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Review: Effect of Spherical Texture Profile on the Performances of A Hydrodynamic Thrust Bearing.

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Abstract- Surface texture in the surface of the thrust bearing reduces the friction coefficient and wear rate of the matting surface. Significant improvement in load carrying capacity, wear resistance and friction coefficient of the mechanical component can be achieved by forming micro structure in form of micro dimple on the thrust pad bearing surface. In the present work, review of numerical analysis is carried out to investigate the effect of spherical, cylindrical texture on the thrust pad hydrodynamic bearing. These textured pads are tested on the various type of operating condition. Numerical study based on finite difference methods is used to study the behavior of coefficient of friction and load carrying capacity on variation of aspect ratio, texture height ratio etc.
Keywords- Hydrodynamic thrust bearing, texture surface, micro cavities, aspect ratio

I. INTRODUCTION

The load carrying capacity of journal and thrust bearings depends on various factors; among them wedging action add significantly for pressure development. In thrust bearing geometry modification is required for wedging action. In current work an attempt is given to find various frictional parameters by altering geometry of thrust bearing surface. With the help of sophisticated technology positive and negative microstructures are possible on thrust bearing face seal, surfaces etc. The theoretical study is carried out in hydrodynamic thrust bearing with spherical texture profile to uncover the importance of tribiological parameters such as load carrying capacity, co-efficient of friction, for different texture height ratio and aspect ratio. [1, 2]

II. SURFACE TEXTURES

By definition, a surface is an interface, a marked discontinuity from one material to another. The texture of any surface is defined by the intrinsic surface topography it exhibits. All surfaces have only one kind of texture and structure. Textured surfaces construct a lubrication film, which turn out a load carrying capacity when there is no condition for the wedge effect. Surface texturing has proved to be very efficient in full and mixed lubrication, reducing wear rate, friction coefficient, increase load capacity, damping coefficient and dynamic stiffness. [2]

2.1 Positive Asperities

These are micron scaled surface characteristic of any random geometry that is in the form of projection on a surface. The projection like bumps, posts are called positive asperities. Figure 1 and figure 2 shows the positive asperities. [2]

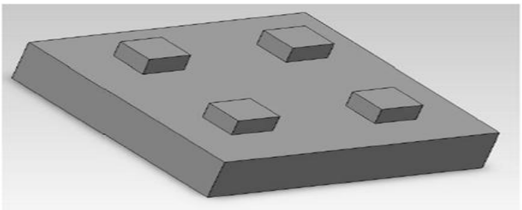

Fig. 1 Square Positive Asperity [2]

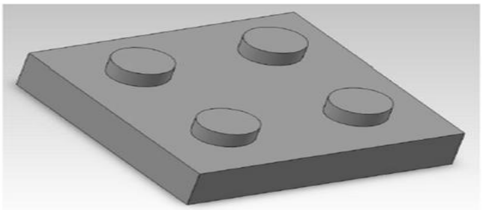

Fig. 2 Cylindrical Positive Asperity [2]

III. THEORETICAL AND NUMERICAL ANALYSIS

The lubrication theory is based up on the fussy form of Navier-Stokes equations. The prime Reynolds equation is a differential form equation which is fusion of continuity equation and Navier-stokes containing pressure which is frequently used in the hydrodynamic lubrication theory under some assumptions. From the Reynolds equation, the equation of state and the energy equation, accurate hydrodynamics fluid film can be obtained. [2]

3.1 Governing Equation

The momentum equation and continuity equation are adequate to describe lubrication flows for Newtonian fluids; combination of two is called Reynolds equation. The generalized Navier-Stroke equation in cylindrical coordinate system i.e. r, θ, z for thrust bearing is given by equation 3.1.1, 3.1.2 and 3.1.3

$$\rho\left(\frac{\partial U_r}{\partial t}+U_r\frac{\partial U_r}{\partial r}+\frac{U_\theta}{r}\frac{\partial U_r}{\partial \theta}-\frac{U_\theta^2}{r}+U_z\frac{\partial U_r}{\partial z}\right)=-\frac{\partial p}{\partial r}+\eta\left[\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial U_r}{\partial r}\right)+\frac{1}{r^2}\frac{\partial^2 U_r}{\partial \theta^2}+\frac{\partial^2 U_r}{\partial Z^2}-\frac{U_r}{r^2}-\frac{2}{r^2}\frac{\partial U_\theta}{\partial \theta}\right]+\rho g_r \quad \ldots (3.1.1)$$

$$\rho\left(\frac{\partial U_\theta}{\partial t}+U_r\frac{\partial U_\theta}{\partial r}+\frac{U_\theta}{r}\frac{\partial U_\theta}{\partial \theta}+\frac{U_r U_\theta}{r}+U_z\frac{\partial U_\theta}{\partial z}\right)=-\frac{1}{r}\frac{\partial p}{\partial r}+\eta\left[\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial U_\theta}{\partial \theta}\right)+\frac{1}{r^2}\frac{\partial^2 U_\theta}{\partial \theta^2}+\frac{\partial^2 U_\theta}{\partial z^2}-\frac{U_\theta}{r^2}+\frac{2}{r^2}\frac{\partial U_r}{\partial \theta}\right]+\rho g_\theta \quad \ldots (3.1.2)$$

$$\rho\left(\frac{\partial U_z}{\partial t}+U_r\frac{\partial U_z}{\partial r}+\frac{U_\theta}{r}\frac{\partial U_z}{\partial \theta}+U_z\frac{\partial U_z}{\partial z}\right)=-\frac{\partial p}{\partial z}+\eta\left[\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial U_z}{\partial r}\right)+\frac{1}{r^2}\frac{\partial^2 U_z}{\partial \theta^2}+\frac{\partial^2 U_z}{\partial z^2}\right]+\rho g_z \quad \ldots (3.1.3)$$

The continuity equation in the polar co-ordinate(r,θ,z) is given in equation 3.1.4.

$$\frac{\partial \rho}{\partial t}+\frac{1}{r}\frac{\partial}{\partial r}\left(\rho r U_r\right)+\frac{1}{r}\frac{\partial\left(\rho U_\theta\right)}{\partial \theta}+\frac{\partial\left(\rho u_z\right)}{\partial z}=0 \quad \ldots (3.1.4)$$

Combining Navier-Stoke equations (equations 3.1.1, 3.1.2 and 3.1.3) and continuity equation (equation 3.1.4), the generalized form of Reynolds equations can be found as equation 3.1.5

$$\frac{\partial}{\partial r}\left(rh^3\frac{\partial p}{\partial r}\right)+\frac{1}{r}\frac{\partial}{\partial \theta}\left(h^3\frac{\partial p}{\partial \theta}\right)=6\eta U\frac{\partial h}{\partial \theta} \quad \ldots 3.1.5$$

Where ,
h=c-cg
h= above the texture height
Cg= Height of the texture and
C= total clearance

To find the non-dimensional form of Reynolds equation, h, r and p terms are non-dimesionalized as

$\bar{h}=\dfrac{h}{c}$, $\bar{r}=\dfrac{r}{r_i}$ and $\bar{p}=\dfrac{pc^2}{\eta u r_i}$

in equation number 3.1.5
Rearranging equation

$$3\bar{r}\bar{h}^2\frac{\partial\bar{h}}{\partial\bar{r}}\frac{\partial\bar{p}}{\partial\bar{r}}+\bar{r}\bar{h}^3\frac{\partial^2\bar{p}}{\partial r^2}+\frac{3}{\bar{r}}\bar{h}^2\frac{\partial\bar{h}}{\partial\theta}\frac{\partial\bar{p}}{\partial\theta}+\frac{\bar{h}^3}{r}\frac{\partial^2\bar{p}}{\partial\theta}=6\frac{\partial\bar{h}}{\partial\theta} \quad \ldots 3.1.6$$

Now equation 3.1.6 gives variation of non-dimensional pressure profile in r and θ direction, which depends of variation of parameters like non-dimensional film thickness ratio, non dimensional radius in r and θ direction. In Current study the textured surface selected for analysis have above parameters to produce positive pressure profile. [1, 3, 4]

3.2 Numerical Solution

There are many ways to solve non-dimensional Reynolds equation like Finite Volume Methods, Spectral Methods and Finite Element Method, etc. Amongst all Finite Difference Method (FDM) is abstractly uncomplicated and simple to put into practice on a computer for regular shapes and fast iterative methods for solving this set of equations. So in the review of current analysis Finite Difference Method is used to solve Reynolds equation.

3.2.1 Finite Difference Method

The pressure distribution was computed using the finite difference method. Then the system of numerical equations was solved iteratively using the Gauss-Seidel method. The cavitations pressure was kept nil. This cavitations condition was integrated in the Gauss-Seidel iterative method so the Reynolds condition results by numerical diffusion. Central difference method is used in the present analysis, because it produces better estimates than forward or backward differences. All the differentials in the non-dimesionalized modified Reynolds equation are to be replaced by finite difference approximations. [3, 4]

IV. RESULTS AND DISCUSSION FOR CYLINDRICAL TEXTURE

Graphical and theoretical analysis is carried out to find out the performance characteristics of the hydrodynamic thrust pad bearing .In the present analysis effect of texture height ratio, aspect ratio on coefficient of friction parameter and non dimensional load. Positive cylindrical asperities type modeled and analyzed for the performance parameter.

4.1.1 Positive Asperities

These are micron scaled surface featured with spherical protrusion on a surface. Model of a single positive spherical asperity is shown in figure 3.

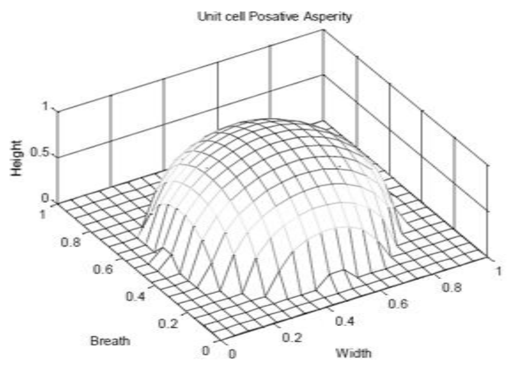

Fig. 3 Single Positive Spherical asperity

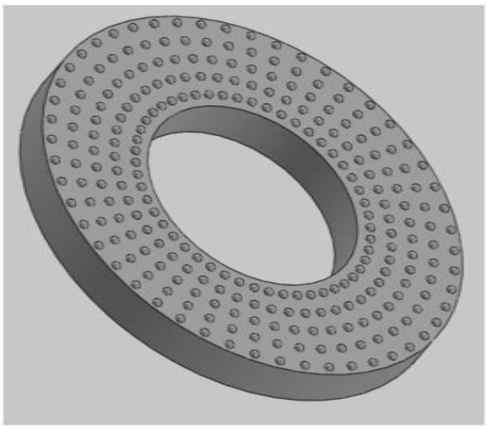

Fig.4 Distribution of positive spherical lasperities

The distribution of engineered spherical textures over complete thrust bearing surface is shown in figure 4

(a)Effect of Aspect Ratio on Non-Dimensional Load and Coefficient of friction Parameter

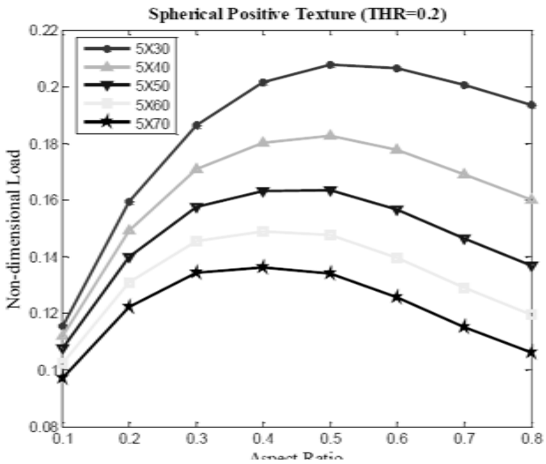

Fig. 5: Non- dimensional load Vs. Aspect ratio for different distribution of positive spherical texture.

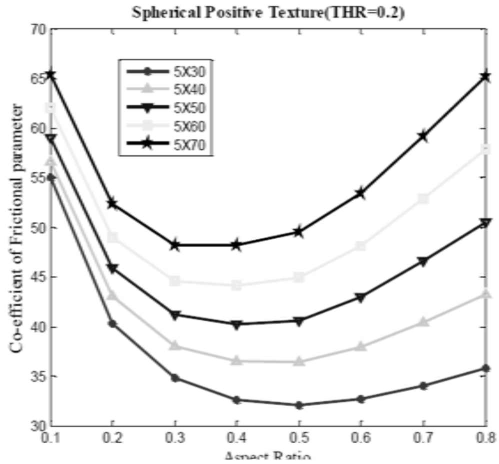

Fig. 6: Coefficient of friction parameters Vs. aspect ratio for different of positive spherical texture

Effect of Aspect Ratio on Non-dimensional Load

Hydrodynamic analysis for non-dimensional load on spherical textured thrust bearing is plotted in figure 5, which shows increase in non dimensional load carrying capacity with increase in aspect ratio, This is due to increase of roughness or increases in asperity interaction with adjacent asperity which result increase in pressure over the bearing surface so load carrying capacity increase. The load carrying capacity increase up to a certain value of aspect ratio (0.4 to 0.6), after that load carrying value decrease for higher value of the aspect ratio .The drop in load carrying capacity with increase in aspect ratio beyond a limiting value is because of smoothness of bearing surface. Figure 5 also indicates increase in load carrying capacity with decrease in texture density. When the less number of texture present in the bearing surface it behave like rough surface so the load carrying capacity is high .How ever if the number of texture is very high or very small ,the bearing surface behaves as smooth surface so the load carrying capacity decrease in bearing. This shows there is limiting value of aspect ratio as well as texture density for maximum load carrying capacity for a specific texture height ratio.

Effect of Aspect Ratio on Coefficient Friction Parameter

Figure 6 shows variation of coefficient friction parameter with the aspect ratio. Coefficient of friction parameter decrease with increase of aspect ratio, however it has a limiting value, on further increase the aspect ratio again the coefficient of friction increase. The limiting value is in the range of 0.4 to 0.6. For very low value of aspect ratio the roughness of pad surface is low, so load carrying capacity is small but frictional force is high so coefficient of friction is high. But when aspect ratio increases load carrying capacity increases hence coefficient of friction decreases. However further increase the aspect ratio the bearing surface act as very smooth surface so macroscopically roughness decreases so that adhesion force between lubricant and solid surface increases as a result coefficient friction increase. Coefficient of friction also depends on texture density, when texture density increases coefficient of friction increase.

(b). Effect of Texture Height Ratio on Non-Dimensional Load and Coefficient of Friction Parameter

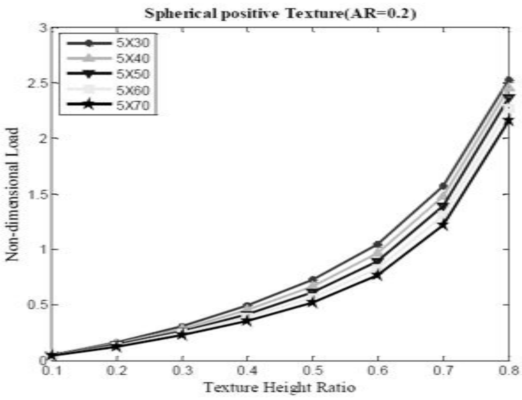

Fig.7 Non – dimensional load Vs. texture height ratio for different distribution of positive spherical texture

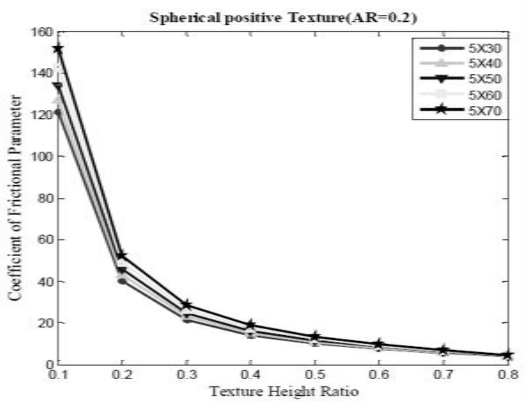

Fig.8 coefficient of friction parameter Vs. texture height for different distribution of positive spherical texture.

Effect of Texture Height Ratio on Non-dimension

Figure 7 shows non dimensional load of the bearing increase with increase in the texture height. .The increase in the texture height of the bearing restricts the flow of lubrication so cross-sectional area of flow of lubrication decrease due to decrease in cross-sectional area pressure in the bearing increase. The increase in pressure results in increase of non-dimensional load. Increase in texture density the bearing acts as smooth surface so the load carrying capacity of the bearing decrease.

Effect of Texture Height Ratio on Coefficient of Friction parameter

Coefficient of friction parameter in the bearing surface decrease with increase the texture height of the bearing(Figure 8).When the texture height increase the roughness of the bearing surface increase so adhesion force between lubricant and bearing surface decrease so coefficient of friction decrease. With increase in texture density co efficient of friction increase this is due to increase in texture density roughness of the bearing surface decrease. The texture height ratio has to be maintained above 20% for smaller coefficient of friction.

CONCLUSION

From above literature review, it is concluded that,

1. In controlling load and friction texture height ratio and aspect ratio plays an important role.
2. The aspect ratio and texture height ratio to be maintained above 20% as there is better performance parameters beyond that point.
3. Non dimensional load is increases as texture density decreases.
4. Non dimensional load is increasing when texture height ratio is increased.
5. Coefficient of friction parameter low when aspect ratio ranges 0.4-0.6.
6. Coefficient of friction parameter is low when texture height ratio ranges from 0.7- 0.8.

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