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A Numerical Simulation of Temperature Distribution and Power Loss of Slider Bearings Lubricated With Fluids Having Constant Viscosity

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Abstract

A numerical procedure is developed for the analysis of temperature distribution and power loss in inclined and exponentially shaped slider bearings lubricated with constant viscosity lubricants. The Reynolds equation is solved using the finite element formulation. The stream wise pressure gradient, shear stresses and flow rate obtained from post processing of the finite element solution of the Reynolds equation act as inputs when the energy equation is discretized to obtain the stream wise temperature variation. The numerical model is applied to the comparative analysis of the temperature profiles and power losses that characterize the lubricant flow in the slider configurations under focus. The results are presented in graphical form.

Keywords: Energy equation, slider, finite element, power loss, temperature, Reynolds

Nomenclature

C_p	Specific heat (J/Kg^0C)
ρ	Density (Kg/m^3)
ΔT	Temperature rise (^0C)
a	Film thickness ratio
h_2	Film thickness at entry of bearing
h_1	Film thickness at bearing exit
$h(x)$	Oil film thickness
p_{inlet}	Oil supply pressure (Kpa)
p_{exit}	Pressure at exit of bearing (Kpa)
Ω	Power loss
T_{in}	Inlet oil film temperature
T_{out}	Outlet oil film temperature

1. Introduction

With the improvement in the efficiency of machineries, the operating conditions of slider bearings are increasing in speed and load. Therefore, accurate prediction of bearing operating temperature has become one of the most important technologies in designing machines [1]. The prediction of operating temperature of slider bearings is achieved through thermohydrodynamic analysis. Typically, the thermohydrodynamic analysis of slider bearings reduces to solving the generalized Reynolds and the energy equations governing the fluid film behavior.

A number of researchers have approached the thermohydrodynamic lubrication problem using different numerical methods. Pal et al [2] considered an iterative technique for theoretical study of thermal effects in finite slider bearings. Ogata

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[1] solved the Reynolds equation using central difference method and discretized the energy equation using power law to predict the temperature distribution in step bearings. Schomack [3] used the pseudospectral method to solve the thermodynamic lubrication problem for a slider bearing.

Lonescu and Mihai [4] proposed a quantitative calculation of slider bearings using the analytical solutions of the energy equation and of the Reynolds differential equation and computed some hydrodynamic parameters. Sharma and Pandey [5] reported the thermal analysis of hydrodynamically lubricated cycloidal pad thrust bearing using an efficient numerical method based on Lobatto quadrature technique and computed performance parameters for various operating inputs. Jang and Khonsari [6] developed approximate expressions for rapid estimation of the maximum temperature and the effective temperature as well as the performance parameters of hydrodynamic bearings. Jang and Khonsari [7] carried out a theoretical thermohydrodynamic analysis of finite slider bearings with the Bingham rheological model and presented a set of parametric studies of the Bingham model.

It can be observed from the cited literature above that the finite element method has not been applied to the computation of temperature distribution and power loss in slider bearings lubricated with constant viscosity lubricants. The finite element method is a very useful numerical tool, but its applicability is limited as a result of its mathematical complexity. In this paper, a comparative study of the temperature distribution and power loss in inclined and exponentially shaped slider bearing is presented. The information from this paper will be useful to engineers who are involved in bearing design.

Mathematical Model

Slider bearing problem involves the solution of the Reynolds and energy equations. The slider bearing problem reduces to finding the pressure and temperature profiles in the oil film of the bearing. The pressure distribution is obtained from a solution of the Reynolds' equation, while the latter is obtained from solution of the energy equation. The schematic drawings of the two slider bearings under consideration are shown in figures 1 and 2 respectively.

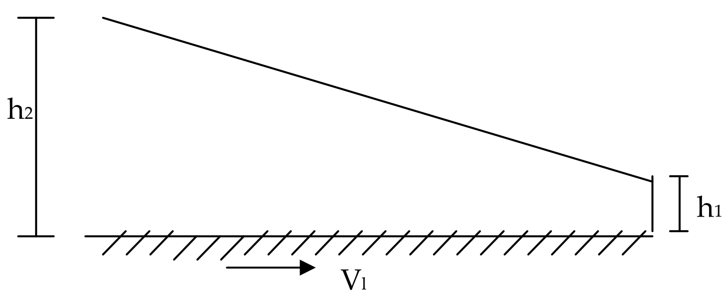

Figure 1: Schematic drawing of inclined slider bearing

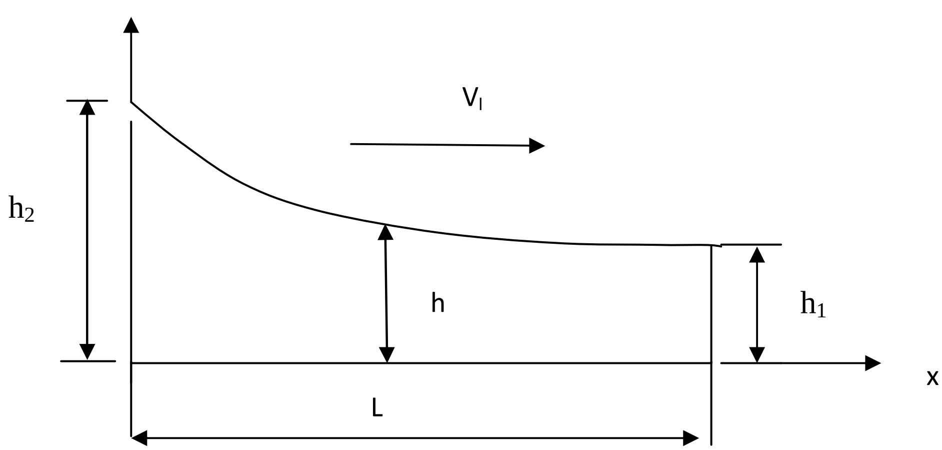

Figure 2: Schematic drawing of exponential film shape slider bearing

The oil film profile for the exponential slider is described by equation (1) and for the inclined slider bearing the oil film profile is described by equation (2)

$$h(x) = h_2 e^{-x\ln(\frac{a}{L})} \tag{1}$$

$$h(x) = h_2 - (h_2 - h_1)\frac{x}{L} \tag{2}$$

The film thickness ratio (a) is given by $\frac{h_2}{h_1}$ for both bearing configurations. Reynolds equation for a laminar, incompressible one dimensional flow in a slider bearing is given by equation (3). [8]

$$\frac{d}{dx}\left(\frac{h^3}{12\mu}\frac{dp}{dx}\right) = \left(\frac{V_u + V_l}{2}\right)\frac{dh}{dx} \tag{3}$$

In equation (3), V_u V_u and h stand for the upper and lower surface velocities and film thickness respectively. The temperature distribution in the bearing is given by equation (4).

$$\rho C_p q_x \frac{dT}{dx} = V_u\tau_u + V_l\tau_l - q_x\frac{dp}{dx} \tag{4}$$

In equation (4), q_x is the volumetric flow rate defined by equation (5), τ_u and τ_l are the shear stresses in the upper and lower surfaces of the slider given by equations (6a) and (6b) respectively.

$$q_x = -\frac{h^3}{12\mu}\frac{dp}{dx} + \frac{(V_u - V_l)h}{2} \tag{5}$$

$$\tau_l = -\frac{\mu V_l}{h} - \frac{h}{2}\frac{dp}{dx} \tag{6a}$$

$$\tau_l = -\frac{\mu V_l}{h} + \frac{h}{2}\frac{dp}{dx} \tag{6b}$$

The power loss of the slider bearing is obtained by using equation (7)

$$\Omega = q_x C_p \Delta T \tag{7}$$

Computational Procedure

The pressure distribution is obtained by solving Reynolds equation. Galerkin formulation was utilized in order to apply the finite element method to solve Reynolds equation. The residual of equation (3) is shown in equation (8)

$$R(x,p) = \frac{d}{dx}\left(\frac{h^3}{12\mu}\frac{dp}{dx}\right) - \left(\frac{V_u + V_l}{2}\right)\frac{dh}{dx} \tag{8}$$

Multiplying equation (8) by a weight function w_i and integrating over a typical element with end nodes x_1 and x_2, we obtain equation (9)

$$\int_{x_1}^{x_2} w_i R(x,p)\,dx = 0 \qquad i = 1,2 \tag{9}$$

Substituting equation (8) into equation (9) and integrating the first term of the resulting equation, equation (10) is obtained

$$\int_{x_1}^{x_2} \frac{dw_i}{dx}\frac{h^3}{12\mu}\frac{dp}{dx} + \left[w_i\frac{h^3}{12\mu}\frac{dp}{dx}\right]_{x_1}^{x_2} - \int_{x_1}^{x_2}\left(\frac{V_u + V_l}{2}\right)\frac{dh}{dx}dx \quad = 0 \tag{10}$$

Now we assume a trial solution for the nodal degree of freedom of the form of equation (11)

$$p = \sum_{j=1}^{2} p_j \varphi_j(x) \tag{11}$$

Obtaining $\left(dp/dx\right)$ from equation (11) and substituting into equation (10) with the weight functions (w_i) set identical to the trial functions, φ_j we obtain the Galerkin finite element model for the slider problem.

$$\sum_{j=1}^{n}\left[\int_e \frac{d\varphi_j^e}{dx}\frac{h^3}{12}\frac{d\varphi_i^e}{dx}\right]p_j+\left[\frac{h^3}{12}\frac{dp}{dx}\varphi_i^e\right]^e-\int_e\left(\frac{V_u+V_l}{2}\right)\frac{dh}{dx}\varphi_i^e dx=0 \quad (12)$$

The spatial domain of the bearing is divided into linear finite elements. Stiffness equations are obtained for all the elements in the mesh, they are then assembled by enforcing continuity of the nodal pressures, after which the pressure boundary conditions, namely, inlet pumping pressure p_{inlet} and the outlet pressure p_{outlet} are imposed on the assembled system of equation. The resulting system of equation is solved using Gauss Seidel iterative method with a convergence criterion of 10^{-4}
The results derived from the solution of the Reynolds equation which includes, the stream wise pressure gradient $\left(dp/dx\right)$, the shear stress in the upper and lower bearing surfaces computed using equations (6a) and (6b), act as inputs when the stream wise temperature distribution is under consideration. To determine the spatial temperature distribution, equation (4) is discretized using a centered finite difference scheme and a thermal relation between two consecutive nodes i and $i+1$ is obtained as shown in equation (13)

$$T_{i+1}=T_i+\left[\frac{V_u\tau_u+V_l\tau_l}{\rho C_p q_x}\Delta x-\frac{q_x\left(p_{i+1}-p_i\right)}{\rho C_p q_x}\right] \quad (13)$$

The temperature rise of the lubricant is obtained by using equation (14)

$$\Delta T=T_{out}-T_{in}$$

THEORITICAL MODEL

Two types of slider bearings are considered in the present study. The first has its oil film narrowing in linear style similar to that considered separately by [9] and [10] whereas the second has an exponential film profile, similar to that considered by [11]. The inlet height is set at 1mm and the exit height is set in the range 0.50 – 0.125mm. These values are within the commonly used bearing pad heights in the literature. The heights used for the inlet and the outlets result in aspect ratios in the range 2.0 – 8. The aspect ratio employed is comparable with those of [12] and [13] where the corresponding lower and upper limits of their aspect ratio ranges was 6 and 10 respectively. The lubricating oil used for the analysis is SAE 30 which has a density, $\rho = 855\text{Kg/m}^3$ a specific heat; $C_p = 2090\text{J/Kg}^0\text{C}$, viscosity $\mu = 0.04305 Ns/m^2$, with oil feed temperature equal to 25^0 C. which is similar to that used by Ashour [14]. The pumping pressure (p_{inlet}) of the lubricating oil at the inlet of the bearing is set within the range $101-501\text{Kpa}$ while the exit pressure (p_{exit}) is set atmospheric pressure. $(100Kpa)$
The upper bearing surface is kept stationary $(V_u = 0)$ as in the study of Honchi et al [15]. For the present analysis, the velocity of the lower surface (V_l) is set at $100ms^{-1}$. The bearing length used in the analysis is equal to $10mm$ which is between the choice of [15] which equals 1.25mm and [16] which equals 14.5mm

Numerical Results and Discussion

Figs. 3 and 4 shows the oil film temperature profiles for the two slider configurations with two instances of p_{inlet} namely, 101Kpa and 501Kpa respectively.

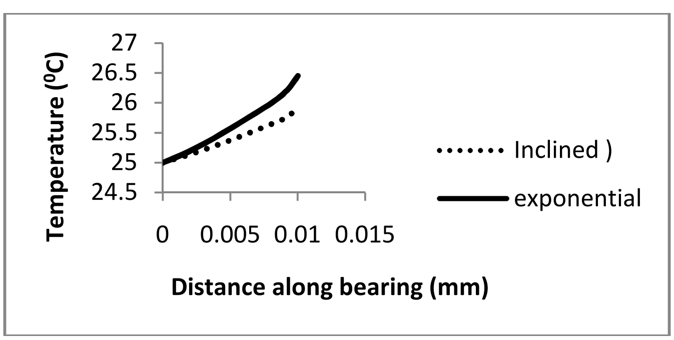

Figure 3: Temperature profile for inclined and exponential slider bearings for $p_{inlet} = 101\text{Kpa}$

The bearing length is equal to 10mm, the maximum film thickness, h_2, is set equal to 1mm and minimum film thickness,

h_1, is set equal to $0.25mm$ The spatial domain of the bearing is uniformly discretized with 128 linear finite elements. It was found that, in general, the temperature profile decreases with increase in pumping pressure for both slider configurations as depicted in figures 3 and 4 respectively. This finding is consistent with those obtained by Brito et al [17], who determined experimentally that an increase in oil feeding pressure yields a decrease in oil outlet temperature. However, under the same bearing structural and rheological parameters, the temperature buildup in the exponential slider bearing is greater than that in the inclined slider bearing for a fixed inlet pumping pressure. Specifically, a temperature rise of $1.45^0 C$ is noted for an exponential slider bearing compared to $0.95^0 C$ for the inclined slider bearing at a pumping pressure $p_{inlet} = 101 Kpa$ .Numerical computations reveal that a decrease in the film thickness ratio brings about a decrease in lubricant temperature rise as depicted in Fig. 5. A decrease in the film thickness ratio from 8 to 4, obtained by setting the inlet film thickness, h_2, at $1mm$, and increasing the outlet film thickness, h_1, from $0.125mm$ to $0.25mm$ yields approximately a reduction in lubricant temperature rise by a factor of 2.78, for an inclined slider bearing in contrast with an exponential slider where a reduction in temperature rise by a factor of 3.38 was obtained using a slider velocity equal to $100ms^{-1}$, lubricant viscosity $\mu = 0.04305 Ns/m^2$ and an inlet pumping pressure, $p_{inlet} = 101 Kpa$. At $p_{inlet} = 501 Kpa$ however, the reduction in temperature rise with increase in film thickness ratio is lower for inclined slider bearing, but appears to be higher for the exponential slider bearing compared to those at $p_{inlet} = 101 Kpa$. A reduction in temperature rise of 2.62 was obtained for an inclined slider compared to a reduction of 3.44 for an exponential slider. These results resembles those obtained by Gardner [18] who carried out measurements on a tilting pad thrust bearing and reported reduced temperature when oil film thickness was reduced. The relation between film thickness and temperature rise established in the current study is also corroborated by studies carried out by Wang and Zhu [19] and Storteig and White [20]. The earlier determined lower temperature rise at low eccentricity cases and recorded elevated temperature rise with increase in eccentricity. Storteig and White [20].recorded elevated temperature with higher cavity heights.

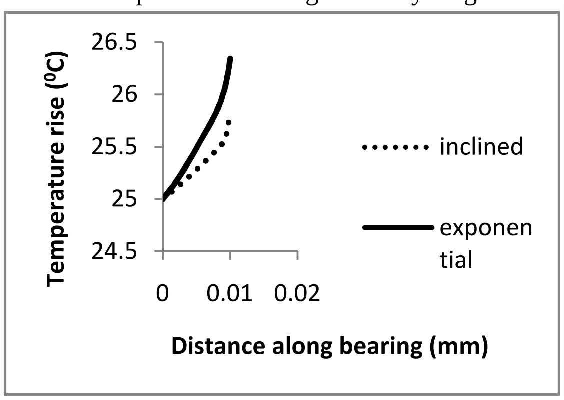

Figure 4: Temperature profile for inclined and exponential slider bearings for $p_{inlet} = 101 Kpa$

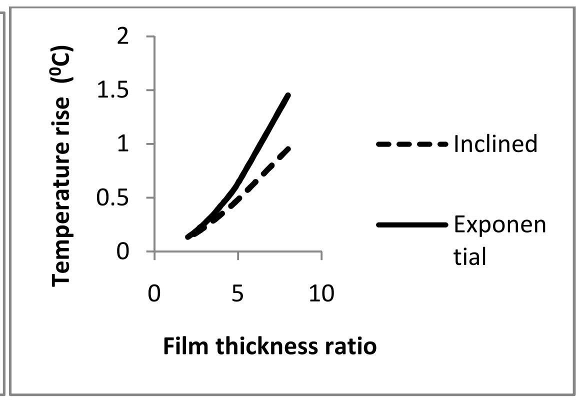

Figure 5: Temperature rise as a function of film thickness ratio for $p_{inlet} = 101 Kpa$

Fig. 6 displays the relation between power loss and inlet pumping pressure for the slider configurations with a film thickness ratio equal to 8. The plot shows that the power loss increases with inlet pumping pressure (p_{inlet})

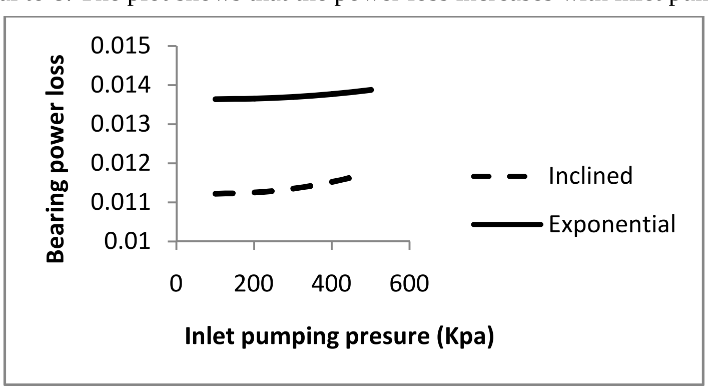

Figure 6: Bearing power loss as a function of inlet pumping pressure $(r = 8)$

This finding is consistent with those of Ozalp and Ozalp [21] who reported power loss augmentation with higher inlet pressure cases for slider bearings with micro machined wavy cavity. The increase in bearing power losses with increase in inlet pumping pressures is as a result of increased oil flow rate. Computations put forth that the bearing power loss is more significant in the inclined slider bearing than in the exponential slider under similar structural and rheological parameters. Numerical experimentation also demonstrates that with decrease in the film thickness ratio, lower bearing power losses result for the same inlet pumping pressure. Specifically, a decrease in the film thickness ratio from 8 to 4 result in a decrease in the bearing power loss by a factor equal to 1.6 for the model inclined slider bearing with inlet pumping pressure equal to 101Kpa and a reduction in bearing power loss by a factor equal to 1.32 for an inlet pumping pressure equal to 501Kpa.

Conclusion

In this paper, a comparative study of temperature distribution and power loss of inclined and exponentially shaped slider bearings has been presented. Extensive numerical computations have been carried out to show the effect of the bearing structural and rheological parameters on the performance of the bearings considered. It has been shown that the magnitude of power loss which depends on the volumetric flow rate is more significant in inclined slider bearing compared to the exponentially shaped slider. This conclusion is in tandem with results reported in literature by Lin et al [22] who reported lower volumetric flow rates for exponentially shaped slider lubricated with both Newtonian and non Newtonian couple stress lubricants. The temperature rise in the lubricant has also been shown to be higher for exponentially shaped slider bearing. This finding is consistent with those obtained by Lin et al [22] who carried out a comparative analysis of inclined and exponentially shaped slider lubricated with both Newtonian and non Newtonian couple stress fluids and reported higher lubricant temperature rise for the exponentially shaped slider.

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