Article Info

Influence of Pentagonal Texture on Performance of Slider Bearing

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ABSTRACT

Surface texture acts as reservoir and generating additional micro-hydrodynamic lift effect resulting improving tribological performance of bearing. The parametric analysis for influence of pentagonal surface texture on stationary surface of non-parallel slider bearing has been studied. In the isothermal condition, Navier-Stokes equation has been solved by finite volume method using ANSYS FLUENT. The bearing performance in terms of load carrying capacity, friction force and coefficient of fluid film friction have been calculated and normalized. Pentagonal texture with different area ratio has been modeled. Optimization of dimple parameters i.e. area ratio, height ratio, orientation and number of dimples in terms of hydrodynamic performance of bearing are investigated. It is observed that single dimple with right edge parallel to transverse axis having better performance. The load carrying capacity is increased with width and depth of texture up to a certain value after this vortex formation is occurred. Also the friction force is reduced with wider and deeper texture. This optimized dimple is offset toward left to transverse axis. Offset dimple shows best tribological performance of bearing.

Keywords: Slider Bearing; Pentagonal Texture; Optimization; Offset.

1.0 Introduction

Surface texturing is key in enhancing tribo-elements performance. Texture makes a cavity on the surface of bearing which acts as reservoir for lubricant and enhance the performance of the bearing. These reservoirs generate more pressure on the lubricant film for supporting the external load. A great deal of research has confirmed that the texture/dimple on the surface of the tribological elements is proving important in terms of reduction in coefficient of fluid film friction and increase load carrying capacity.

Etsion and Burstein (1996) mathematically analyzed regular hemispherical pores on non-contacting mechanical seals. The maximum axial stiffness and minimum friction torque are corresponding to optimization of the pore size.

A critical pore size and pore ratio of ring surface area are investigated in which seal failure is possible. Etsion et al. further (1999) investigated texture influence on the mechanical seals experimentally and compared with the theoretical model.

Geiger et al. (1998) improved tribological properties under hydrodynamic and elasto-hydrodynamic slider condition of ceramic surfaces with surface texturing. Alumina and silicon carbide confirmed that the structuring of the surface topography resulted in a satisfying improvement in the lubricant film thickness compared with untreated surfaces.

Kligerman and Etsion (2001) developed a theoretical model to examine the hydrodynamic effect of micro textures developed by laser surface texturing in a circumferential gas seal and results for a specific circumferential seal demonstrate a substantial hydrodynamic effect that can raise the opening average pressure in the seal clearance above the ambient one by up to 50 percent.

Ronen et al. (2001) used micro spherical surface structure in the form of micro pores in reciprocating automotive components and reported the reduction in friction.

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Etison and Halperin (2002) experimentally investigated laser surface texturing on mechanical seal to reduced friction and heat. It is found the surface texture seal have much pressure capability as compared to the simple unbalanced seal.

Ryk et al. (2002) experimentally studied on micro surface texturing on piston rings and cylinder liner segment to improve tribological properties of reciprocating automotive component. Brizmer et al. (2003) investigated surface texturing in the parallel thrust bearing using laser surface texturing and found optimum dimple parameter and best laser surface texture mode for maximum load carrying capacity.

Zhou et al. (2003) investigated the influence of surface texture on fly ability and lubricant migration under near contact conditions.

The textureson the air-bearing surface of magnetic recording sliders developed using Magnetron sputtering and ion beam sputtering. It is reported that texture sliders to cause less lubricant depletion on the disk surface than untexture sliders under these conditions. Etsion et al. (2004) experimentally investigated surface texturing on the parallel thrust bearing.

The hemispherical dimple texture on the parallel thrust bearing reduces friction and increased clearance as compared to untexture bearing.

McNickle and Etsion (2004) experimentally investigated the concept of near contact gas face seal for a high-speed gas turbine engine and compared it with baseline conventional contacting-type seal. Results are reported with smoother running, lower friction torque, and lower face temperature at a specified speed of 12,000 rpm over a range of face loading. Sahlin et al. (2005) numerically investigated micro texture in hydrodynamic lubrication of two parallel walls.

It is reported that load carrying capacity of bearing is increased and friction is reduced with deeper and wider groove.

Kligerman et al. (2005) studied the performance of piston rings with partial surface texture. Time behavior of the friction force is computed from the shear stresses in fluid film and the time dependent clearance and optimum parameters like dimple depth, area density and texture portion of the nominal contact surface of piston rings are calculated.

Murthy et al. (2007) developed a numerically model to study the influence of texture on air bearing slider for large Knudsen numbers. The flying height modulation, pitch, and roll motion of a texture slider (pico and femto form factors) are investigated by exciting the slider. It is observed that texture sliders show comparative good dynamic performance compared to the untexture sliders in terms of stiffness and damping.

Etsion et al. (2009) investigated the influence of texture piston rings on the fuel consumption and exhaust gas composition of a compression–ignition IC engine. Shinkarenko et al. (2010) theoretically analyzed soft elasto hydrodynamic lubrication (SEHL) that consists of a stationary surface-texture soft elastomer sleeve and a rigid, smooth rotating shaft.

The Reynolds equation and the linear elasticity equation were used simultaneously for the pressure distribution and viscous shear stresses in the fluid film and for the elastic stresses and deformations in the elastomer. It was reported that the model was also useful for predicting friction coefficient of different types of circumferential elastomer seals.

Cupillard et al. (2009) developed a numerical model to analyses three dimensional inlet texture slider bearing with a temperature dependent fluid. Simulations are executed for a laminar and steady flow and examined for various operating conditions. It is reported that texture has a good effect on load carrying capacity when thermal effects are considered and increased by up to 16% in operating conditions (high slider speed).

Rahmani et al. (2010) investigated the texture surface in hydrodynamic lubrication regime and reported the optimum texturing parameters promoting maximum load capacity, load capacity to lubricant flow rate ratio and minimum friction coefficient for asymmetric partially texture slider bearings. Ma and Zhu (2011) conducted an experiment on texture surface with elliptical shape dimples having varying depths, diameters, area ratios under condition for hydrodynamic lubrication.

A model for the optimum design of texture surface was modeled and then validated with the experiments and found optimum depth increases while the optimum diameter decreases as

the velocity becomes larger and the load becomes smaller. Ibatan et al. (2015) showed about recent development on surface texturing in enhancing tribological performance of bearing sliders. It is reported that micro dimples act as reservoir and give additional hydrodynamic lift and also there is reduction in the wear and increase the load carrying capacity.

Kango et al. (2014) developed a mathematical model for the thermal analysis of micro texture journal bearing using non-Newtonian rheology of lubricant and JFO boundary conditions. The decrease in average temperature of lubricant is found of texture bearing with respect to untexture bearing. Also JFO boundary condition provides numerical stability in getting the converged results with micro-texture surfaces of journal bearing. Tala - Ighil et al. (2011) investigated the effect of texture area on the performance of journal bearing. Different shapes of micro cavities (textures) and different locations of the texture zone were evaluated to improve the performance of bearings. The optimal design found with help of the geometrical parameters and the operating conditions of the journal bearing with both convergent (hydrodynamic pressure) and divergent (cavitations) zones. It is reported that partial texturing has a small positive effect and full texturing has a negative effect.

From the above literature review of the surface texturing, it is observed that surface texturing in tribo-components enhances its performance and texturing parameters are well optimized by different researchers.

A few studies were performed to examine the performance of the different types of texturing using different numerical models and techniques. From the available literature, the study conducted on pentagonal texture is scant.

In the current study, pentagon texture is modeled on stationary surface of finite bearing with different orientations. These dimples are analyzed for the performance of finite bearing with different geometrical parameters.

The Navier-Stokes and continuity equations are utilized to calculate the load carrying capacity and friction coefficient with the help of finite volume method which is solved by ANSYS Fluent. In present study the height ratio, dimple depth, change in orientation, number of dimples, offset of dimple and

area ratios geometrical parameters are considered for optimizing the dimple cavity in terms of maximizing load carrying capacity and minimizing friction force of the texture slider bearing.

2.0 Numerical Models

2.1 Basic governing equations

Assuming that the lubricant flow between bearing surface is laminar and in steady state under isothermal conditions. There is no variation of pressure across the fluid film and no slip in fluid. Also Inertia and body force terms are negligible compared with the pressure and viscous terms and no external force act on bearing. The continuity and Navier-Stokes equations are used to calculate the tribological performance parameters with the help of finite volume method which is solved using ANSYS Fluent.

If a steady-state problem is being solved iteratively, it is not necessary to fully resolve the linear pressure-velocity coupling, as the changes between consecutive solutions are no longer small. The SIMPLE (Semi-Implicit Method for Pressure-Linked Equations) algorithm an approximation of the velocity field is obtained by solving the momentum equation. The pressure gradient term is calculated using the pressure distribution from the previous iteration or an initial guess. These equations are expressed as:

$$\nabla.u = 0 \tag{1}$$
$$\rho(u.\nabla)u = -\nabla p + \eta\nabla^2 u \tag{2}$$

2.2 Geometric studies and boundary condition

In order to validate the proposed model, the simulated results for plane finite slider bearing are computed and the results are compared with reference (B.C. Majumdar, "Introduction to Tribology of Bearings"). The details of the finite slider bearing are shown in figure 1.

Thegeometric parameters of the bearing are commonly adopted value and have been taken from reference (B.C. Majumdar, "Introduction to Tribology of Bearings").

The length L=0.05m, breadth B=0.05m, maximum film thickness t_a =0.00004m, attitude (t_a/t_b) λ=2, absolute viscosity η=0.02Pa s and slider

velocity u=5m/s. The computed results are presented in Table 1. It may be seen that the results are in good agreement and thus establishes the accuracy of modeling procedure.

In Figure1showsthe finite slider bearing having upper and lower surfaces separated by lubricant film. The upper surface is moved with velocity 5m/s and lower surface is stationary. At the moving surface the fluid moves with the same velocity and at the stationary surface the fluid velocity is set to zero. All sides of the bearing, the pressure is set at zero (p=0 at x=0, x=B, z=0 and z=L).

The pentagonal dimples are generated on the stationary surface and optimization of the dimple is done with the help of the geometrical parameters like dimple depth (td), height ratio (HR=td/tb), change in orientation, number of dimples, area ratio (AR= ratio of area of dimple cavity to area of bearing surface) and offset of the dimple to obtain the most extreme load carrying capacity (W) of the texture bearing with maximum reduction of friction force (Ff) and coefficient of fluid film friction (μ) and the equations of the load, friction force and friction coefficient are written below:

$$W = \int_0^L \int_0^B p\,dx\,dz \qquad (3)$$
$$F_f = \int_0^L \int_0^B \tau\,dx\,dz \qquad (4)$$
$$\mu = \frac{F_f}{W} \qquad (5)$$

Results are represented in non-dimensional form in terms of the ratio between parameters of the texture bearing to divide with plane slider bearing parameters. The tribological parameters in terms of load carrying capacity ratio and coefficient of fluid film friction ratio are studied and their ratios are Wd/W and μd/μis computed for comparison.

2.3 Meshing method

The current investigation, finite volume method is used to discretize the lubricating film domain. The ANSYS FLUENT software is used to discretize the fluid film domain. A uniform multi zone hex mesh element is used for grid generation. Before using dimple bearing, a plane bearing is used to discretize with uniform multi zone hex mesh for validation of results with reference (B.C.

Majumdar, "Introduction to Tribology of Bearings"). The 5530 elements are generated for the

accuracy of the results and calculate the tribological parameters which are shown in the Table1.

The finest and dense grids are generated for dimple domain which is the most important area of the bearing to calculate the pressure accurately. The 5500-6000 elements are used to investigate the performance of the texture bearing. Dimple domain is divided into 5 sub-layers for the accuracy of results.

Table1 Comparison of Results

S.No.	Parameters	Present work	B.C. Majumdar
1.	Load carrying capacity (W)	2152.69 N	2153 N
2.	Friction force (Ff)	9.1056 N	9.035 N
3.	Coefficientof fluid film Friction(μ)	4.229×10^{-3}	4.197×10^{-3}

Table 2 Texture Parameters for Combined Effect of Orientation and Optimum Area Ratio

Height ratios	0.1	0.25	0.5	0.75	1	1.25	1.5
(HR)							
Area ratios (AR)	10%	20%	30%	40%			

Fig. 1 Slider Bearing
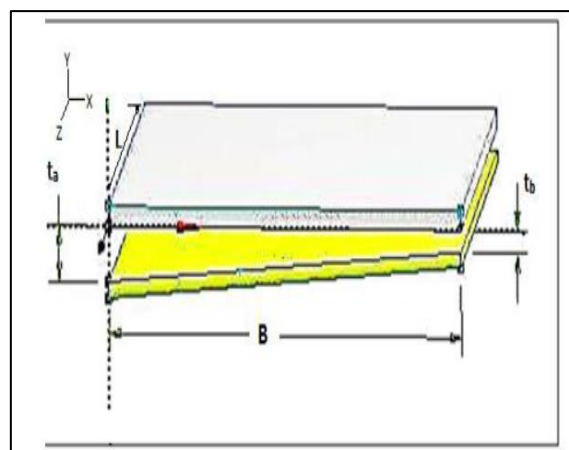
3.0 Results and Discussion

In order to carry out the simulation task, a single dimple and multi dimples are modeled at the center of the stationary surface. Assuming the isothermal condition the lubricant viscosity and attitude are kept constant. The slider bearing with pentagonal texture on the stationary surface is studied and simulation is performed to study for following effect on bearing performance.

1. Combined effect of area ratio and texture orientation
2. Effect of texture numbers
3. Effect of offset

3.1 Combined effect of area ratio and texture orientation

The pentagonal texture shown in figure 2 is located at the center of the stationary surface. In position A, the left edge of texture is parallel to the transverse axis. In position B, the lower edge of texture is parallel to the longitudinal axis. In position C, the right edge of texture is parallel to the transverse axis. Texture parameters like height ratio and area ratios are considered for a wide range shown in Table 2. The influence of these parameters on load carrying capacity ratio and coefficient of fluid film friction ratio for different positions is studied. In the simulation task, on area ratios varying from 10% - 40% is examined. The layout of different surface texture with varying area ratios are shown in figure 3.

3.1.1 On load carrying capacity ratio

In order to compute optimum value of area ratio, the simulation is carried out for different value of area ratio. Based on the simulation work it is observed that corresponding to 36% area ratio i.e. (AR = 36%) for position A the maximum load carrying capacity ratio is observed as shown in Figure 4. Further the load carrying capacity ratio is gradually increasing for height ratio ranging 0.1-1 and then a decrease in the curve is observed. A similar trend for load carrying capacity ratio is observed for position B also and the plot is shown in Figure 5. The plot indicates the maximum performance in terms of load carrying capacity ratio corresponding to area ratio of 35%.

The graph shows an increasing trend for height ratio from 0.1-0.75 and then decreases beyond the value HR >0.75.

Fig. 2 Orientation of Regular Pentagonal Texture (a) When Left Edge is Parallel to Transverse Axis(b)When Lower Edge is Parallel to Longitudinal Axis(c) When Right Edge is Parallel to Transverse Axis
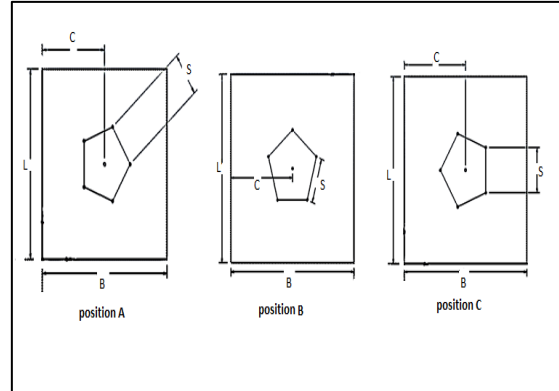

Fig. 3 dimple with Position C for Area Ratios (a) 10% (b) 20% (c) 30% (d) 40%
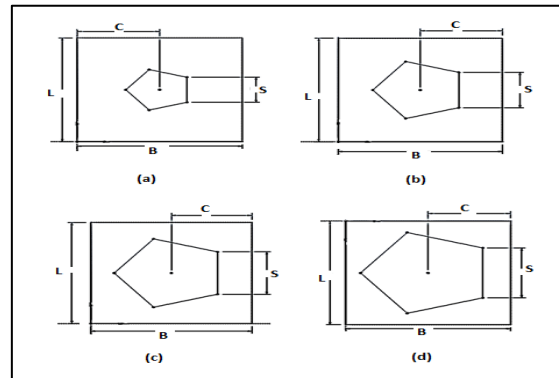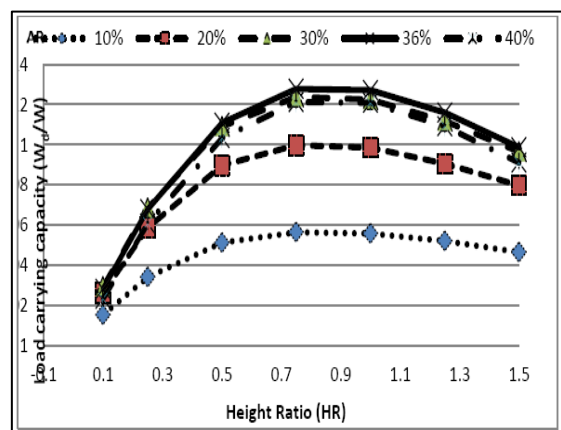

Fig. 4 Load Carrying Capacity Ratios Versus Height Ratios For Position A

Load carrying capacity ratio is also computed for position C. It is observed that corresponding to 36.25% area ratio for position C, the maximum load carrying capacity ratio may be seen in Figure 6.The load carrying capacity ratio is increasing for height ratio 0.1–1 and then a decrease in the curve is observed. In general, it is observed that corresponding to the area ratios 30% -40%, each texture with varying orientation showing the maximum load carrying capacity.

Fig. 5 Load Carrying Capacity Ratios Versus Height Ratios For Position B
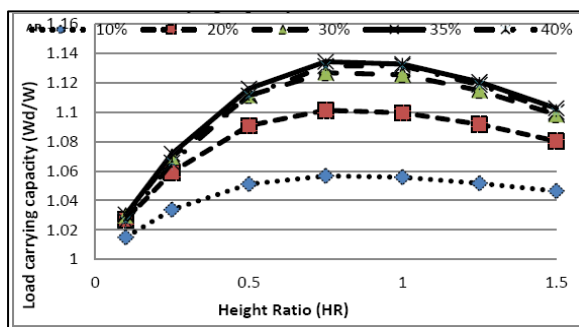
Fig.6 Load Carrying Capacity Ratios Versus Height Ratios for Position C
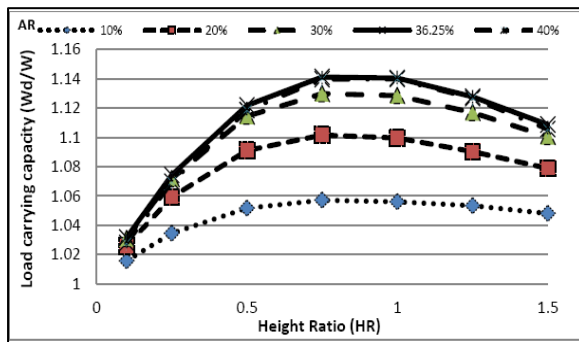
Fig.7 Coefficient of Friction Ratios Versus Height Ratios for Position A
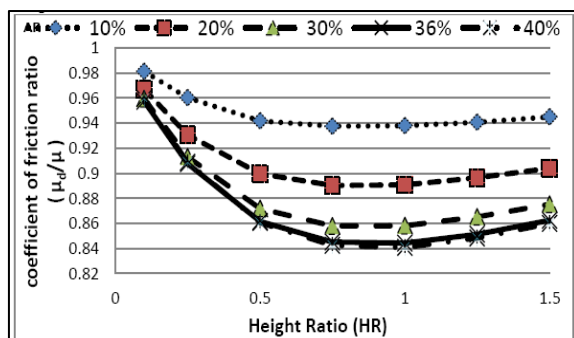

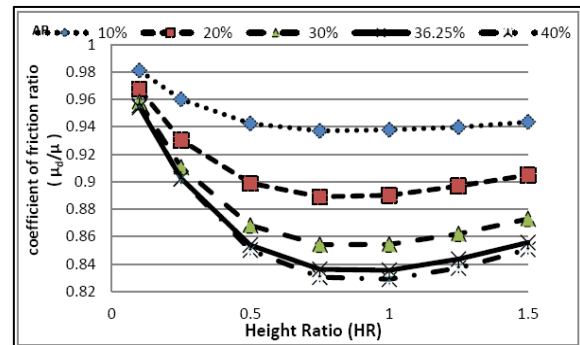**Fig. 8 Coefficient of Friction Ratios Versus Height Ratios for Position B**

Fig. 9 Coefficient of Friction Ratios Versus Height Ratios for Position C
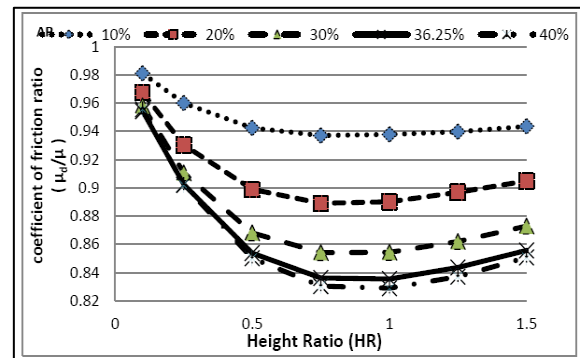

Fig.10 Multi Dimples with 10% Area Ratios (a) Single Dimple (b) Double Dimples Across Length (c) Double Dimples Along Length (d) four Dimples
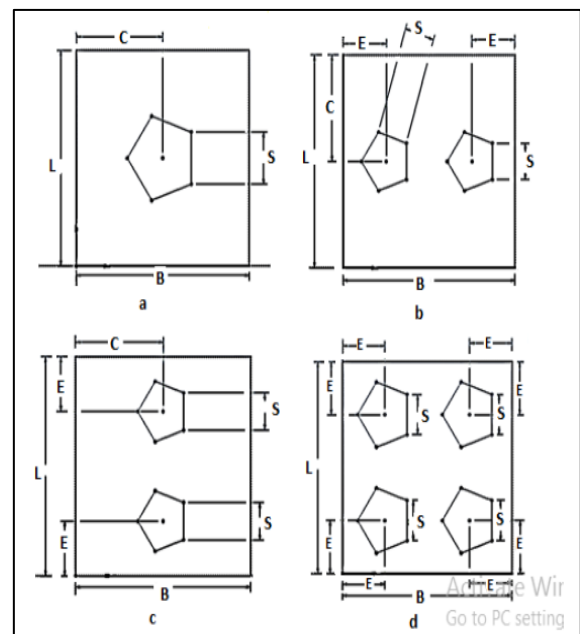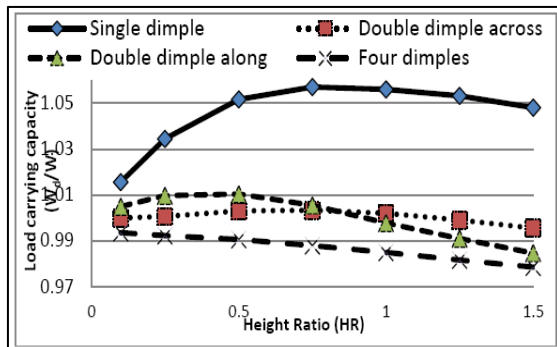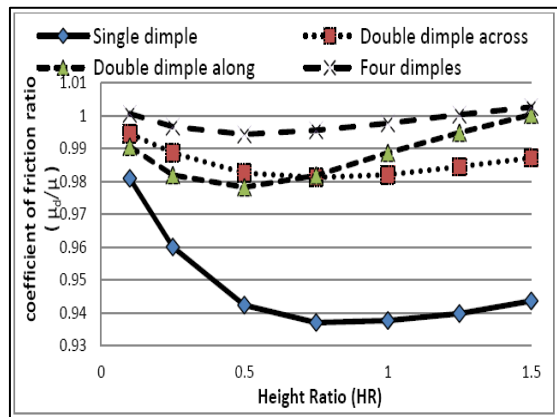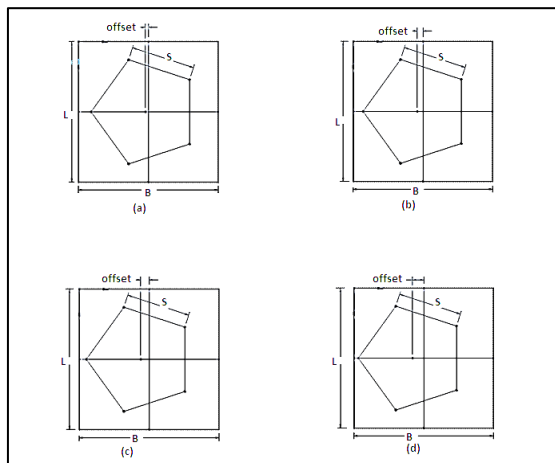
Fig. 11 Load Carrying Capacity Ratios Versus Height Ratios for Multi Texture

Fig. 12 Coefficient of Friction Ratios Versus Height Ratios for Multi Texture

Fig. 13 Single Dimples with 36.25% Area Ratios off Set for (a) 2% Length of Bearing (b)4% Length of Bearing (c) 6% Length of Bearing (d)8% Length of Bearing

It may be seen from Figures 4-6, that the bearing shows better load carrying capacity ratio for height ratio ranging HR = 0.75–1, beyond this region there is reduction in load carrying capacity as vortex formation is occurred inside of the domain. Dimple orientation for position C shows best performance amongst the selected parameters as shown in Figure6. Corresponding to 36.25% area ratio, load carrying capacity is increased by 14.1% at HR = 0.75 as compared to plane bearing.

Fig.14 Load Carrying Capacity Ratios Versus Height Ratios for Different Offset of Single Dimple
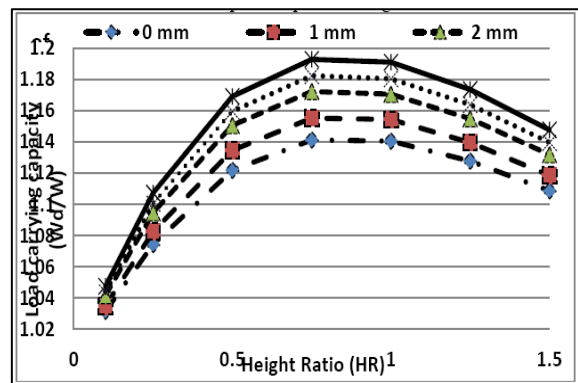
Fig. 15 Coefficient of Friction Ratios Versus Height Ratios for Different Offset of Single Dimple
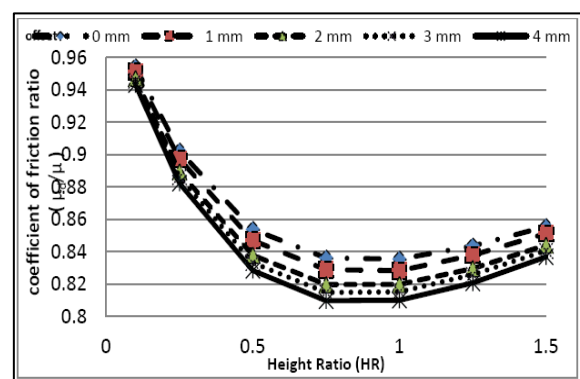
3.1.2 On coefficient of friction ratio

Following the same simulation, the results are computed for coefficient of friction ratio. A coefficient of friction ratio is computed for different orientation and for wide area ratios. It is observed that 40% area ratio shows best performance as minimum

value of coefficient of friction ratio for different orientation may be seen in Figures 7-9. Plots indicate that coefficient of friction is decreasing for height ratio 0.1 – 1 and then an increase in curve is observed. For position C coefficient of friction is reduced by 17% for HR= 0.75 as compared to plane bearing. It means that deeper and wider dimple shows the more reduction in friction coefficient but for this area ratio the load carrying capacity is slightly less as compared to corresponding optimum dimple for different orientations. Therefore, dimple orientation for position C with area ratio of 36.25% may be treated as best dimple in single dimple.

3.2 Effect of texture numbers on bearing performance

Based on above investigation it is found that dimple orientation corresponding to position C for area ratio 36.25% shows best performance. Therefore for the same optimum position C, a single dimple, double dimples along length, double dimples across length and four dimples on bearing surfaces have been compared as shown in Figure10. For this investigation area ratio 36.25% if adopted for double dimple, then the texture/dimple crosses bearing dimension.

Thus the area ratio is uniformly adopted to be 10%. In case, the area ratio is varied to some higher values say 20%, a similar trend on bearing performance is observed. Texture parameters considered are texture number and height ratio.

3.2.1 On load carrying capacity ratio

It is observed that the load carrying capacity is increased by 5.7% for single dimple with respect to plane slider bearing. The load carrying capacity ratio for single dimple is best among the different dimple undertaken for simulation task. The results are shown in Figure 11. Also it is found that the huge difference in multi texture is cause of the location of dimples on bearing surface.

3.2.2 On coefficient of friction ratio

The coefficient of fluid film friction is reduced to maximum for a single dimple as shown in Figure12.

A reduction of 12.7% on friction coefficient for single dimple as compared to plane bearing has been observed in the study. Thus the single dimple is more effective as compared to the multi dimples.

3.3 Effect of offset on bearing performance

In order to carry out the simulation task for studying the effect of offset, the best performance single dimple orientation corresponding to position C for area ratio 36.25% is considered. In the current simulation work, the dimple is offset toward left to transverse axis. The offset of dimple for various positions shown in Figure13.

The simulation work is computed in four numbers of sets for offset of 2%, 4%, 6%, 8% of the length of bearing. Texture parameters considered are offset of dimple and height ratio. The offset distance is varied in incremental steps of 1mm each, till its left vertex reaches nearer to the left side of bearing length.

3.3.1 On load carrying capacity ratio

Based on simulation data, it is observed that dimple at extreme left location shows best performance in among all location undertaken for simulation task that may be seen in Figure14. Load carrying capacity of 8% offset dimple is increased by 19.27% at HR= 0.75 and AR =36.25 as compared to the plane slider bearing. An optimum results for load carrying capacity ratio is observed for height ratio ranging 0.75 – 1.

3.3.2 On coefficient of friction ratio

From the simulation data, the result for the coefficient of friction ratio is plotted and is shown in Figure 15. A maximum reduction in coefficient of fluid film friction ratio is observed for offset of 8% of length of bearing.

The coefficient of friction of 8% offset dimple is reduced by 19.03% at HR= 0.75 and AR =36.25 as compared to to the plane bearing.

4.0 Conclusions

The numerical model, validation and influence of pentagonal texture parameters on slider bearing have been studied. Based on the results, the following conclusions are made:-

1. The bearing gives best performance for height ratio 0.75 and area ratio 36.25% when the right edge of texture is parallel to the transverse axis.

2. A single texture gives superior performance Vis a Vis multi texture.

3. The best tribological performance of the finite slider bearing is obtained when the surface texture is shifted towards left to transverse axis.

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Nomenclature

p = pressure N/mm^2
η = viscosity Ns /mm^2
HR=height ratio (t_d/t_b)
t_d=dimple depth mm
t_a=maximum film thickness mm
t_b=minimum film thickness mm
AR = area ratio (A_d/A)
A_d = area of dimple mm^2
A = area of bearing mm^2
u=velocity of slider plate m/s
B = width of bearing mm
L = length of bearing mm
S = edge of pentagonal texture mm
C = distance between edge of bearing to the center of bearing mm
E = distance of center of dimple to the adjacent edge of bearing mm
ρ = ratios kg/m^3
W_d =load carrying capacity of dimple bearing
W = load carrying capacity of plane bearing
μ_d = dimple coefficient of fluid film friction
W_d/W =Ratio of load carrying capacity of dimple Bearing to plane Bearing
μ_d/μ =Ratio of coefficient of fluid film friction of dimple to plane bearing.
F_f = Friction force of smooth bearing
F_{fd} =Friction Force of texture bearing
τ = Shear force