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CFD Investigation of Influences of Reverse Textures on Bearing Surface of a Journal Bearing

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ABSTRACT

It has always been the subject of interest and research to find out most efficient method to minimize the frictional loss of any mating surface in any mechanism. But it has been identified that only improvement of lubricating material cannot help much. For this reason research on design modification was started and still going on to optimize the design of mating surfaces so that losses be minimum. In this paper a research has been done on the design of bearing surface of a journal bearing. Journal bearing is a very important part of many machines specially turbines used in Power generation and rolling mills used in steel making sectors. By modifying the design of the bearing surface, load carrying capacity of a journal bearing can be increased and thus its performance can be improved. It is proved by many researches that, by putting cylindrical dimples on bearing surface the performance of a journal bearing can be increased. To measure the performance of a journal bearing three parameters are usually used. Those are, (i) Load Carrying Capacity, (ii) Frictional resisting force and (iii) Ratio of Frictional resistance and Load carrying capacity, which is called Frictional coefficient. In this work a modification has been done on the work of Cupillard. Cupillard considered only the cylindrical dimples for the sake of manufacturing simplicity, but in this work a different configuration of dimple has been considered and its influence on bearing performance has been studied.

Keywords: Journal bearing, Textured surface, Friction, Cavitation, Load carrying capacity.

1. INTRODUCTION

Hydrodynamic Journal bearings rely on the formation of relatively thick film between the journal and the bearing. Despite of the significant advancement in the lubrication technology, these bearings do fail in practice with serious consequences, particularly in large installations such as power plants, rolling mills etc. Many scientists investigated the reasons for such failures and researches for the remedy of such failures are in. The research to improve the performance of journal bearing is also on. Floberg (1961) investigated the influence of cavitation in journal bearings. Cavitation has a strong influence on the stability of a journal bearing. Rao & Swaick (2002) did the research on the stability considering stability as the prime aspect.

A beautiful attempt to increase the load carrying capacity of a journal bearing was made by Etison (2004). Etison showed that texturing on a bearing surface can improve the performance of a journal bearing.

Etison has investigated the influence of laser surface texturing on a thrust bearing. Rao & Vencel (2005), has investigated different tribological and design parameters of a lubricated sliding bearing in detail. Fredric *et al* (2005) has done CFD investigation that how texture of different shapes influence tribological and design parameters of the two mating surfaces.

Cuppilard *et al* (2008) implemented findings of Fredric *et al* (2005) results on a journal bearing and also used Floberg (1961) cavitation model on the journal bearing. Cuppilard *et al* (2008) used CFD for the simulation of the textured journal bearing.
Etison's (2005) has worked on the bearing surface texture. Navthar *et al.* (2010) worked on stability analysis of journal bearing using stiffness coefficient.
Based on the above litreture, In this paper, extended the Cuppilard's work by putting a reverse dimple on the bearing surface texture and observed the bearing performance.

2. NUMERICAL MODEL

2.1 Equations

The code used for the simulations is Fluent 6.3.26. No Reynolds or Stokes assumptions are made in the equations. Dimensional momentum Eq. (1) coupled with the continuity Eq. (2) is solved over the domain using the finite-volume method.

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

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

2.2. Cavitation Model

The cavitation model used is the Rayleigh–Plesset model as described and tested successfully in Bakir *et al.* (2004). This is a multi-phase flow model in which lubricant vapour is produced when pressure falls under the saturation pressure p_{sat}. The equations of growth and collapse of air bubbles controlling the quantity of lubricant vapour produced in the cavitated zone have been introduced by Plesset (1949). This multi-phase model is chosen to be homogeneous, i.e. all fluids share the same velocity and pressure field.

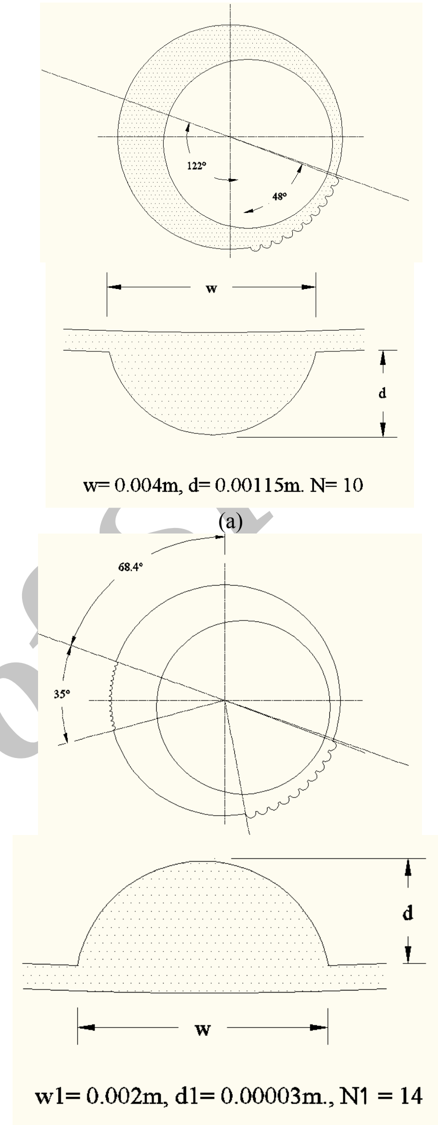
2.3. Geometries Used and Parameters Studied

Two-dimensional bearing geometries are used in the simulations. Two-dimensional geometry is used to accelerate computational time. The lubricant properties are also as given in reference [9]. The flooded bearing used in reference [9] had a length l = 0.133 m, shaft radius R_s = 0.05 m, radial clearance c = 0.145 mm, eccentricity ratio e = 0.61, and angular velocity ω= 48.1 rad/s. The lubricant used had a density of 840 kg/m^3 and a dynamic viscosity of 0.0127 Pa-s. At the walls, no-slip boundary conditions are assumed. At the shaft surface, the fluid moves with the same velocity as the shaft. At the bearing surface the fluid velocity is set to zero. A symmetry condition is applied on one side of the three-dimensional bearing in order to simulate half of the domain. On the other side of the three-dimensional bearing, the pressure is set at the ambient value, i.e. the relative pressure is set to zero. At this boundary, the volume fraction of the lubricant is set to one.

The surface texture adopted by Cuppillard *et al* (2008). Cupillard used a series of 10 dimple on the surface of a two-dimensional bearing, with geometry identical to the central cross-section of the three-dimensional smooth bearing described above. 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).

Cuppillard *et al* (2008) has shown the influence of cylindrical dimples which are created by metal removal and in this paper a set of reverse dimples have been added in form of metal addition which has been shown in Fig.1(b).

The shape of the dimples or the topology of the dimples has only been changed. The 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 have been kept same as the corresponding magnitude considered by Cuppilard *et al* (2008).



(b)

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

Dimples' dimensions have been parameterized by the width and depth of the dimples. A dimensionless parameter which is equal to the ratio of the width (d) and minimum thickness of the lubricant film (h_{min}), has been considered as the reference. For the above bearing this $\frac{d}{h_{min}}$ is greater than unity.

The following parameters are calculated for a fixed eccentricity: the load carrying capacity (W), the friction force (Fr), and the friction coefficient (f). 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-

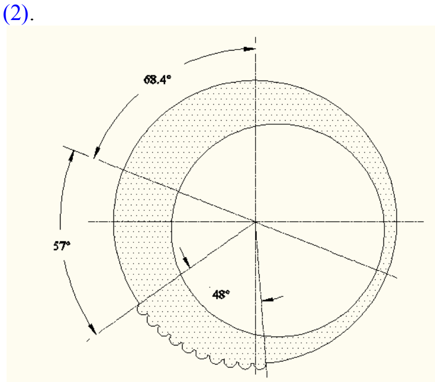
$$f = \frac{F_r}{W} \quad (5)$$

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

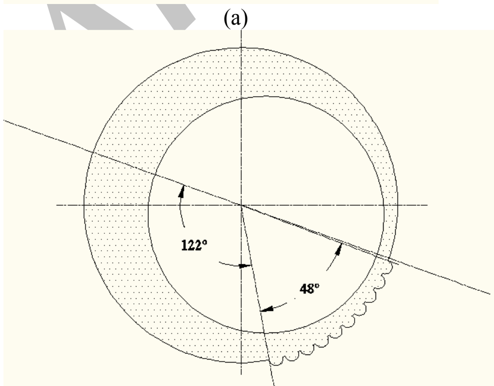
3. Validation Of The Matematical Model

Cuppilard *et al* (2008) studied the influence of dimples with help of CFD analysis and proposed that a series of cylindrical dimples can improve the performance of a bearing. Whatever mathematical model has been used to validate the outcome of Cuppilard *et al* (2008) implemented here to verify the Fluent 6.3.26. The result obtained has been presented below.

Cuppilard *et al* (2008) considered three types of dimple configurations. 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).



(a)



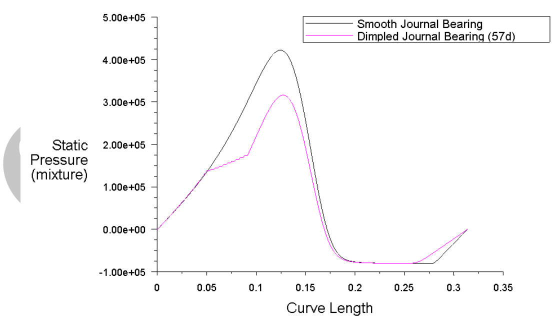
(b)

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

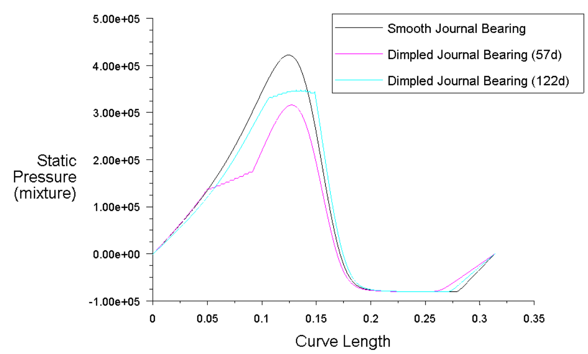
All the other dimensions of dimples have been remaining same, that is, the number of bearings, start angle of placement of bearing, the ratio of dimple width to minimum thickness of fluid film, angle of span and inter dimple angle have been kept same as the corresponding magnitude considered by Cuppilard *et al* (2008).

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

The pressure data which have been generated from the simulation with Fluent have been stored in a text file and later has been used for the integration with MATLAB to calculate the load carrying capacity and the friction force on the bearing surface due to skin friction. After calculating the load carrying capacity and friction force, calculate the friction coefficient and examined with the result calculated by Cuppilard *et al* (2008). The above mentioned comparisons have been mention in the table below.



(a)



(b)

Fig. 3. Comparison of pressure distribution: (a) Bearings with dimple starting angle 57° and 122° (b) Smooth bearing and bearings with dimple starting angle 57° and 122°.

The above results are completely in complying with the result obtained by Cuppilard *et al* (2008).

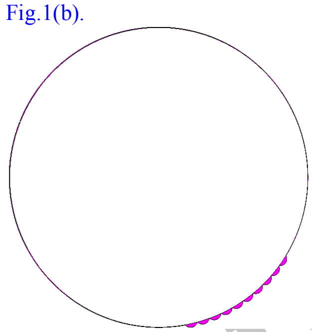
Table 1 Load Capacity and Friction Coefficient

SNo.	Bearing Detail	Load carrying capacity (W) in newton	Friction Force (F_r) in Newton	Coefficient of friction $F= F_r/W$	%age change
1	Smooth bearing	4664.10	13.7205	0.00294172	-
2	Start angle 57°	3581.28	12.47	0.00348198	18.3655 287
3	Start angle 122°	4527.33	12.275	0.00271131	- 7.83261 53

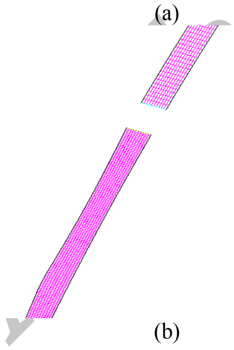
4. Design Modification Adopted

In the modification adopted here a set of reverse dimples has been introduced. The dimensional details have been shown in Fig.1(b). After CFD analysis of the above topology it has been calculated that load carrying capacity and friction force due to shear between fluid film and journal are 4537.315 N and 12.305 N. This combination gives the friction coefficient as **0.002711956** which is greater than the value of friction coefficient corresponding to the modification mentioned in Cuppilard *et al* (2008).

Dimensional representation of this dimple has been shown in Fig.1(b).



(a)

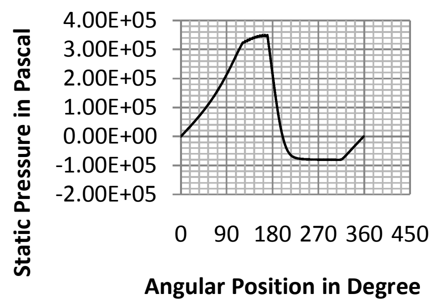


(b)

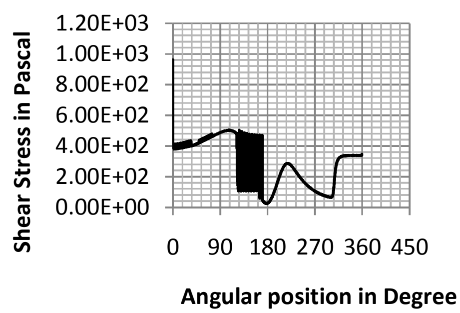
Fig.4. (a) and (b) meshed view of modified dimpled bearing in Fluent 6.3.26 have been shown.

After simulation with same mathematical model as used in reference [9] with help of Fluent 6.3.26 pressure distribution and shear-stress distribution with respect to angular position have been obtained and have been shown in Fig. 5(a) and Fig. 5(b) below.

The data from the above simulation have been stored in a text file which has been imported in MATLAB for the integration to calculate load carrying capacity and friction force. The result has been mentioned below.



(a)



(b)

Fig. 5. (a) Pressure distribution curve (b) Shear-stress distribution curve

Table 2 Comparison of Different Bearing Parameters Between Cuppilard and Proposed Modification.

SNo.	Parameters	Bearing with dimple configuration mentioned in (dimple start angle θ_i=122° and No of dimples= 10)	Bearing with modified dimples as mentioned above (dimple start angle θ_i = 122° and No. of dimples= 10)
1	Load carrying capacity (W) in newton.	4527.330	4537.315
2	Friction Force (F_r) in Newton	12.275	12.305
3	Coefficient of friction (f)	0.002711311	0.002711956

5. Result and Discussion

The result above shows that cylindrical shape of dimples in form of material removal help a journal bearing to give better performance than the reverse dimples in form of material addition. A future scope is there to investigate the influence of other topology of dimples on its performance. Effort may be given to generalize the mathematical model to identify the dimensions; number and position of dimples with respect to any given smooth journal bearing.

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