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Effect of Spherical Surface Textures on the Static Performance of Inclined Slider Bearings at Different Locations

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ABSTRACT: The present study investigates the effect of spherical textures at different positions on the performance characteristic of finite slider bearings. Reynolds equation is numerically solved using finite difference method. The performance of inclined slider bearings is calculated for two different cases (Constant Convergence ratio & Constant Load carrying capacity). It is concluded that the partial surface texturing (PT-I) gives the fruitful results as compared with others for 1st case. However, in the case of constant load, the full and partial texturing II shows fruitful results compare with partial texture I and smooth case.

Keywords: Central differencing scheme; friction coefficient; hydrodynamic slider bearing; isothermal analysis and spherical surface textures

INTRODUCTION: Surface texturing is an effective way to improve the performance characteristics of inclined slider bearings i.e., it may act as lubricant reservoirs in the case of starved lubrication, entrap the wear & debris particles, minimizing third-body abrasion and also increase the load carrying capacity of slider bearings. [1-3]

Etsion's group presented the benefits of texturing for mechanical seals,[4] thrust bearings,[5-6] journal bearings[7] and piston rings.[8–10] Authors observed that the partial surface texturing is generally considered more efficient than full texturing in all above applications. Rahmani and co-workers[11] tried to find the optimal texturing parameters with an analytical approach, applying the one-dimensional Reynolds equation for infinitely long parallel sliders having textures with flat bottom profiles. Authors concluded that a minimal number of textures results in best performance, which implicitly stands for a maximum texture density. Recently, authors[12] have been optimized input parameter of dimples for the enhancement of hydrodynamic lubrication performance of parallel slider surfaces. It has been found 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[13-14] and concluded that the texturing has fruitful results in the case of slider bearings. Based on the literature review, it is noticed that in most of the studies the load carrying capacity is calculated by keeping the convergence ratio constant. Researchers observed that the partial surface texturing (at inlet) has fruitful results as compared with full textures. In above discussed 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 inclined slider bearings having spherical surface and calculate the friction coefficient for two cases (1st case- Constant convergence ratio & 2nd case- Constant load).

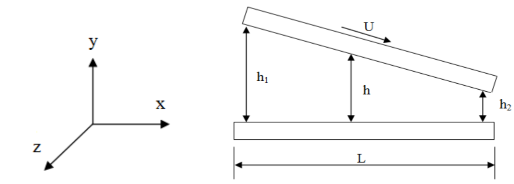
MATERIAL AND METHODS:

Mathematical and Computational Methods: In present work, the development of the mathematical and numerical models for the inclined slider 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.

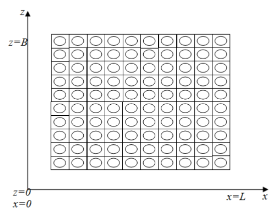
The schematic diagram for smooth inclined slider is presented in Figure 1. The length in the *x*-direction is *L*, width in the *z*-direction is *B* and nominal film thickness for smooth bearing in the *y*-direction is taken as *h*. The analysis in present work has been carried out for a steady state, 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:

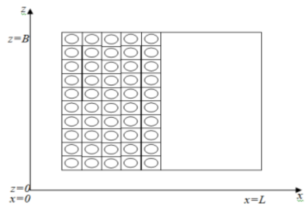
$$\frac{\partial u}{\partial x}+\frac{\partial w}{\partial z}=0 \tag{1}$$

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

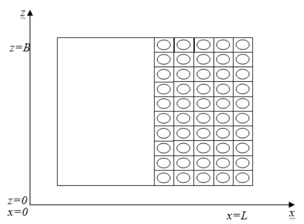


(a) Smooth case





(b) Full Textures (FT) **(c) Partial Textures I (PTI)**



(d) Partial Textures II (PTII)

Figure 1: Schematic diagrams for smooth and textured inclined slider bearing.

Combined effect of above equation (1) and (2) with applying suitable boundary conditions, the 2-dimensional Reynolds-type equation for an inclined slider bearing 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 inclined slider bearing is written as:

$$h_{smooth}=h_1-\frac{(Kh_2x)}{L} \tag{4}$$

Where, $Convergence\ ratio\ (K)=\frac{h_1-h_2}{h_2}$

In the proposed investigation, the spherical surface textures have been considered as shown in Fig. 2.
Three dimensional geometry and scheme of a spherical dimple is defined as follows[15]:

$$\Delta h_{spherical}=\sqrt{\left[\left(\frac{r_s^2+r_y^2}{2r_y}\right)^2-(x-x_c)^2-(z-z_c)^2\right]}-\left(\frac{r_s^2+r_y^2}{2r_y}\right)+r_y \tag{5}$$

Where, r_s is the radius of spherical dimple

Dimple centre O_c (x_c, y_c, z_c) has been indicated in Fig. 1. 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 [15]:

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

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:

$$h=h_{smooth} \qquad \text{if} \quad r\ge r_s$$
$$=h_{smooth}+\Delta h_{spherical} \qquad \text{if} \quad r<r_s \tag{6}$$

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:

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

Where, i, k represent the number of nodes in x and z direction respectably and I is the number of Iterations.
Friction force is calculated from the integration of shear stresses parallel to the x-axis and z-axis as:

$$\text{Friction force } (F)=\int_0^B\int_0^l \left|\tau_{yx}\right|_{y=0} dxdy \tag{8}$$
$$\tau_{yx}=\eta\frac{\partial u}{\partial y}$$

Coefficient of friction (*f*) =*F/W* (9)

% variation in coefficient of friction =

$$\left(\frac{f_{Texture}-f_{Smooth}}{f_{Smooth}}\right)\times 100 \tag{10}$$

RESULTS AND DISCUSSION: The results presented in Figure 2 have been found to be matching considerably well.[16] While validating due care has been taken for considering all the input parameters and other conditions.

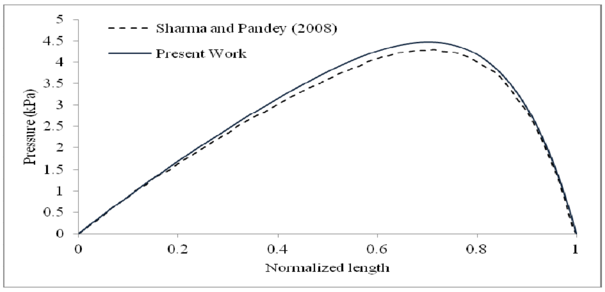
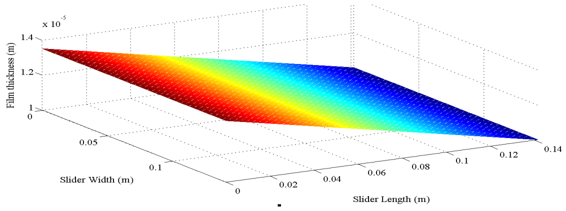


Figure 2: Comparison of pressure results for smooth slider.



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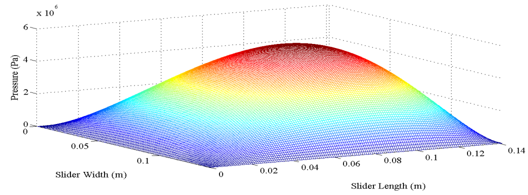

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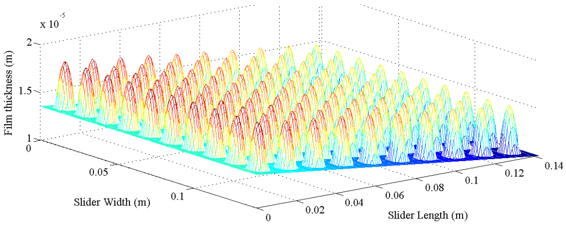

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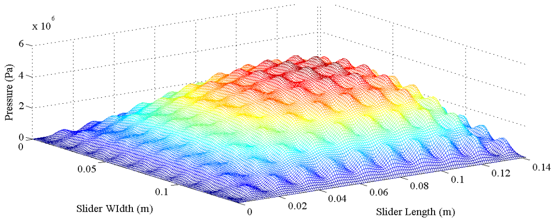

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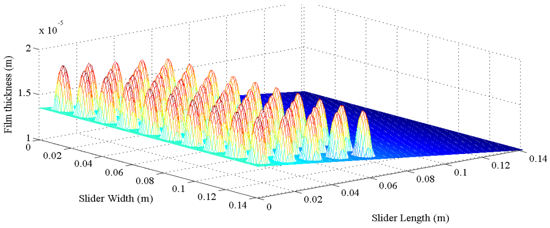

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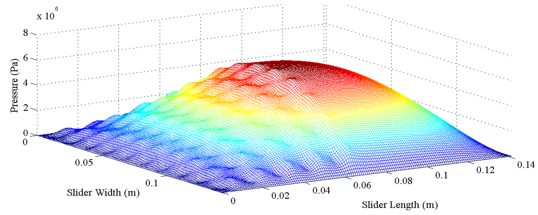

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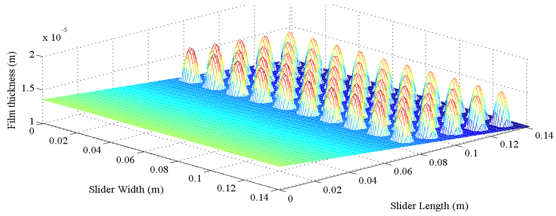
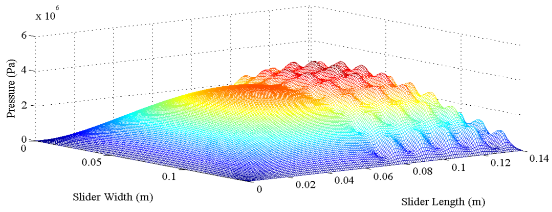


(g)



(h)

Figure 3: 3-D film thickness and pressure profiles for smooth and different textures.

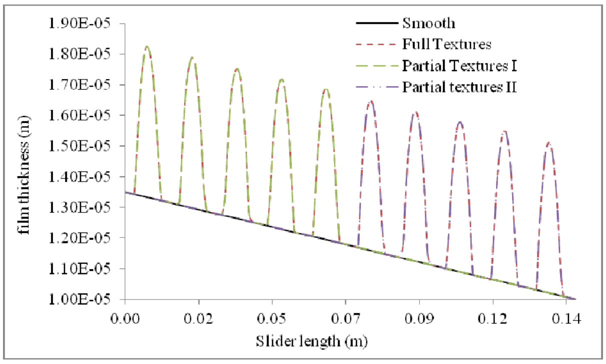
Table 1: Input data.

Sr. No.	Input parameters	Value
1.	Length of the plates(L)	0.1413m
2.	Width of the plates(B)	0.1413m
3.	Dynamic viscosity(η)	0.0064 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)	10
8.	Number of textures in z-direction (Nt_z)	10
9.	Dimple depth (r_y)	5µm
10.	Dimple radius (r_s)	0.006m
11.	Length along x-direction (L_x)	0.014m
12.	Length along z-direction (L_z)	0.014m
13.	Minimum film thickness (h_2)	10µm
14.	Convergence ratio (K)	Varying according to applied load
15.	Load (W)	1000 N

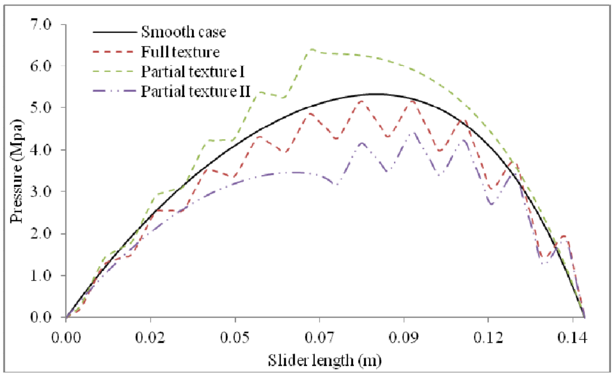
The input data for inclined slider bearing the attributes of textures and operational parameters employed in this work are presented in table 1 and some of the data has been taken from published experimental paper.[13]

The input data are also provided along with tables and figures being presented. In the present work, the friction coefficient for smooth and textured inclined slider bearing has investigated for the two cases i.e. keeping the convergence ratio (K) is constant (fixed pad bearing or constant maximum and minimum film thickness) & second case related with the constant load (only minimum film thickness is constant and vary the convergence ratio).

Case I: Convergence ratio (K) is constant and load (W) changes



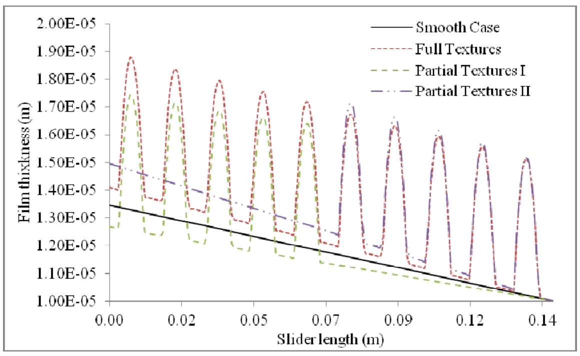
(a)



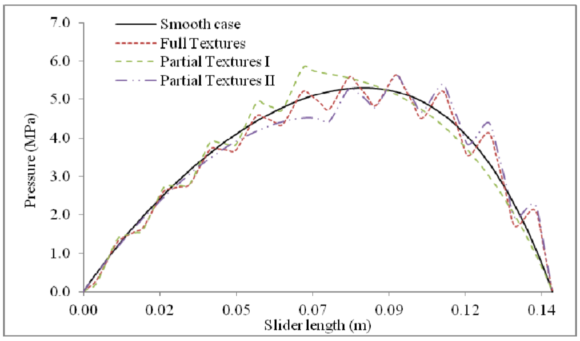
(b)

Figure 4: Film thickness and pressure variations along the slider length (Constant Convergence ratio).

Case II: Load (W) is constant and Convergence ratio (K) changes



(a)



(b)

Figure 5: Film thickness and pressure variations along the slider length (Constant Load).

Table 2: Comparison of different textures for both the cases.

%age change in friction coefficient		
Case I	Full Textures	**+3.76**
	Partial Textures I	**-13.65**
	Partial Textures II	**24.96**
Case II	Full Texture	**-7.99**
	Partial Textures I	**+0.75**
	Partial Textures II	**-8.83**

The three dimensional representation of the effect of different surface textures in terms of film thickness and lubricant pressure on inclined slider bearings shown in figs 3 (a-h). Fig. 4 (a) and 5 (a) shows the film thickness variation along the slider length for two cases. When convergence ratio kept constant, the partial surface texturing (PT I) show the high pressure generation as compared with other cases as shown in fig. 4 (b). However, in the second case, the variation of lubricant pressures along slider length almost same, achieve the constant load carrying capacity condition as shown in fig. 5(b). The overall comparison between two conditions for different surface textures is presented in table 2. It has been observed that PT I show fruitful results for 1st case, however, in the 2nd case, PT II reduced friction coefficient more as compared with other cases. In the second case, the maximum film thickness increases or decreases according to the position of surface textures, which caused to change the behavior of the results as compared with first case as shown in fig. 5 (a).

CONCLUSION: The spherical textures have been considered on inclined slider bearing. Following are the broad outcomes of present study:

1. Spherical surface texturing influenced the bearing performance.
2. The fruitfulness of surface texturing changed according to the operating conditions.

3. Partial texturing (PT I) reduced the friction coefficient up to 13.65 % in first case, however, 8% reduction in friction coefficient achieved in second case for Full Texturing (FT) & Partial Texturing (PT II).

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REFERENCES:

1. Etsion I. (2005) State of art in laser surface texturing, *Journal of Tribology*, 27, 248-253, 0742-4787.

2. Etsion I. (2013) Modeling of surface texturing in hydrodynamic lubrication, *Friction*, 1(3), 195-209, 2223-7690.

3. Gropper D., Wang L. and Harvey T.J. (2016) Hydrodynamic lubrication of textured surfaces: A review of modeling, *Tribology International*, 94, 509-529, 0301-679X.

4. Etsion I. and Halperin G. (2002) A laser surface textured hydrostatic mechanical seal, *Tribology Transactions*, 45, 430–434, 1547-397X.

5. Etsion I., Halperin G., Brizmer V. and Kligerman Y. (2004) Experimental investigation of laser surface textured parallel thrust bearings, *Tribology Letters*, 17, 295–300, 1573-2711.

6. Brizmer V., Kligerman Y. and Etsion I. (2003) A Laser Surface Textured Parallel Thrust Bearing, *Tribology Transactions*, 46, 397-403, 1547-397X.

7. Brizmer V. and Kligerman, Y. (2012) A laser surface textured journal bearing, *Journal of Tribology*, 134, 031702, 0742-4787.

8. Kligerman Y., Etsion I. and Shinkarenko A. (2005) Improving tribological performance of piston rings by partial surface texturing. *Journal of Tribology*, 127: 632–8, 0742-4787.

9. Ryk G., Kligerman Y., Etsion I., and Shinkarenko A. (2005) Experimental investigation of partial laser surface texturing for piston ring friction reduction. *Tribology Transactions*, 48, 583–8, 1547-397X.

10. Ryk G. and Etsion I. (2006) Testing piston rings with partial laser surface texturing for friction reduction, Wear, 261, 792–6, 0043-1648.

11. Rahmani R., Shirvani A. and Shirvani H. (2007) Optimization of partially textured parallel thrust bearings with square shaped micro dimples. *Tribology Transactions*, 50, 401–406, 1547-397X.

12. Uddin M.S. and Liu Y.W.(2016) Design and optimization of a new geometric texture shape for the enhancement of hydrodynamic lubrication performance of parallel slider surfaces *Biosurface and Biotribology*, 2, 59-69, 2405-4518.

13. Shen C. and Khonsari M.M. (2016) Texture Shape Optimization for Seal-like Parallel Surfaces: Theory and Experiment, *Tribology Transactions*, 1-32, 0569-8197

14. Menga X. and Khonsari M.M.(2017) On the effect of viscosity wedge in micro-textured parallel surfaces, *Tribology International*, 107, 116-124, 0301-679X

15. Kango S., Sharma R.K. and Pandey R.K. (2014) Comparative analysis of textured and grooved hydrodynamic journal bearing, *Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology***,** 228, 82-95, ISSN 2041-305X.

16. Sharma R.K. and Pandey R.K. (2008) Influence of surface profile on slider bearing performance, *Int. J. Surface Science and Engineering*, 2 (3/4), 265-280, 1749-7868.