Effect of Longitudinal Surface Roughness on the Performance of Thermohydrodynamically Lubricated Slider Bearing - A Stochastic roughness Model

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Abstract

The influence of longitudinal surface roughness on the thermohydrodynamic lubrication of infinitely wide plane slider bearing has been investigated numerically by using stochastic roughness model. The effects of the roughness height and its orientation on the performance parameters of bearing have been studied. As the height of surface roughness increases, the value of film thickness ratio at which maximum load carrying capacity (isothermal) has been recorded reduces. With increase in roughness parameter, significant reduction in load carrying capacity (thermal) of bearing has been found due to increase in lubricant temperature.

1. Introduction

Since last two decades, it has been recognized that many hydrodynamically lubricated bearings operate under the thin film regime where surface roughness plays an important role in influencing the bearing performance parameters. A small change in the distribution of the heights, widths, and curvature of asperity peaks can have a noticeable effect on bearing performance. In the past, many researchers [1-4] have proposed various models for lubrication of rough surfaces, In the recent past, some investigators have widely used stochastic and average flow models for lubrication modeling. Burton [5] and Hargreaves [6] have analyzed the effect of asperity by incorporating deterministic models on the performance parameters of hydrodynamically lubricated slider bearings. Hargreaves compared his theoretical results with his experimental findings. He observed that the presence of transverse surface asperities enhances the load carrying capacity of bearing. Khonsari [7] has done a comprehensive review of works pertaining to thermal effects in hydrodynamic thrust slider bearing. His review even does not report any paper dealing with asperity effects along with thermal effects. In general it can be seen that after some runs of thrust slider bearing, pad develops surface roughness pattern that usually appear to be aligned with the direction of sliding. Therefore, the objective of this paper is to evaluate the effect of viscous heat dissipation in infinitely wide slider bearing having longitudinal roughness distribution on thrust pad. In this analysis, the rough surface has been assumed stationary and the moving surface is perfectly smooth. The coupled solutions of governing equations have been obtained by using an efficient numerical method developed by Elrod and Brewe [8].

2. Governing Equations

A schematic diagram for the slider bearing is shown in Fig. 1. The random distribution of longitudinal asperity has been considered based on stochastic roughness model given by Christensen [1].

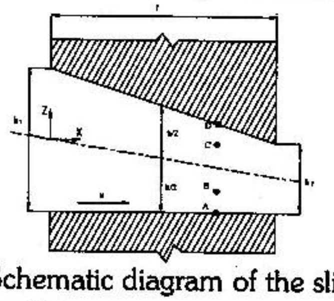

Fig. 1 Schematic diagram of the slider bearing

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Thus. Reynolds equation for longitudinal roughness is:

$$\frac{\partial}{\partial x}\left[\frac{d\overline{p}}{dx}E(h^3)\right] = 6\eta u\frac{\partial}{\partial x}E(h) \tag{1}$$

Stochastic component due to directional roughness is expressed as:

$$E(h^3\frac{\partial p}{\partial x}) = \frac{\partial \overline{p}}{\partial x}\left((\cos^2\theta)E(h^3) + \frac{(\sin^2\theta)}{E(1/h^3)}\right) \tag{2}$$

Where, "θ " is the angle between the roughness direction and the sliding direction.

The Reynolds equation for directional stochastic roughness is modified as:

$$\frac{\partial}{\partial x}\left[\frac{\partial \overline{p}}{\partial x}\left((\cos^2\theta)E(h^3) + \frac{(\sin^2\theta)}{E(1/h^3)}\right)\right] = 6\eta u\frac{\partial}{\partial x}E(h) \tag{3}$$

The concerned energy equation is:

$$\rho C_p u\frac{\partial T}{\partial x} = \frac{\partial}{\partial z}\left(k\frac{\partial T}{\partial z}\right) + \eta\left(\frac{\partial u}{\partial z}\right)^2 \tag{4}$$

Generalized Reynolds equation is obtained from mass continuity equation as:

$$\nabla\cdot\frac{\dot{m}}{\rho} = 0 \tag{5}$$

Load carrying capacity per unit width for bearing is computed as:

$$W = \int_0^l p\,dx \tag{6}$$

Roughness model has been incorporated in governing equations as per roughness concept and numerical methodology reported in ref. [1,8].

3. Computational Procedure

Computational methodology involves coupled solution of isothermal Reynolds Eq. (3), equations obtained from zeroth and first moment of energy Eq. (4), and generalized thermal Reynolds Eq. (5) by using appropriate boundary conditions. In computation, wherever reverse flow arises, upwind differencing has been resorted.

The solutions for governing partial differential equations have been made converged by satisfying the following criterion:

For pressure:

$$\frac{\left|(\Sigma p_i)_{n-1} - (\Sigma p_i)_n\right|}{\left|(\Sigma p_i)_n\right|} \le 0.001 \tag{7}$$

For temperature:

$$\frac{\left|(\Sigma T_i)_{n-1} - (\Sigma T_i)_n\right|}{\left|(\Sigma T_i)_n\right|} \leq 0.001 \tag{8}$$

Where, n represents number of iterations.

4. Results and Discussion

The input parameters used are listed in Table-1. Results have been presented for various performance parameters for infinite wide plane slider bearing at different roughness parameters (C=0.2 to 0.6), film thickness ratios (α=1.7 to 2.4), and sliding speeds (5 to 30 m/s).

Table 1 Input parameter

Inlet viscosity of the lubricant, Pa s	0.13885
Inlet temperature of Lubricant, K	311.11
Temperature viscosity coefficient of the lubricant, K^{-1}	0.045
Thermal diffusivity of the lubricant, m^2/s	7.306×10^{-8}
ρC_p of the Lubricant, J/m^3-K	1.7577×10^6
Minimum film thickness, µm	91.44
Length of bearing, m	0.18288

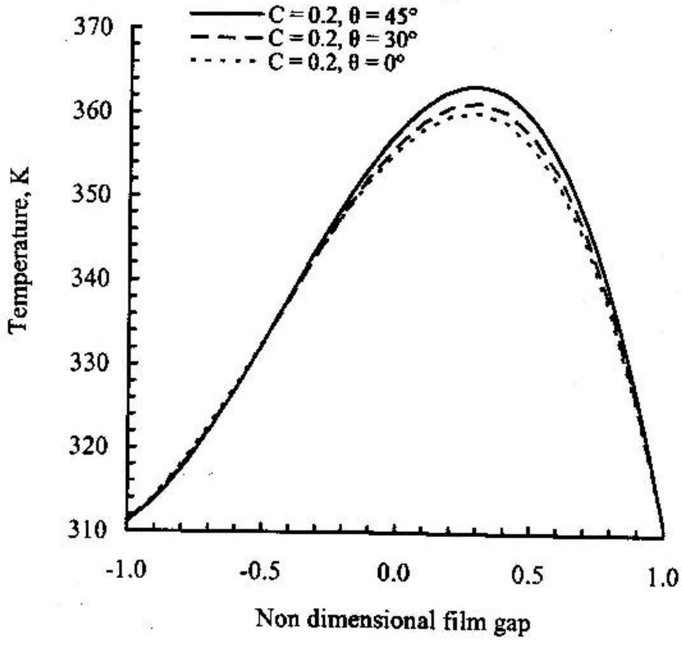

Fig 2 Temperature variation at different orientations of stochastic roughness [h_2= 91.44 µm, α =2.18, u =30m/s, X=1.0]

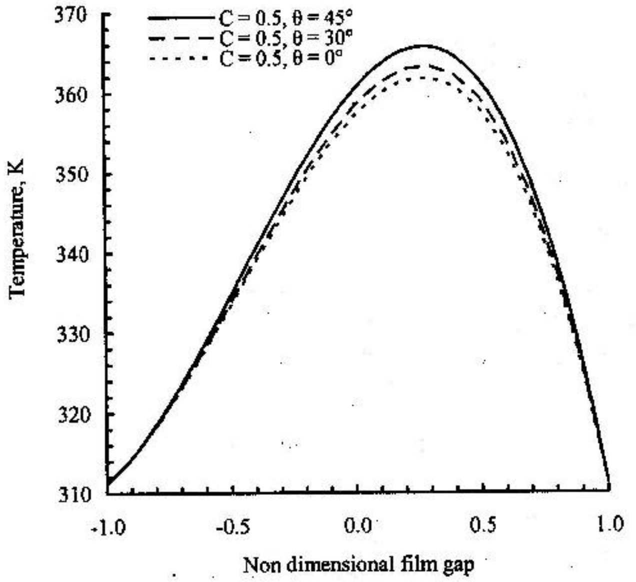

Fig. 3. Temperature variation at different orientations of roughness to sliding direction
[h_2=91.44 μm, α =2.18, C=0.5, u=30m/s, X=1.0]

Influence of roughness height and its orientation with respect to sliding direction on the temperature rise hs been shown in Fig. 2 and Fig. 3 for roughness parameters C=0.2 & C=0.5 respectively. A marginal increase in temperature (2°C) of film has been noticed at C=0.5 in comparison to C=0.2 for θ=0°. As orientation angle increase, rise in temperature increases. About 3°C and 5°C temperature rise has been observed when θ changes from 0° to 45° for C=0.2 and C=0.5 respectively. The probable reason for this phenomenon seems to be due to restriction offered to the flow by reducing the film thickness.

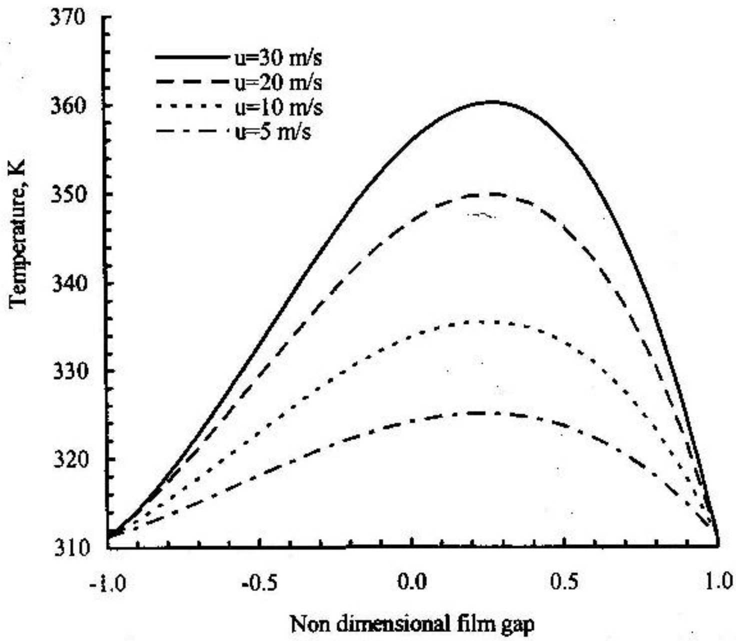

Fig. 4 Temperature variation across the sliding direction at different sliding speeds
[h_2=91.44 μm, α=2.18, C=0.2, θ=0°, u=30m/s, X=1.0]

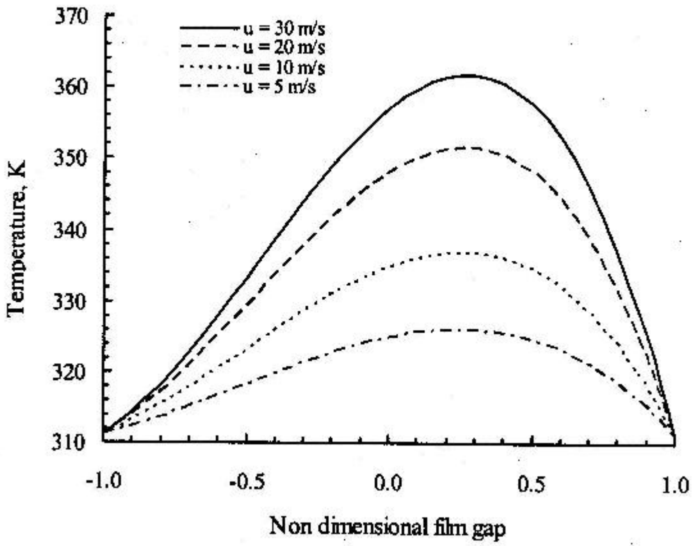

Fig 5 Temperature variation across the sliding direction at different sliding speeds
[h_2= 91.44 μm, α =2.18, C = 0.5, θ=0°, u=30m/s, X=1.0]

The temperature of the lubricating film increases with the increase in the runner/slider speeds. The temperature rise trend can be seen in Fig. 4 and Fig. 5 for C=0.2 and C=0.5 respectively. The computed temperatures in Fig.5 are marginally higher than the temperatures presented in Fig.4 for respective sliding velocities. The cause behind temperature rise is more viscous heat dissipation due reduced effective film thickness.

Presence of asperity on the bearing surface is bound to influence the load carrying capacity and other bearing performance parameters. Christensen [1] through his isothermal investigation found that the normalized load reduces for higher value of roughness parameters (refer [1] for detailed explanation). A similar trend has been observed for load carrying capacity in the isothermal and thermal cases as have been illustrated in Fig. 6 and Fig.7 respectively.

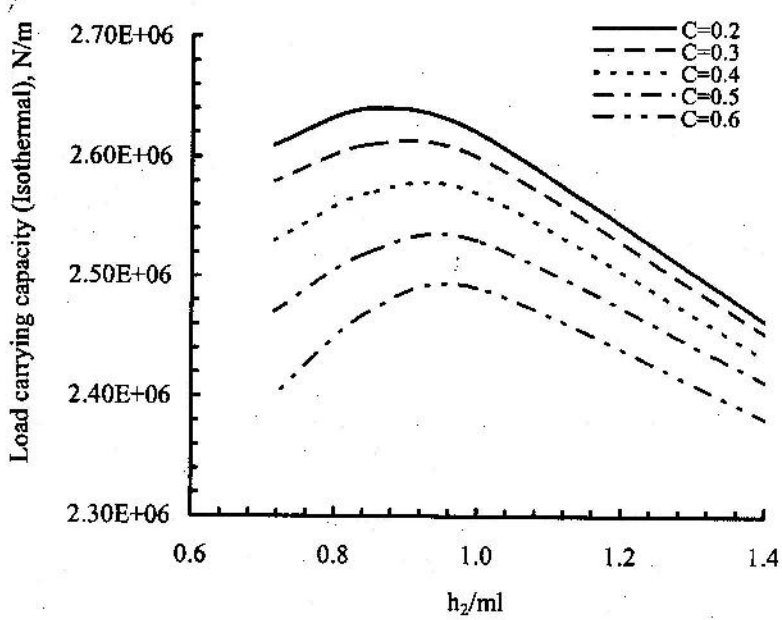

Fig. 6 Thermal load carrying capacity variation for stochastic roughness
[h_2 = 91.44 μm, u=30m/s, θ =0°]

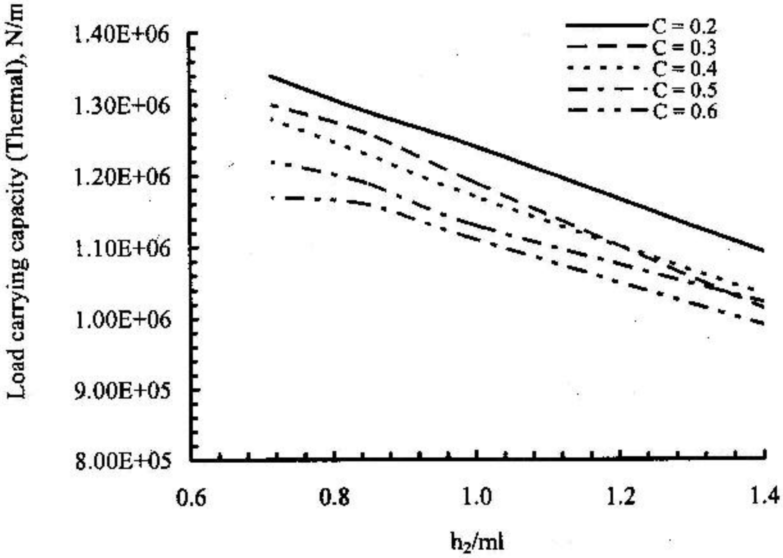

Fig 7 Thermal load carrying capacity variation for stochastic roughness
[h_2= 91.44 μm, u =30m/s, θ = 0⁰]

As the orientation of roughness changes from longitudinal direction to towards transverse direction, the load carrying capacity of bearing considerably increases this can be noticed from plot given in Fig.8. About 20 to 50 % rise in load carrying capacity of bearing found when θ changes from 0^0 to 45^0. The increase in the load carrying capacity with the incorporation of directional roughness happens due to the formation of lubricant barriers across the sliding direction. These barriers help in more pressure generation.

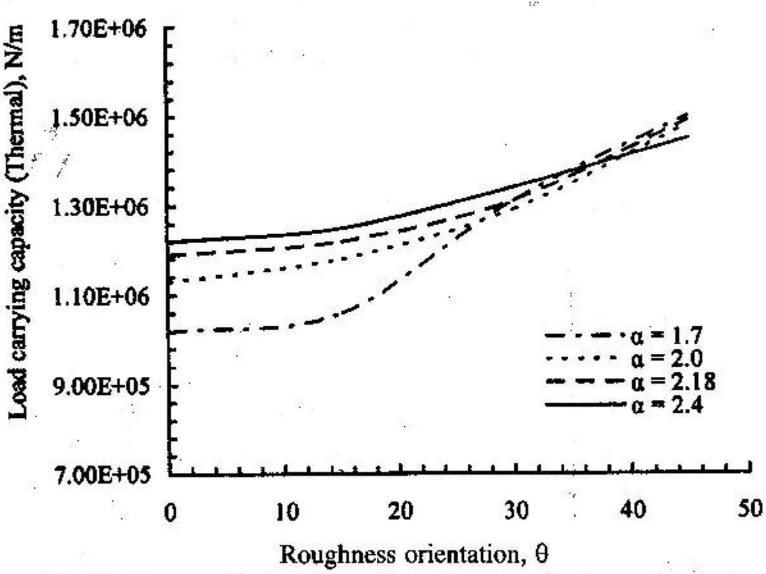

Fig 8 Influence of orientation of roughness on load carrying capacity
[h_2= 91.44 μm, u =30m/s]

5. Conclusions

Following conclusions are drawn from the present analysis.

1. Temperature rise of lubricating film increases with increase in longitudinal asperity height.
2. Isothermal and thermal load carrying capacities reduce with increase in roughness height.
3. As the surface roughness increases, the value of film thickness ratio at which maximum load carrying capacity (isothermal) has been recorded reduces.
4. The direction of roughness considerably makes an impact on bearing performance parameters under thermal considerations.

Nomenclature

c_d random roughness on bearing surface, m

C roughness parameter, c_d/ml

E( ) expectancy factor

m inclination of the slider bearing

T temperature, K

u sliding velocity, m/s

x x-coordinate, m

X non-dimensional x-coordinate, x/l

α film thickness ratio, h_1/h_2

h_1 film thickness at inlet, m

h_2 film thickness at exit, m

l length of the slider, m

η lubricant viscosity, Pa s

η_0 lubricant viscosity at p=0 and T=T_0, Pa s

$$\frac{\partial \overline{p}}{\partial x} = E\left(\frac{\partial p}{\partial x}\right)$$

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