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# Isothermal Analysis of the Cylindrical Textured Hydrodynamic Parallel Plates

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**ABSTRACT: The performance of finite parallel plates having cylindrical textures is investigated numerically. Deriving a Reynolds equation for textured parallel plate is necessary for the assessment of static characteristics in terms of load of a lubricating system by using finite difference method (FDM) with central differencing scheme. It has been observed that the textured stationary plate (bottom) carry the load carrying capacity of the smooth moving plate (upper) due to the generation of lubricating pressures. It has been also found that the load carrying capacity of textured parallel plate increases with the increment of the number of textures in z direction. However, the fruitfulness of texturing deteriorated in the x-direction after certain point.**

**Keywords:** Cylindrical surface textures; finite difference method; isothermal analysis; lubricant pressure; load and reynolds equation.

**INTRODUCTION:** Surface texturing is the intentional introduction of well defined identical features on the surfaces. It is an effective way to increase load carrying capacity between sliding surfaces separated by a thin compressible or incompressible lubricant film. A patterned micro texture on one of the sliding surfaces, usually implemented as a dense array of micro sized concave features (“dimples”), increases the pressure in the lubricant, thereby increasing the load-carrying capacity of the bearing and reducing friction [1-3]. The fabricated micro-dimples work as lubricant reservoirs, cause to generate the lubricant pressure as well as act as the wear debris/foreign particles trappers too [3].

**LITERATURE REVIEW:** The micro-dimples of texture pattern fabricated by many precision manufacturing technologies [4] on the surfaces of tribo-elements. Brizmer et al. [5] theoretically investigated the effect of laser surface texturing (spherical) on the performance of parallel thrust bearing. Authors made a comparison with optimum linear and stepped sliders showing that parallel laser surface texturing sliders can provide similar load carrying capacity. Meng et al. [6] has been investigated the effect of dimples on friction of parallel surfaces under different sliding conditions. Authors observed that the rectangular dimples can reduce the friction coefficient for the smaller value of film thickness to roughness ratio. Milik & Kakoty [7] analyzed the effect of dimples on parallel and inclined slider bearing and concluded that the forward texturing has better performance than backward texturing. Olver et al. [8] observed that the maximum load carrying capacity occurs, when the pocket is located near the inlet to the bearing. Pascovici et al. [9] analytically investigated the effect of partial texturing on parallel slider bearings and found that texturing concept was very fruitful in the case of parallel slider bearings. Recently, Uddin & Liu [10] optimized the geometric texture for the enhancement of hydrodynamic lubrication performance of parallel slider surfaces. Optimized study indicated that the potential benefit of the new shape proposed by the authors also enhancing the hydrodynamic lubrication performance of slider bearing contacts. Texture shape optimization and the effect of viscosity wedge on the parallel plate also investigated by the researchers [11-12] and concluded that the texturing has fruitful results in the case of slider bearings. Based on the literature review, it is noticed that surface texturing concept is very fruitful in the case of lubrication problems. However, in above literature, some of studies based on theoretical or analytical methods and some of them on numerical methods for one dimensional problem. Thus, the objective of this paper is to present a numerical model for the isothermal analysis of hydrodynamically lubricated parallel plates having cylindrical surface dimpling on the bottom plate.

**METHODOLOGY:** In present work, the development of the mathematical and numerical models for the parallel plate bearing problems that has been studied. Finite difference method has been used for calculating the pressures numerically by changing the differential Reynolds equation into discretised form.

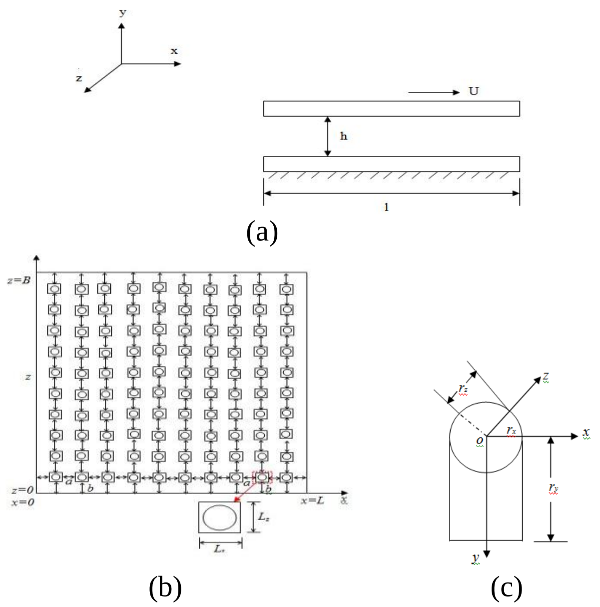



**Figure 1: Schematic diagrams for (a) smooth parallel plates (b) textured bottom plate (c) Cylindrical Surface texture details**

The schematic diagram for smooth parallel plate is presented in Figure 1 (a). The length in the *x*-direction is *L*, width in the *z*-direction is *B* and nominal film thickness for smooth bearing in the *y*-direction is taken as *h*. The analysis in present work has been carried out for a steady state, laminar flow, incompressible oil while neglecting the effects due to lubricant inertia and squeeze. The variation of pressure in *y*-direction is very small as compared to variation in pressure in *x* and *z* directions as the film thickness is very small; the continuity equation (1) and Navier-stokes equation (2) in Cartesian coordinates thus reduce to:

$$\frac{\partial u}{\partial x}+\frac{\partial w}{\partial z}=0 \tag{1}$$

$$\frac{\partial p}{\partial x}=\frac{\partial \tau_{yx}}{\partial y}$$
$$\frac{\partial p}{\partial z}=\frac{\partial \tau_{yz}}{\partial y} \tag{2}$$
$$\frac{\partial p}{\partial y}=0$$

Combined effect of above equation (1) and (2) with applying suitable boundary conditions, the 2-dimensional Reynolds-type equation for a parallel plate has been derived.

$$\frac{\partial}{\partial x}\left[\frac{h^3}{\eta}\frac{\partial p}{\partial x}\right]+\frac{\partial}{\partial z}\left[\frac{h^3}{\eta}\frac{\partial p}{\partial z}\right]=6U\frac{\partial h}{\partial x} \tag{3}$$

Where, *p* & *h* are the bearing pressure and nominal film thickness respectively whereas, η & *U* are notations for lubricant's viscosity and speed respectively.
The lubricant film thickness in case of smooth parallel plate is written as:

$$h_{smooth}=C \tag{4}$$

In the proposed investigation, the cylindrical surface textures have been considered as shown in Fig. 1 (b). Three dimensional geometry and scheme of a cylindrical dimple is defined as follows [14]:

$$r_c=\sqrt{(x-x_c)^2+(z-z_c)^2} \tag{5}$$

Where, $r_c$ is the radius of cylinder

Dimple centre $O_c$ ($x_c$, $y_c$, $z_c$) has 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, $x_c$, and $z_c$ are written as [14]:

$$x_c=n_1 a+\frac{(2n_1-1)}{2}L_x \quad \& \quad z_c=n_1 b+\frac{(2n_1-1)}{2}L_z$$

Where, $L_x$ & $L_z$ are the dimensions of unit cell & $n_1$ is number of dimples i.e. 1, 2, 3……..so on.
Expression for film thickness (h) in the computational domain is expressed as:

$$\begin{aligned} h &= h_{smooth} && \text{if} \quad r \geq r_c \\ &= h_{smooth}+r_y && \text{if} \quad r < r_c \end{aligned} \tag{6}$$

The pressure field in the lubricant film is numerically computed through equation (3) using finite difference method (FDM) with central differencing scheme. The pressure is computed iteratively through Gauss-Seidal method and an over relaxation factor of 1.7 is used for finding the solutions. The convergence criterion for Reynolds equation (3) is given below for pressure:

$$\frac{\sum\sum\left|(p_{i,k})_l-(p_{i,k})_{l-1}\right|}{\sum\sum\left|(p_{i,k})_l\right|}<10^{-07} \tag{7}$$

Where, i, k represent the number of nodes in x and z direction respectably and I is the number of Iterations.
Load carrying capacity is calculated from the integration of pressure parallel to the x-axis and z-axis as:

Load carrying capacity (W) = $\int_0^B\int_0^l p(i,k)\,dx\,dy$ (8)

## RESULTS AND DISCUSSIONS:

The results presented in Figures 2 have been found to be matching considerably well [13]. While validating due care has been taken for considering all the input parameters and other conditions.

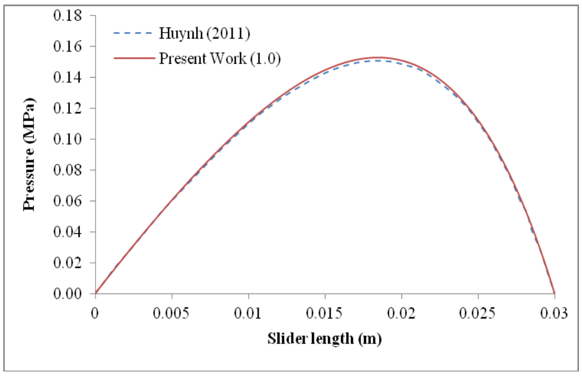


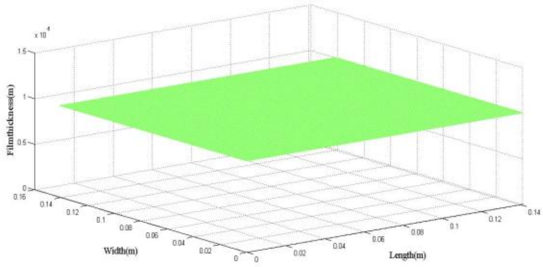

**Figure 2: Comparison of pressure results for smooth slider**

The input data for parallel plates attributes of textures and operational parameters employed in this work are presented in table 1 and some of the data has been taken from published experimental paper [11]. The input data are also provided along with tables and figures being presented.

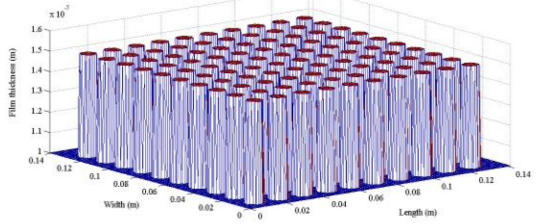
3-Dimensional representations of lubricating film thickness are shown in Figure 3 (a) & (b) for smooth and cylindrical textured plate respectively.

**Table 1: Input data**

| S.N. | Input parameters | Value |
|---|---|---|
| 1. | Length of the plates(L) | 0.1413m |
| 2. | Width of the plates(B) | 0.1413m |
| 3. | Dynamic viscosity($\eta$) | 0.38 Pa.s |
| 4. | Speed(N) | 400rpm |
| 5. | Number of nodes in x-direction($N_x$) | 150 |
| 6. | Number of nodes in z-direction($N_z$) | 150 |
| 7. | Number of textures in x-direction ($Nt_x$) | 1-10 |
| 8. | Number of textures in z-direction ($Nt_z$) | 1-10 |
| 9. | Dimple depth ($r_y$) | 5µm |
| 10. | Dimple radius ($r_c$) | 0.004m |
| 11. | Length along x-direction ($L_x$) | 0.008m |
| 12. | Length along z-direction ($L_z$) | 0.008m |
| 13. | Distance between two successive unit cells in x-direction (a) | 0.0055m |
| 14. | Distance between two successive unit cells in z-direction (b) | 0.0055m |
| 15. | Clearance(C) | 10µm |

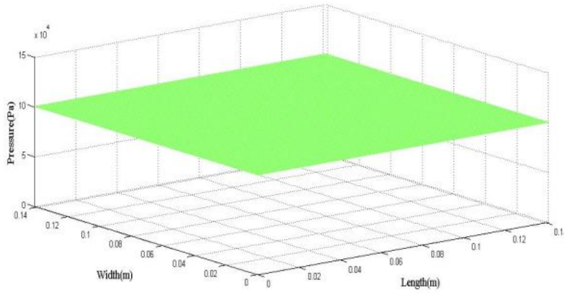



(a) $Nt_x = 0$ & $Nt_z = 0$

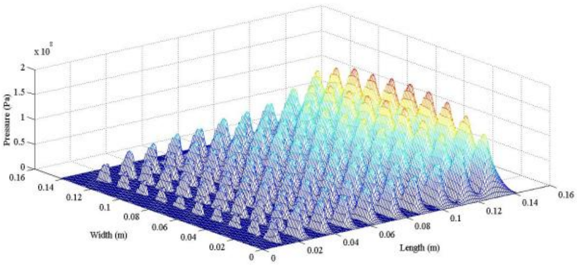



(b) $Nt_x = 10$ & $Nt_z = 10$

**Figure 3: Dimensional representation of lubricant film thickness**




(a) $Nt_x = 0$ & $Nt_z = 0$




(b) $Nt_x = 0$ & $Nt_z = 0$

**Figure 4: Dimensional representation of lubricant pressure**

Figure 4 (a) shows that there is no generation of lubricant pressures in the case of smooth plate. However, the generation of lubricant pressures between the smooth & textured surface as shown in Figure 4 (b).

With the increase in number of nodes in x & z directions, the load carrying capacity of the textured parallel plate changes as shown in Figure 5. It has been observed that load carrying capacity of parallel plate is increasing with the increase of number of nodes in longitudinal direction. However, the load carrying capacity of the textured plate first increases up to certain point ($Nt_x$=6), after this point the fruitfulness of textures decreased.

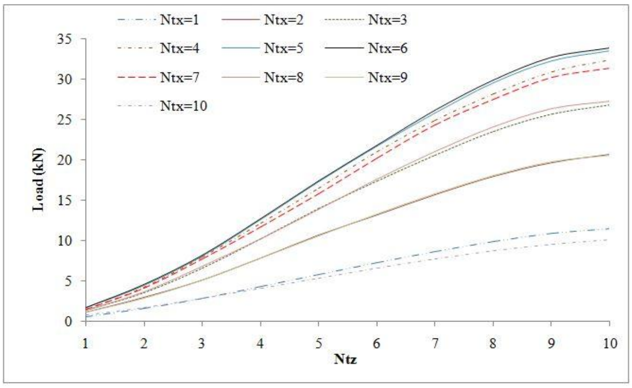



**Figure 5: Variation of load in x & z direction for textured parallel plate**

**CONCLUSIONS:** The cylindrical textures have been considered on bottom plate and the influence of these textures on the load carrying capacity of parallel plate has been investigated.
Following are the broad outcomes of present study:
1. There is the zero load carrying capacity in the case of smooth parallel plates.
2. Cylindrical surface textures help to generate load carrying capacity of the plates.
3. Load carrying capacity of textured plate increases up to certain point, when number of textures increases in x-direction.
4. Load carrying capacity is almost constant at the high value of the number of textures in z-direction.

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