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THREE-DIMENSIONAL THERMAL FIELD IN SLIDER BEARINGS

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ABSTRACT

A three dimensional thermo-hydrodynamic lubrication model which couples the Reynolds and energy equations is developed. The model uses the streamline upwind Petrov-Galerkin (SUPG) method to solve the non-symmetric stiffness matrix that results from the convective-dominated flow. Model results indicate that the peak temperature is not on the mid-plane surface, a fact that cannot be predicted with two dimensional models. This position shifts towards the mid-plane as the width to length ratio is reduced from ten to one (square slider) as well as when pressure boundary conditions are altered in such a way that the inlet/outlet pressure is higher than the side pressure. The square slider has a peak temperature 4°K less than the wider slider. This is due to the higher side flow in the square slider.

INTRODUCTION

Slider bearings are widely used in applications such as mechanical seals, plain collar thrust bearings, machine tool guides, and piston rings. They have good load-carrying capacity, excellent stability and durability. Investigation of the thermal effect leads to a better understanding of the load-carrying capacity.

Recent studies of thermo-hydrodynamic lubrication (THDL) slider models include the work of Rodkiewicz [1], Schumack [2], and Kumar et al [3,4]. These models mainly focus on two dimensional aspects. A comprehensive three dimensional model which couples the Reynolds and energy equations using the streamline upwind Petrov-Galerkin (SUPG) finite element method in the program Sepran [5] is developed. The non-dimensional form of the Reynolds equation is

$$\frac{\partial}{\partial x^+}\left(\frac{{h^+}^3}{\eta^+}\frac{\partial p^+}{\partial x^+}\right)+\frac{\partial}{\partial y^+}\left(\frac{{h^+}^3}{\eta^+}\frac{\partial p^+}{\partial y^+}\right)=6\frac{\partial h^+}{\partial x^+}$$

where a '+' indicates a non-dimensional quantity. The non-dimensional form of the energy equation is

$$\rho^+\left(u^+\frac{\partial T^+}{\partial x^+}+v^+\frac{\partial T^+}{\partial y^+}+w^+\frac{\partial T^+}{\partial z^+}\right)=\frac{1}{P_e}\left(\frac{\partial^2 T^+}{\partial {y^+}^2}+\frac{\partial^2 T}{\partial {z^+}^2}\right)$$
$$+\frac{P_r E_c}{P_e}\left(u^+\frac{\partial p^+}{\partial x^+}+v^+\frac{\partial p^+}{\partial y^+}\right)$$
$$+\frac{P_r E_c}{P_e}\eta^+\left(\left(\frac{\partial u^+}{\partial z^+}\right)^2+\left(\frac{\partial v^+}{\partial z^+}\right)^2\right)$$

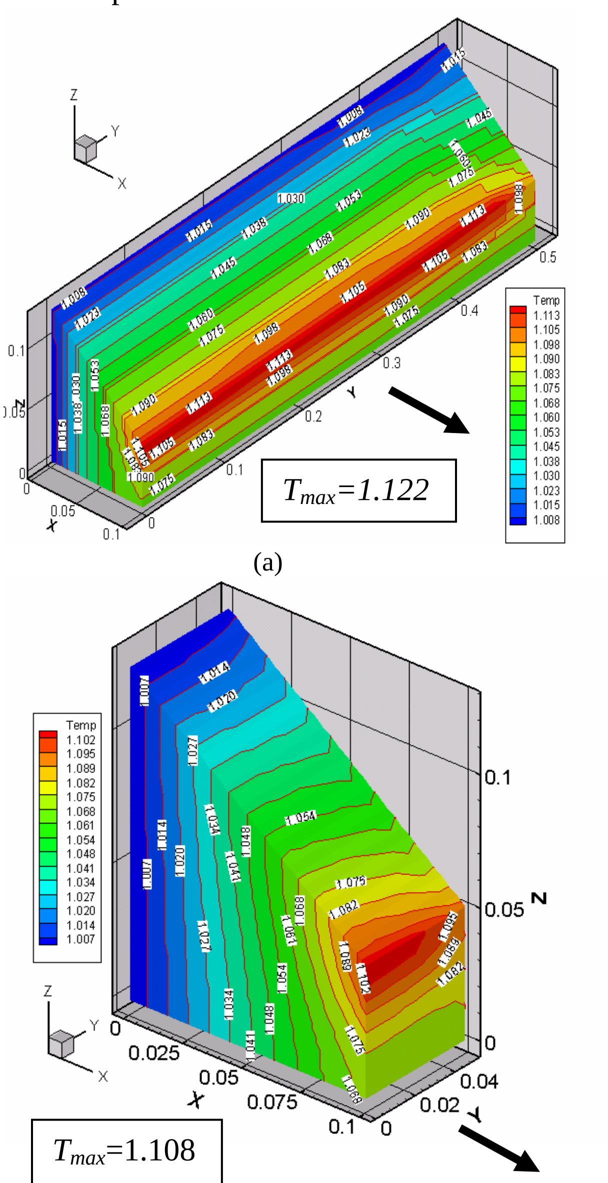
where x^+, y^+, and z^+ are the non-dimensional axes respectively. The non-dimensional form allows for comparison with data in the literature. Half the slider is modeled due to symmetry. The solution procedure is iterative. Current values of viscosity η^+ are used to compute the pressure field in the Reynolds equation followed by pressure gradients $\partial p^+/\partial x^+$ and $\partial p^+/\partial y^+$ which are used in the momentum equations to determine velocity fields u and v. The continuity equation is then solved for w. The temperature field is determined from the energy equation with the new velocity field followed by determination of the viscosity. The procedure repeats until equilibrium is achieved.

MODEL PREDICTIONS

The test case presented by Lebeck [6] with input values of: $\eta=0.0174$Pa.s, $U=20$m/s, $h_i=5x10^{-5}$m, $c_p=1926$J/kgK,

$L=0.1$m, ambient temperature $T_a=310$K, $\rho=897.1$ kg/m^3, and $k_f=0.132$ W/mK.

Figure 1a shows the temperature field for a width to length (W/L) ratio of ten. The peak temperature position is not on the mid-plane surface.



(a) (b)

Figure 1 Isotherms for slider with W/L=10 (a) and square slider (b). The arrow shows the sliding direction. The position of the peak temperature shifts towards the mid-plane surface as W/L is reduced from ten to one.

The side pressure has the effect of shifting the peak temperature point towards the mid-plane surface if it is reduced to zero. These results are important as they show where 'hot spots' will most likely occur.

NOMENCLATURE

c_p specific heat of lubricant
E_c Eckert number = $U^2/(c_p\Delta T)$
h film thickness
h_i film thickness at bearing inlet
k_f thermal conductivity of the fluid
L Bearing length (x-dimension)
P_e Pecklet number = $\rho_a c_p U h_o^2/(k_f L)$
P_r Prandtl number = $c_p\eta/k_f$
p_i Inlet pressure
P_r Prandtl number = $c_p\eta/k_f$
u fluid velocity component in x-axis
U Velocity of the runner
v velocity component in y-direction
w velocity component in z-axis

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