Influence of Sinusoidal Grooving On Pressure of Short, Finite, Infinite Slider Bearings

Chetan Sharma & Dalgobind Mahto

Abstract- In this article a sinusoidal grooved slider bearing is numerically analysed. A Reynolds equation and finite difference method are used to calculate the pressures of short, infinite, and finite slider bearings. The partial sinusoidal grooves are placed on different locations of slider bearings. The effect of change in speed and viscosity is also investigated in this work. It is observed that partial short grooved bearing shows maximum increment in pressure as compared with infinite and finite bearing. In the case of isothermal analysis the influence of speed and viscosity in percentage is negligible.

Keywords: *Sinusoidal Groove Short, Finite, Infinite Slider Bearing, Pressure, Finite Difference Method.*

1. Introduction

Tribology is the study of lubrication, friction, wear, and contact mechanics in order to understand surface interactions and to suggest solutions to fundamental problems. The expanding range of tribological applications, from the traditional industrial machinery to recent applications in micro fabrication has demonstrated its importance and also revived interest in this field.

The introduction of a range of micro fabrication techniques coupled with developments in microscopy has a profound effect on the reappearance of tribological applications at microscopic level. With the help of this new technology, it is now possible to produce microstructures on journal bearing surfaces to improve the overall tribological performance including reduction in friction, improvement in reliability, increase the pressure and load carrying capacity and lowering the power consumption. Surface texturing has been a subject of several theoretical and experimental studies. This is due to the fact that small improvements in bearing performance can be greatly economically beneficial. Most theoretical results reported so far are for isothermal studies and some are obtained for single stationary unit cell with periodic boundary conditions. The case of flow between two surfaces, one of which is textured, has been investigated by different researchers in past [1-27]. A positive effect of surface texturing on the lubricated contact performance is found in almost all the cases analyzed.

Patir et al. (1979) pointed out that the load carrying of the sliding surfaces increases appreciably if the moving surface is smooth and decrease if the moving surface is rough. Tonder (2001) suggested that by introducing a series of dimples or roughness at inlet of a sliding surface contact could generate extra pressure and thus support higher load. Ronen et al. (2001) presented a model to study the potential use of micro surface structure in the form of micro pores to improve tribological properties of reciprocating automotive components. It is shown that surface texturing can efficiently be used to maintain hydrodynamic effects even with nominally parallel surfaces, and that optimum surface texturing may substantially reduce the friction losses in reciprocating automotive components. Brizmer et al. (2003), discussed the potential use of a new technology of laser surface texturing (LST) in parallel thrust bearings. In this work the surface texturing has the form of micro dimples with preselected diameter, depth, and area density.

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These authors adopted optimum parameters of the dimples, and best LST mode, to obtain maximum load carrying capacity for a thrust bearing having parallel mating surfaces. There exists an optimal depth for which the dimples can provide the highest load carrying capacity as underlined by Yu et al. (2000).

Siripuram et al. (2004) presented a numerical study of the effects of different shapes of micro asperities in sliding surface lubrication for hydrodynamic films. Positive and negative asperities of constant heights are considered with circular, square, diamond, hexagonal, and triangular asperities give the smallest leakage rate whereas, the square asperities provide the largest. The minimum coefficient of friction for all shapes is found to occur at an asperity area fraction of 0.2 for positive asperities and 0.7 for negative asperities as reported by Siripuram et al. (2004). In a study of corrugated surfaces, Huynh (2005) pointed out that the load could be increased when the corrugated surface is well located on a fixed-incline slider bearing. Nevertheless, by introducing such a pattern, comparison with the smooth case is not reasonable since the global film thickness is reduced such that pressure increases when the corrugation amplitude increases. Brajdic Mitidieri et al. (2005) studied the influence of the convergence ratio and position of a pocket on the global friction coefficient for a pocketed pad bearing. Pocket depth and position were also found to be of importance for reduction in the friction coefficient by these authors.

Sinanoglu et al. (2005) compared the experimental result with numerical result. Authors used two types of surface textures one is Trapezoidal surface and other is Saw teeth surface. Authors found that Trapezoidal textured shaft can carry more loads then the saw textured shaft. In many studies, Reynolds or Stokes equations are employed neglecting pressure gradient across the lubricating film and inertia effects. However, Arghir et al. (2003) have shown that Stokes equations are inadequate to predict pressure build-up with the presence of macro-roughness as inertia effects can be of importance. This finding was confirmed later by Sahlin et al. (2005) who also presented an optimization of the geometry. Lu et al. (2007) investigated the dimple effect on the Stribeck curve of journal bearings experimentally. Authors pointed out that the typical friction a characteristic of a dimpled journal bearing is similar in trend to that of a conventional bearing, as prescribed by Stribeck curve. Further they reported that proper dimple size, shape and depth are essential to improve the friction performance and that it becomes more pronounced if oil with lower viscosity is utilized.

Tala-ighil et al. (2007) analyzed two cases, one in which the shaft is assumed to be smooth and rigid and one in which the bearing surface is numerically textured with spherical dimples. The numerical results indicate that textures affect the most important bearing characteristics viz: film thickness, pressure distributions, axial film flow, and frictional torque. The analyses of spherically dimpled surface indicate that

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appropriate selection of size, depth, and number of dimples may affect bearing characteristics. Tala-ighil et al. (2008) also analyzed different configurations (spherical shape, cylindrical shape, parallelepiped shape assuming the shaft is smooth and rigid and the bearing surface is numerically textured and reported that the positive effect of textured on contact characteristics becomes significant for the parallelpipedic shape rather than for cylindrical or spherical shapes, suggesting the advantage of parallelepipedic shape over other geometries. Jourak (2008) studied one dimple effect on the shaft of the bearing and found that due to the one dimple, there is no significant effect on load carrying capacity. Author suggested that increases the number of dimples may improve the journal bearing performance.

Cupillard et al. (2008) studied a complete textured bearing using the full Navier-Stokes equations and a cavitation model. As the authors illustrated, the coefficient of friction can be reduced if dimples of suitable width are introduced. According to the authors, this can be achieved either in the region of maximum hydrodynamic pressure for a bearing with a high eccentricity ratio or just downstream of the maximum film for a bearing with a low eccentricity ratio. The authors reported an additional effect of pressure build up that is generated by surface texturing for journal bearing at low eccentricity ratios. Cupillard et al. (2008) explained a mechanism of pressure build up in a convergent gap due to texture. Authors have found that the mechanism of pressure build up in a convergent gap between two sliding surfaces due to texture is similar to that obtained with variation of convergence ratio for smooth surfaces. As the fluid receives energy from the moving wall, lower losses in the inlet than in the outlet part produce positive variation of the mechanical energy in the inlet part and pressure build up. Pressure gradient decreases when recirculation occurs, i.e. when a too large value of dimple depth or the convergence ratio is used. Lower pressure is generated locally. Wall shape controls the velocity profile, which determines the pressure gradient and the pressure build up by means of the continuity equation.

Wang et al. (2012) investigate the influence of surface texture on the performance of hydrodynamic journal bearing and concluded that the partial texture improved the bearing performance. Kango et al (2010-2012) also investigate the influence of different type of surface grooving on the performance of hydrodynamic journal bearing and absorb that partial grooving improves bearing performance depending on eccentricity ratio.

The influence of sinusoidal grooving numerically introduced by Huynh (2005) and used the finite element numerical model to calculate the performance of infinite slider bearing. However, in the present work, by using the concept of Huynh (2005), the sinusoidal grooved model is used with Reynolds equation and finite difference numerical method is used to calculate the pressure for short, infinite, and finite bearing. The main objective in the present work is to compare the maximum pressure for three type of slider bearing with partial grooving at three different locations. The effect of change in speed and viscosity is also investigated in this work.

2. Numerical Model and Computational Procedure

Governing Equations:-

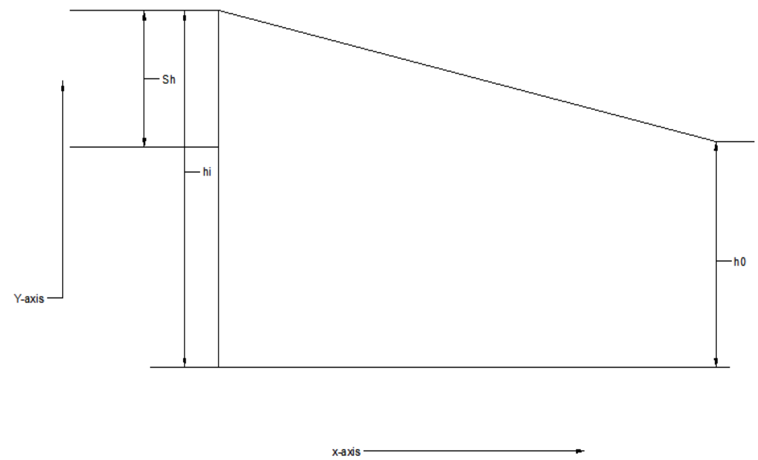


Figure 1.Schematic diagram of smooth slider bearing considered in present investigation with coordinate system

The lower surface is moved with velocity (U) in x direction, and axial width, (b) in z direction, and film thickness (h) in y direction. It is essential to mention here that the proposed investigation has been carried out for incompressible, iso viscous and steady sate laminar flow of lubricating oil. Moreover, body and inertia forces have been ignored herein.

The Reynolds equations used for incompressible fluid hydrodynamic bearing are;

$$\frac{\partial}{\partial x}\left(h^3 \frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(h^3 \frac{\partial p}{\partial z}\right)=6\eta\left(U\frac{dh}{dx}\right) \tag{1}$$

Film thickness expression

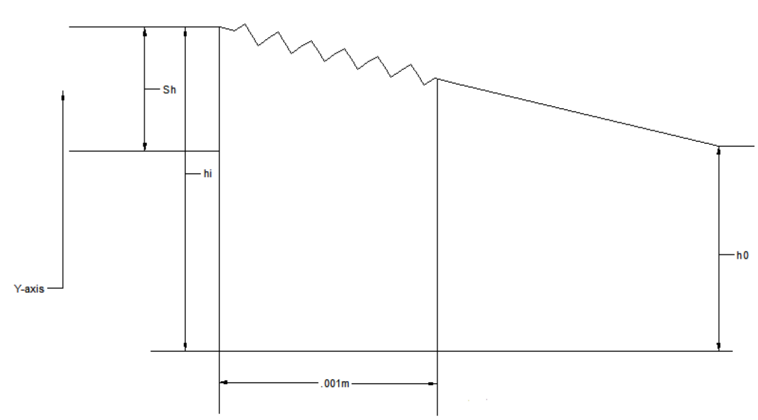
For smooth bearing, the lubricant film thickness created between two surfaces is that

$$h_{smooth} = (h1-((h1-h2)*(x))/l) \tag{2}$$

For grooved bearing, the lubricant film thickness expression is taken from Huynh (2005) and given as:

$$h_{grooved} = h_{smooth} + a\sin(2\pi x/\lambda) \tag{3}$$

Case C (corrugated at inlet zone)



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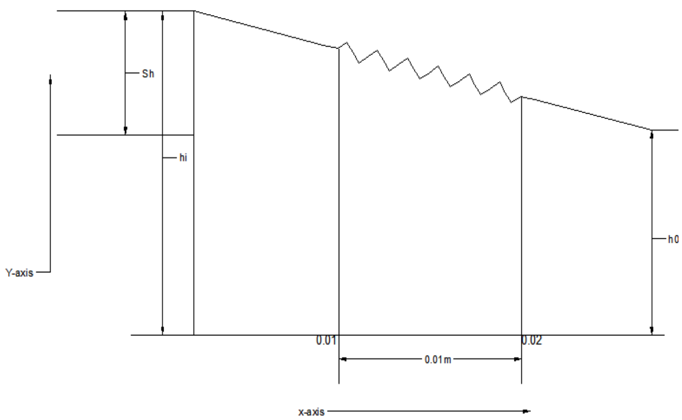
Case F (corrugated at mid zone)



(c) Case I (corrugated at exit zone)

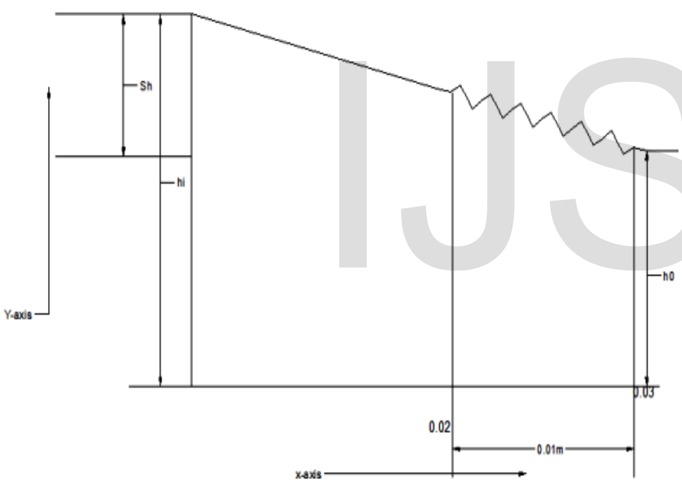


Figure 2.The schematic diagrams for corrugated slider bearing at three different Locations

$$\%age\ \text{var}\,iation\ in\ \max.pr. = \frac{(p_{\max})_{grooved} - (p_{\max})_{smooth}}{(p_{\max})_{smooth}} * 100$$

For the numerical solution of Reynolds equation, finite difference numerical method is used. The pressure is computed iteratively through Gauss-Seidal method while using over relaxation factor (1.7) for increasing the convergence speed. The flow chart regarding computational procedure is shown in figure 3.Process will be repeated till the specific accuracy is attained by convergence criteria for pressure as:

$$\sum\sum\left|\frac{\left(P_{i,k}\right)_I - \left(P_{i,k}\right)_{I-1}}{\left(P_{i,k}\right)_I}\right| < 10^{-06}$$

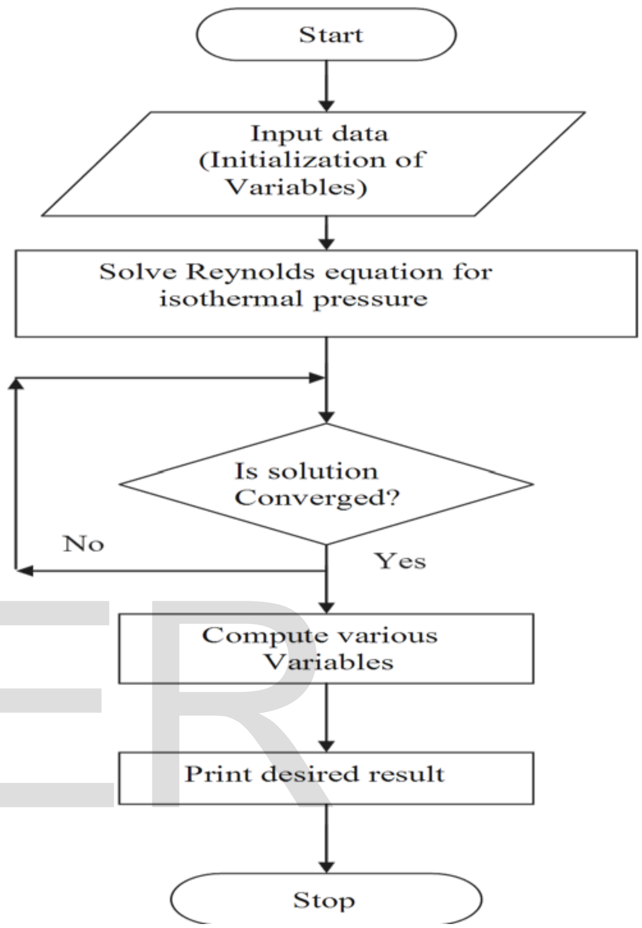


Figure 3; flow chart for computational procedure

3. Validation of Results

Validation of present results has been done with Huynh (2005) considering similar conditions. The results shown in Table 1 Have been found to be matching considerably well with the results available in literature.

Table 1 Comparison of the results for smooth and grooved infinite slider bearing.

Sr.No.	CASE	HUYNH (2005) RESULTS	PRESENT RESULTS
1	Smooth	$1.5376*10^5$	$1.5327*10^5$
2	Case C	$1.4068*10^5$	$1.4730*10^5$
3	Case F	$1.4780*10^5$	$1.6087*10^5$
4	Case I	$1.9517*10^5$	$1.7874*10^5$

4. Results and Discussion

The input parameters are taken from Huynh work to compute the pressure for smooth and grooved slider bearing. In the present work the slider bearings are divided into three cases which are short, infinite and finite bearings. For computing grooved slider bearings the partial grooving are placed on three different locations as shown in figure 2. And also indicated by case C, case F, case I similar to the Huynh work.

Table 2 Input values

Sr.No.	Name	Value	Unit
1	Velocity	3,6,9	m/s
2	Viscosity	0.029,0.03934,0.049	Pa.s
3	Maximum film thickness (h1)	0.00016	M
4	Minimum film thickness(h2)	0.00010	M
5	Length	0.03003	M
6	Width	0.3,0.03,0.003	M
7	Amplitude	0.00002727 for case C 0.00002364 for case F 0.00002 for case I	M
8	Nodes	Nx=244,Nz=20	
9	Wavelength	0.000667	M

4.1 Influence of speed

Figures 4, 5, 6 show the percentage variation in maximum pressure with respect to change in speed for infinite, finite, and short bearing. The percentage variation in maximum pressure is almost same with respect to change in speed for finite and short bearing as shown in figure 5 and 6. However, there is some variation in percentage seen in figure 4 for infinite slider bearing. The case I shows maximum increment in pressure as compared with case C and case F for both the bearings. The influence of partial sinusoidal grooving is more in the case I for short bearing.

From above points, it is absorbed that the effect of speed is approximately negligible for finite and short bearing and very less variation for infinite bearing. The main reason is that the lubricant viscosity is constant (isothermal analysis) in the present work.

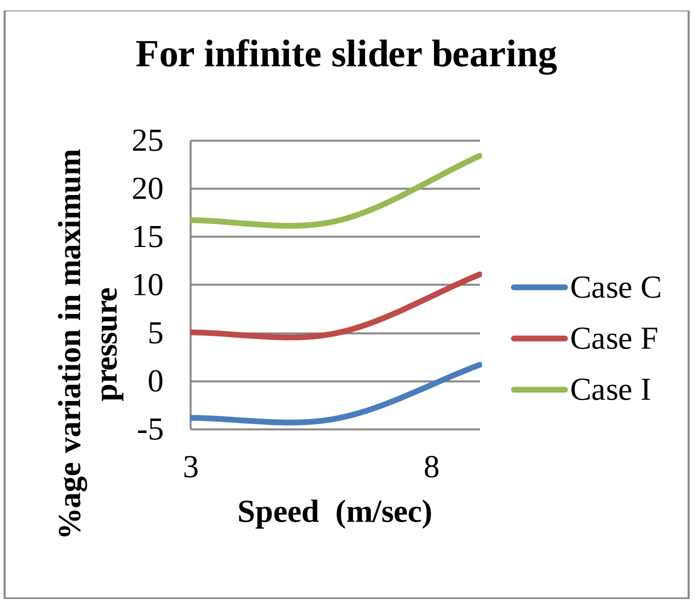


Figure 4: Percentage variation in maximum pressure with variation of speed for infinite slider bearing.

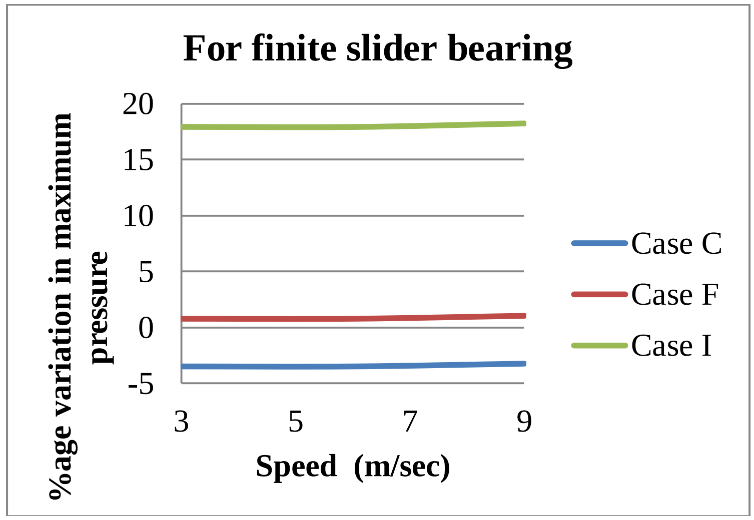


Figure 5: Percentage variation in maximum pressure with variation of speed for finite slider bearing.

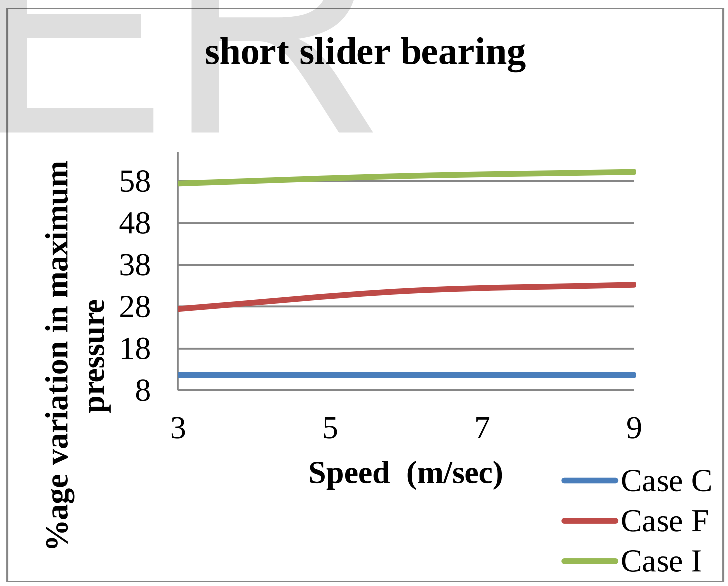


Figure 6: Percentage variation in maximum pressure with variation of speed for short slider bearing.

4.2 Influence of viscosity

Figures 7, 8, 9 show the percentage variation in maximum pressure with respect to change in viscosity for infinite, finite, and short bearing. The percentage variation in maximum pressure is almost same with respect to change in viscosity for finite and short bearing as shown in figure 8 and 9. However, there is some variation in percentage seen in figure 7 for infinite slider bearing. The case I shows maximum increment in pressure

as compared with case C and case F for both the bearings. The influence of partial sinusoidal grooving is more in the case I for short bearing.
From above points, it is observed that the effect of viscosity is approximately negligible for finite and short bearing and very less variation for infinite bearing. The main reason is that the present study is based on isothermal analysis.

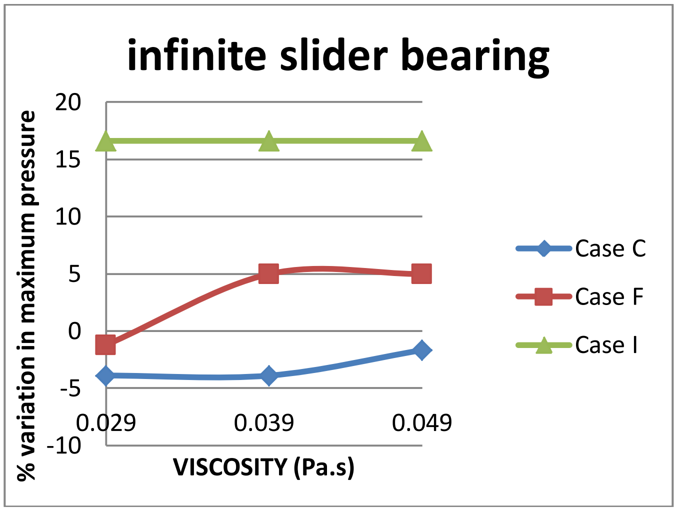


Figure 7: Percentage variation in maximum pressure with variation of viscosity for infinite slider bearing.

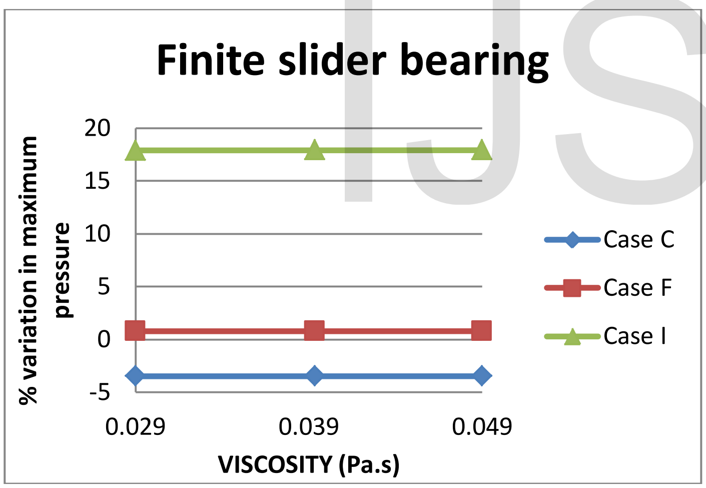


Figure 8: Percentage variation in maximum pressure with variation of viscosity for finite slider bearing.

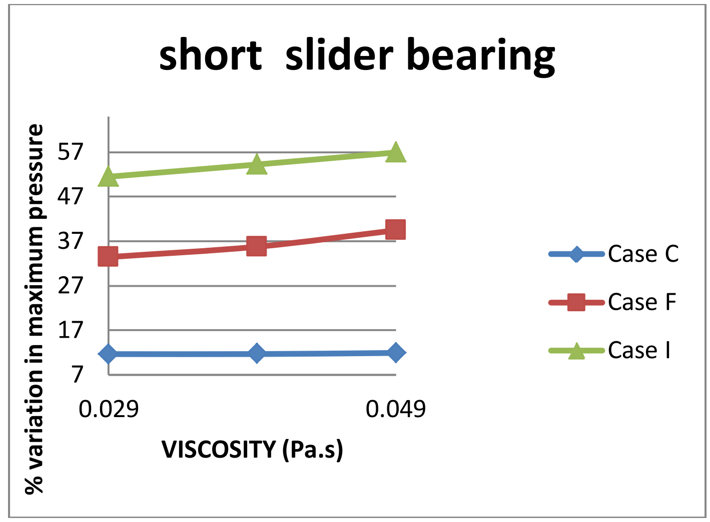


Figure 9: Percentage variation in maximum pressure with variation of viscosity for short slider bearing.

5. Conclusion

A finite difference numerical model is used to study the behaviour of partial sinusoidal grooving for different slider bearings. The influence of speed and viscosity is also investigated for smooth and grooved slider bearing. Conclusions can be drawn that negligible percentage variation with change in speed and viscosity is found for isothermal analysis. Partial short grooved bearing shows maximum percentage increment in pressure as compared with other slider bearings.

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