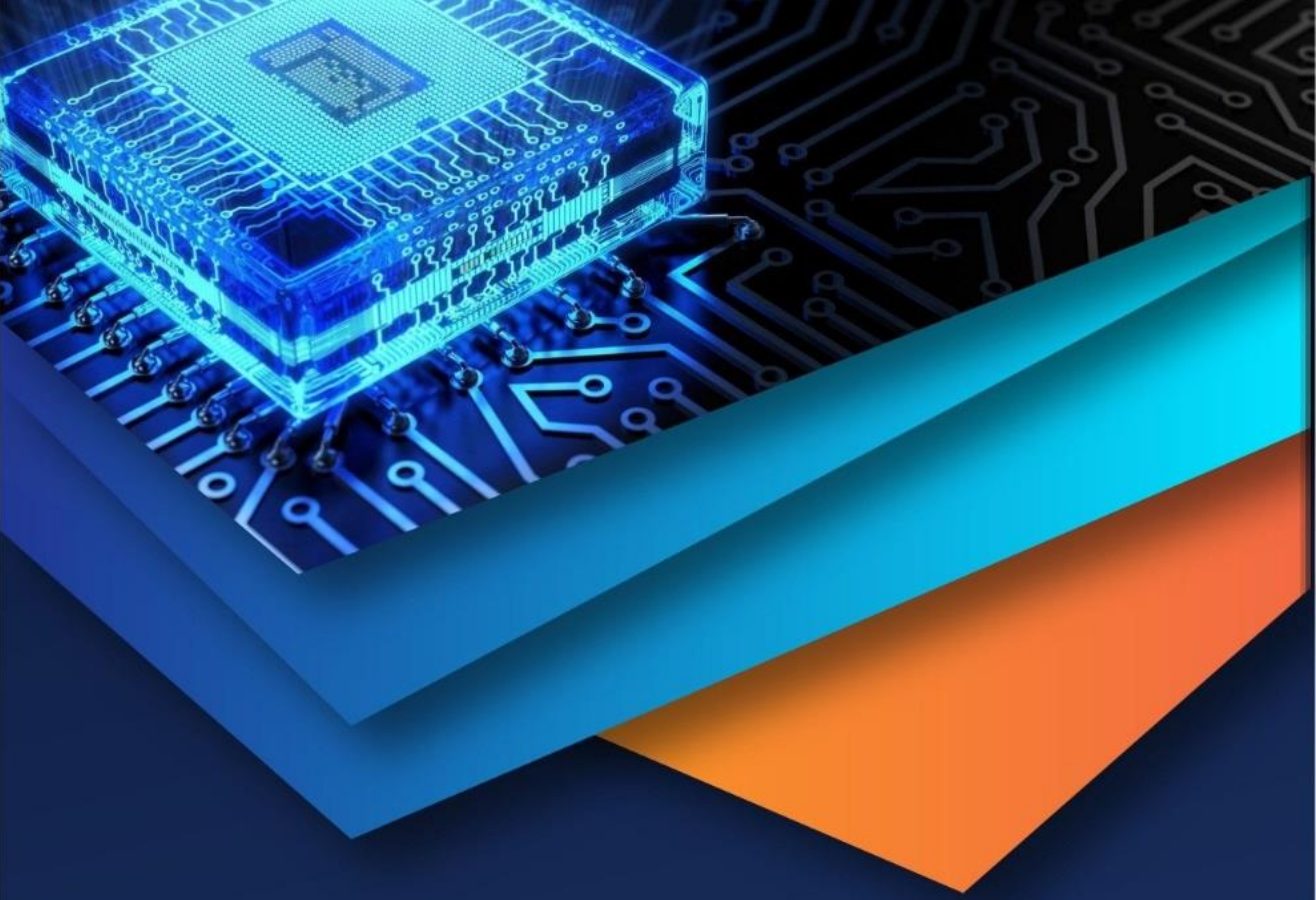

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45.98

IMPACT FACTOR: 7.129

TOGETHER WE REACH THE GOAL

IMPACT FACTOR: 7.429