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The accuracy of the compressible Reynolds equation for predicting the local pressure in gas-lubricated textured parallel slider bearings

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ABSTRACT

The validity of the compressible Reynolds equation to predict the local pressure in a gas-lubricated, textured parallel slider bearing is investigated. The local bearing pressure is numerically simulated using the Reynolds equation and the Navier–Stokes equations for different texture geometries and operating conditions. The respective results are compared and the simplifying assumptions inherent in the application of the Reynolds equation are quantitatively evaluated. The deviation between the local bearing pressure obtained with the Reynolds equation and the Navier–Stokes equations increases with increasing texture aspect ratio, because a significant cross-film pressure gradient and a large velocity gradient in the sliding direction develop in the lubricant film. Inertia is found to be negligible throughout this study.

1. Introduction

Surface texturing is used to enhance the load-carrying capacity of parallel slider bearings and to reduce friction and wear [1,2,3]. While groove-texture, and in particular spiral-grooved bearings [4] and herringbone-grooved bearings [5] have been used for several decades, surface microtexturing is a more recent development. Microtexture is commonly implemented as a dense array of micro-sized concave features ("dimples") fabricated using, e.g., laser surface texturing (LST) [6,7]. Reduced friction and increased load-carrying capacity have been reported for a spectrum of practical applications including journal bearings [8], thrust bearings [9,10], piston rings [11], mechanical seals [12,13], gas seals [14], and magnetic tape drive systems [15].

The local pressure distribution and the load-carrying capacity of a textured slider bearing are typically computed using the Reynolds equation. However, the presence of the surface texture potentially causes some of the key assumptions of the Reynolds equation to break down. Recently, a number of studies have discussed the validity of these assumptions to simulate hydrodynamic pressure in textured bearings with an incompressible lubricant, as a function of surface texture geometry and/or surface roughness and operating conditions [16–23]. Numerical solutions of the Navier–Stokes equations or Stokes equations are typically used to investigate the validity of the assumptions of the Reynolds equation. Hu and Leutheusser [16] studied parallel slider bearings with sinusoidal grooves on one of the bearing surfaces. For large Reynolds numbers they suggested that inertia is important when defining the limits of applicability of the Reynolds equation. Others have demonstrated that the Reynolds equation inaccurately predicts the pressure when the film thickness is on the same order of magnitude as the surface roughness feature wavelength (Stokes roughness) [17–19]. Arghir et al. [17] found that inertia becomes increasingly important when calculating the hydrodynamic pressure for a large Reynolds number flow. They concluded that this effect cannot be accurately simulated with the simplified Reynolds equation. In addition, van Odyck and Venner [18] demonstrated by comparing the solutions of the Stokes equations and the Reynolds equation that even without considering inertia, the results of the Reynolds equation display a significant difference with a more complete model for the case of Stokes roughness. Sahlin et al. [20] and Cupillard et al. [21] investigated inertia effects in infinitely long parallel sliders textured with two-dimensional dimples, by comparing results of the Navier–Stokes equations and the Stokes equations. The dimple depth and the film thickness were chosen to be on the same order of magnitude. Both studies confirm that inertia affects bearing load-carrying capacity. Similarly, de Kraker et al. [22] demonstrated that for simulating mixed lubrication in a textured bearing, the Reynolds equation with a cavitation model is appropriate when the film thickness is much smaller than the dimple depth. When the film thickness is larger than the dimple depth, inertia dominates and the Navier–Stokes equations must be used. Dobrica and Fillon [23] studied the effect of inertia as a function of the texture aspect ratio and concluded that the solutions of the Reynolds equation and the Navier–Stokes equations match well when the texture aspect ratio and the Reynolds

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Nomenclature

c	minimum spacing between the parallel bearing surfaces
c^*	reference minimum spacing, minimum spacing between bearing surfaces in the case of $\varepsilon=0.1$, $\delta=0.01$
$H(X,Y)$	non-dimensional local spacing $H(X,Y)=h(x,y)/c$
$h(x,y)$	local spacing between the bearing surfaces
h_p	dimple depth
L	total length of the textured bearing consisting of ten unit cells
$P(X,Y)$	non-dimensional local pressure, $P(X,Y)=p(x,y)/p_0$
$P^{NS}(X,Y)$	non-dimensional differential bearing pressure solution from the Navier–Stokes equations, $P^{NS}=(p^{NS}-p^{NS}_{avg})/p_0$
$P^{RE}(X,Y)$	non-dimensional differential bearing pressure solution from the Reynolds equation, $P^{RE}=(p^{RE}-p^{RE}_{avg})/p_0$
$\Delta P_r(X,Y)$	non-dimensional local pressure difference, $\Delta P_r=\|\|P^{RE}-P^{NS}\|\|_2/\|\|P^{NS}\|\|_2$
p	local pressure
p_0	atmospheric pressure
p_{avg}	average bearing pressure
$p^{NS}(x,y)$	local bearing pressure solution from the Navier-Stokes equations
$p^{RE}(x,y)$	local bearing pressure solution from the Reynolds equation
p^{NS}_{avg}	average bearing pressure from the Navier–Stokes equations
p^{RE}_{avg}	average bearing pressure from the Reynolds equation
R	velocity gradient ratio, $R=(\partial u/\partial x)/(\partial u/\partial z)$
R_{avg}	average velocity gradient ratio calculated for all finite volume cells
Re_{loc}	average local Reynolds number calculated for all finite volume cells
r_1	half-length of the square unit cell
r_p	dimple characteristic radius
S_p	texture density
U	relative sliding velocity
u	velocity in the x-direction
V	volume of the clearance between the bearing surfaces
V_l	volume of a tetrahedral cell l
v	velocity in the y-direction
$\mathbf{v}$	Velocity vector, $\mathbf{v}=[u, v, w]^T$
w	velocity in the z-direction
X,Y,Z	non-dimensional Cartesian coordinates, $X=x/r_p$, $Y=y/r_p$, $Z=z/c$
x,y,z	Cartesian coordinates
δ	non-dimensional minimum spacing between the bearing surfaces, $\delta=c/2r_p$
ε	texture aspect ratio, $\varepsilon=h_p/2r_p$
Λ	bearing number, $\Lambda=6\mu U r_p/(p_0 c^2)$
Λ_m	modified bearing number, $\Lambda_m=6\mu UL/(p_0 c^2)$
λ	flow parameter, $\lambda=3\mu U/2r_p p_0$
μ	gas dynamic viscosity
ρ	gas density

number are both small. For large values of the Reynolds number they found that the accuracy of the Reynolds equation can be improved significantly by introducing corrections for inertia. However, as pointed out by Feldman et al. [24], the conclusions of some of these studies must be interpreted with care because cavitation, which is the primary mechanism to generate load-carrying capacity in these bearings that use an incompressible lubricant, is either neglected [17–23] or treated in a simplified way [18].

Few studies document the validity and accuracy of the Reynolds equation to simulate bearings lubricated with a compressible fluid. Van Odyck and Venner [25] found that for a compressible parallel slider bearing with an asperity protruding from one surface, a large pressure gradient develops across the lubricant film thickness when increasing the relative sliding velocity between the bearing surfaces. Nevertheless, they concluded that the load-carrying capacity is predicted accurately by the Reynolds equation. Asperity texture on a parallel slider bearing with elastohydrodynamic lubrication was studied by Almqvist and Larsson [26]. They found that the Reynolds equation inaccurately predicts the local bearing pressure when the ratio of the film thickness and the horizontal asperity feature wavelength is larger than 0.01. This is a similar conclusion to the results shown in [17–19]. Additionally, Guardino et al. [27] studied sinusoidal roughness on a Rayleigh step bearing, and concluded that the Reynolds equation accurately predicts the load-carrying capacity when the roughness is small in amplitude compared to the film thickness. The only study to directly evaluate the validity of the Reynolds equation for gas-lubricated textured bearings was performed by Feldman et al. [24]. They evaluated a hydrostatic bearing, i.e., pressure driven flow, and concluded that the load-carrying capacity of the bearing is predicted accurately by the Reynolds equation despite some of the simplifying assumptions being violated locally for certain operating conditions.

Most of these studies focus on validating the prediction of the load-carrying capacity by the Reynolds equation as a function of the texture geometry. Hence, the validity of the Reynolds equation to predict load-carrying capacity in textured bearings is well established for both incompressible and compressible fluids. However, the breakdown of the assumptions inherent to the Reynolds equation when simulating textured bearings with a compressible lubricant has only been partially addressed. In addition to applications where it is sufficient to predict the load-carrying capacity, an accurate prediction of local pressure is the first step in developing fast and accurate particle and mass transport simulations, and is essential in lubricant–solid interaction problems for the case of elastohydrodynamic lubrication and foil bearings [28]. Moreover, no published studies exist that evaluate the validity of the Reynolds equation for modeling shear driven lubricant flow in gas-lubricated, textured slider bearings. Therefore, the objective of this paper is to evaluate the validity and accuracy of the compressible Reynolds equation for gas-lubricated textured parallel slider bearings. We compare simulation results of the Navier–Stokes equations and Reynolds equation, specifically focusing on computing the local bearing pressure and evaluating the accuracy of the assumptions inherent in the application of the Reynolds equation.

2. Methodology

2.1. Model description

Fig. 1 shows a gas-lubricated textured parallel slider bearing. Fig. 1(a) displays the x–z cross-section along the center line of the dimple and Fig. 1(b) shows the top view of the bearing. The dimple is a segment of a sphere of radius r_p and depth h_p, and is centered