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Inertia effect of textured lubricated contact on the bearing performance using CFD approach

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Abstract. Numerous studies reported that inertia of the lubricant has been considered as one of physical parameters which has a strong effect on the load support of textured bearing. In the present study, based on two-dimensional computational fluid dynamics (CFD) technique, the investigation of the inertia effect on bearing performance is carried out varying the texture length. The Navier-Stokes equation coupled with the cavitation model are discretized using finite volume method and solved using the commercial software FLUENT®. The results show that the inertia increases the hydrodynamic film pressure and thus the load support. In addition, it is also found that increasing the texture length as well as Reynolds number will increase the cavitation region.

1. Introduction

As is known, the use of texturing of the lubricated contact is not a new concept. The research on surface texturing gained a new momentum since texturing technique has proven to enhance the lubrication characteristics. Considerable researches have been conducted on textured surfaces with respect to the mechanism of lift generation.

Fowell et al. [1] pointed out that any convergence between the bearing surfaces generates a significant mechanism for lift generation. Gherca et al. [2] numerically explored the effect of geometrical features such as size, density, and shape of texture on the lift generation using a mass-conserving model. Henry et al. [3] experimentally investigated the influence of surface texturing on the performance of bearings. It was found that the textured bearing reduces friction up to 30% at low loads. However, for heavy loads, the textured surface was not recommended to use because their performance is equivalent or even lower than that of an untextured bearing. Yagi et al. [4] concluded that the mechanism of lift generation was strongly affected by boundary condition. Zhang et al. [5] investigated the effect a rectangular array of circle dimples on the film thickness profile. It was highlighted that the lift generation could be improved through appropriate partial arrangement of textures. Later, Shinde and Pawar [6] proposed the optimal surface texturing parameters to improve performance of bearing.

Based on literature survey, one can find that texturing generates more positive effect compared to pure texturing with respect to the load support. However, most of the solution of the lubrication problems have been solved by of Reynolds equation which may be questionable in particular cases



with low viscosity and high runner velocities. Therefore, the present work adopts two methods (Navier-Stokes equation and the Reynolds equation) to investigate the texturing parameter in particular case of texture depth on the hydrodynamic pressure. For obtain more accurate results, the cavitation model is also considered. In this way, the inertia effect as well as the cavitation effect can be investigated more detail.

2. Methodology

2.1. Theory

In the present study, the lubrication problem is solved by the Navier-Stokes equation and continuity equations. The Navier–Stokes (N-S) equations are solved over the domain using a finite-volume method with the commercial CFD software package FLUENT®. The Navier–Stokes and the continuity equations can be expressed, respectively,

$$\rho \frac{Du_i}{Dt} = -\frac{\partial p}{\partial x_i} + \rho G_i + \frac{\partial}{\partial x_j} \left[2\eta e_{ij} - \frac{2}{3}\eta(\nabla \cdot u_i)\delta_{ij} \right] \quad (1)$$

$$\nabla \cdot u = 0 \quad (2)$$

Not like the previously published works in which the cavitation model is ignored, in the present study, the cavitation effect is taken into account. In FLUENT®, there are three available cavitation models: Schneer and Sauer model, Zwart-Gelber-Belamri model and Sighal et al. model [7]. In this study, the Zwart-Gelber-Belamri is employed due to their capability (less sensitive to mesh density, robust and converge quickly [7]).

In addition to the Navier-Stokes equation, in this work, the lubrication problem is also solved by modified Reynolds equation (Eq. 3 as below). For more detail, the derivation of such equation used here is described in [8]. It should be noted that the isoviscous Newtonian one-dimensional Reynolds equation with slip is derived from a simple of the x-component of the Navier-Stokes equation that assumes an incompressible flow, neglecting the inertia effects in the film. However, for the analysis studied here, the slip coefficient is set to zero.

$$P_2 \left[\left(h_p^3 + 3h_p^3 K_p \right) \frac{ab+bc}{ab^2} + \left(\frac{h_0^3 + 3h_0^3 K_0}{a} \right) \right] = P_{atm} \left[\left(h_p^3 + 3h_p^3 K_p \right) \frac{ab+bc}{ab^2} + \left(\frac{h_0^3 + 3h_0^3 K_0}{a} \right) \right] - 6\mu U \left[\left(h_p + h_p K_p \right) - \left(h_0 + h_0 K_0 \right) \right] \quad (3)$$

2.2. CFD Model

Figure 1 gives the schematic illustration of a parallel textured (pocketed) sliding bearing. The assumption of the no-slip is adopted. The main characteristics of the bearing and the lubricant properties studied are presented in Table 1.

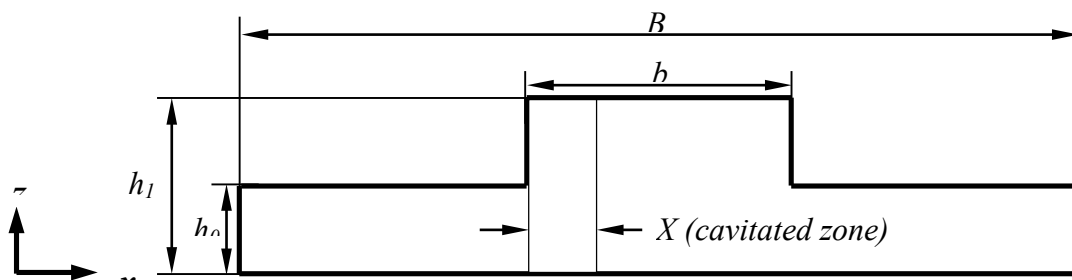
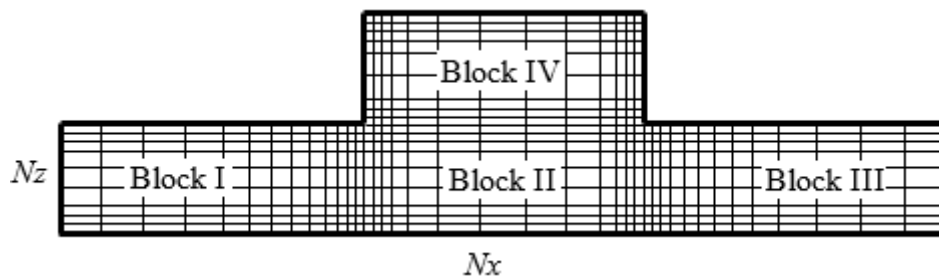


Figure 1. Geometry of slider bearing 2D model.

Table 1. Main bearing parameter and lubricant properties.

Parameter	Symbol	Value	Unit
Length of bearing	B	2.00	mm
Length of texture	b	0.50	mm
Minimum film thickness	h_0	4.00	μm
Maximum film thickness	h_1	8.00	μm
Density of oil	ρ	962	kg/m^3
Density of vapor	ρ	0.013468	kg/m^3
Viscosity of oil	μ_o	0.02556	kg/ms
Viscosity of vapor	μ	1.256×10^{-5}	kg/ms

In the present study, the grid consists of four blocks with grid system as shown in Fig. 2. The detail grid of the mesh is shown in Table 2. The mesh quality consists of a minimum orthogonal quality of 0.994935, maximum ortho skew 0.00506503 and maximum aspect ratio of 129,357. The mesh size in the longitudinal direction (N_x) and transverse (N_z) is 1000 x 140, respectively. These grids generate 70,000 cells and 71,141 nodes based on the independent mesh study.

**Figure 2.** Grid system used in numerical analysis.**Table 2.** Grid detail on each block of geometry.

Block	Direction	Mesh of edge type	Ratio	Interval count
I	N_x	First last ratio	0.1	375
	N_y	Bi-exponent	0.6	60
II	N_x	Bi-exponent	0.6	250
	N_y	Bi-exponent	0.6	60
III	N_x	Last first ratio	0.6	375
	N_y	Bi-exponent	0.6	60
IV	N_x	Bi-exponent	0.6	80
	N_y	Bi-exponent	0.6	250

The boundary condition applies pressure inlet and pressure outlet on the incoming and outgoing sides respectively. The hydrodynamic pressure is used as indicator to describe the inertia effect with respect to the Reynolds equation and Navier-Stokes equation.

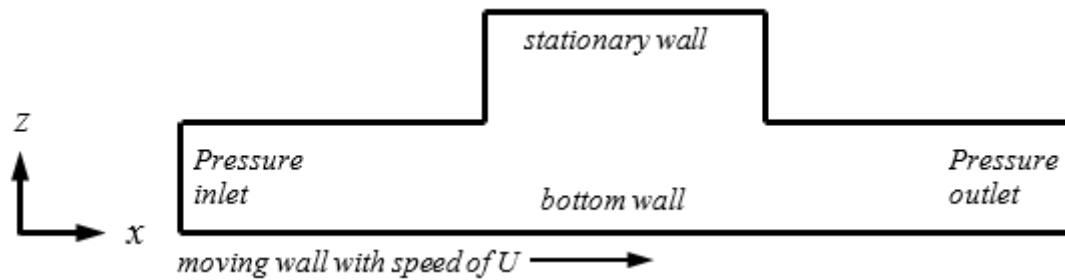


Figure 3. Model of boundary condition.

3. Result and discussion

Results are presented and analysed for inertia effect in slider with Navier-Stokes and Reynold equation. In this study inertia was investigated using two variations of conditions, low inertia and high inertia. The results of the study are represented by the pressure distribution as shown in Figure 4. Based on the results, the pressure distribution under low-inertia conditions is relatively simple and the cavitation does not appear. The same trend is found between Navier-Stokes and Reynold equation for the case of low inertia and it indicates that both methods can be applied to this condition. This is due to low inertia conditions, slider speed is very slow. Different from low inertia, the high inertia conditions has a region cavitation and a higher value of higher pressure distribution. The trend between the prediction of Navier-Stokes and Reynolds equation is different. This difference is caused by the flow of vortex on the texture slider. Vortex flow causes the increase pressure. It is an inertia phenomenon in the texture slider bearing. While on Reynold equation, inertia is ignored. Therefore, the pressure predicted by Reynolds equation is lower when compared to Navier-Stokes. These results show that inertia has a positive effect on hydrodynamic performance.

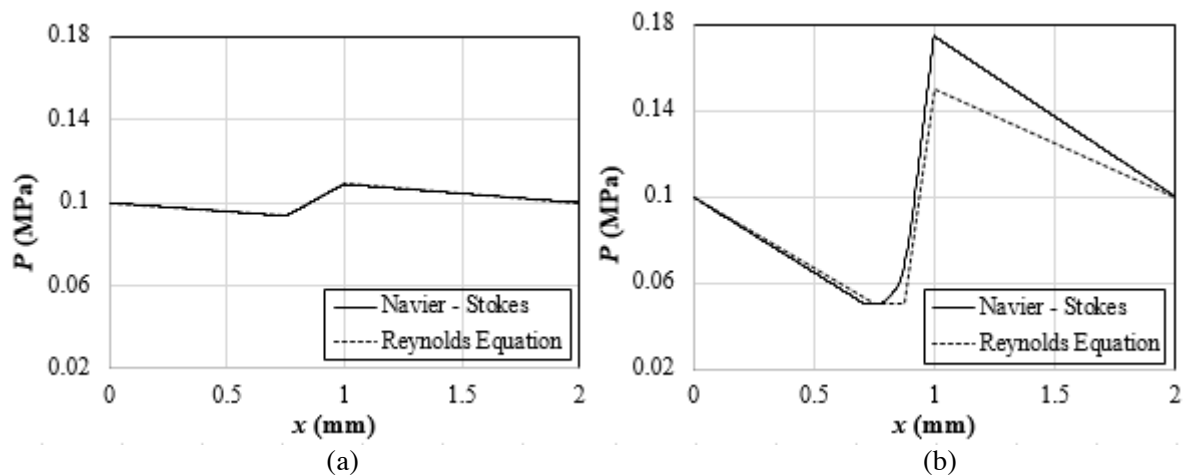


Figure 4. Pressure distribution for (a) low inertia, and (b) high inertia, in the case of $b = 0.25$ mm.

The vortex flow on the slider can be represented in the CFD approach. Figure 5 is a stream function on high inertia conditions with a texture length of 0.25 mm. Vortex occurs at the end position of the texture before cross section reduction. This position is where the maximum pressure occurs from the length of the slider used. While on Reynold equation, such effect is ignored. Therefore, the effect of inertia greatly affects the bearing analysis. That is the reason why inertia becomes important in slider bearing analysis.

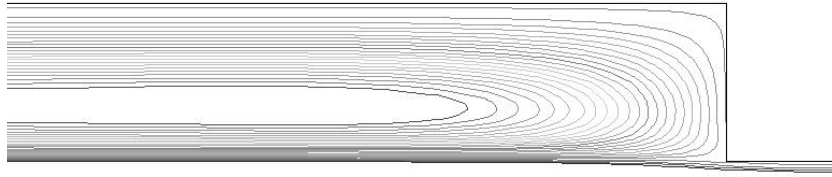


Figure 5. Streamline of fluid flow in high inertia condition.

In the present study, high inertia and low inertia conditions are also applied for the case of textured length with texture length 0.5 mm as shown in Figure 6. It can be seen that in Figure 6(a) the tendency of no cavitation is observed. While in Figure 6 (b) the occurrence of pressure is detected in the beginning of texture. On the other words, the inertia leads to the existence of the cavitation in the textured lubricated contact. With respect to the pressure profile, the prediction for the case of high inertia is greater than that for low inertia. Therefore, based on these results it can be said that inertia is able to increase hydrodynamic pressure. Furthermore, for low inertia conditions, the pressure distribution between Navier-Stokes and Reynolds have the same trend. This means that under low inertia conditions both methods can be applied. The high inertia condition can be modeled with Navier-Stokes.

A similar condition can be found when the texture length is increased to 0.75 mm as shown in Figure 7. For low inertia the small Reynolds number produces the tend in which the cavitation does not occur. While for the case of high inertia, there is a cavitation region. Other interesting result is that the inertia leads to lower hydrodynamic pressure as seen in Fig. 7(b). It indicates that there is a critical value of texture length.

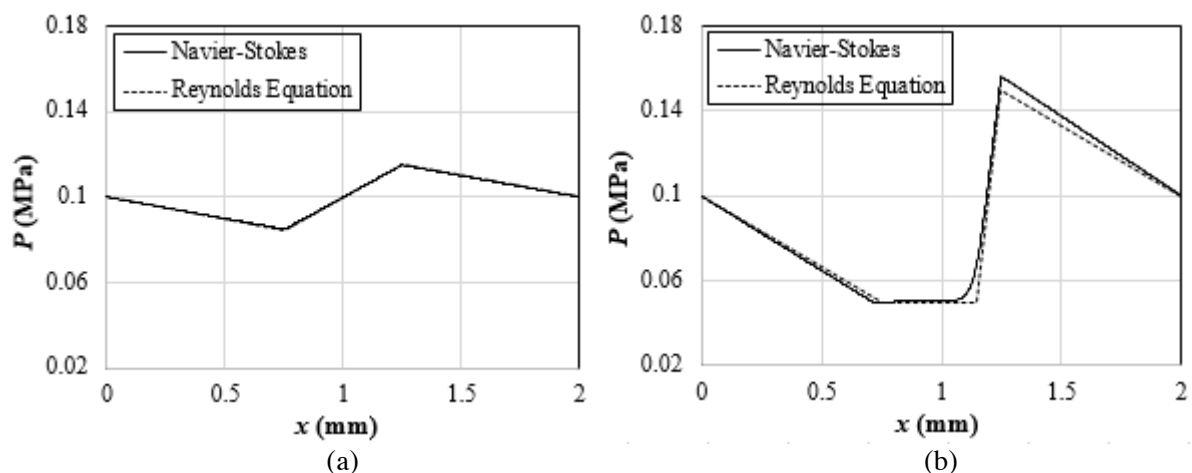


Figure 6. Pressure distribution for (a) low inertia, and (b) high inertia, in the case of $b = 0.25$ mm.

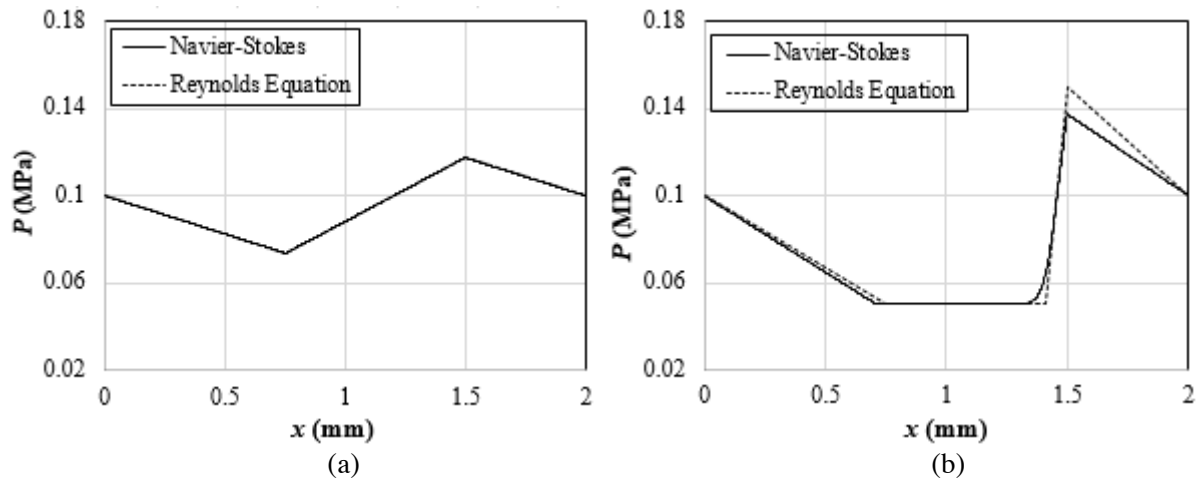


Figure 7. Pressure distribution for (a) low inertia, and (b) high inertia, in the case of $b = 0.75$ mm.

From the simulation results, it is also known that high inertia conditions lead to cavitation. The cavitation region increases with increasing the length of the texture. Inertia also increase the maximum pressure. But under certain conditions, by increasing the length of texture, inertia has a negative effect. It proves that inertia does not always have a positive effect on hydrodynamic pressure.

The comparison of load support is shown in Figure 8. As a note, the load support is obtained from the calculation of the integral of the area under the graph of pressure distribution. The highest load support is found when with a texture length = 0.25 mm with high inertia conditions.

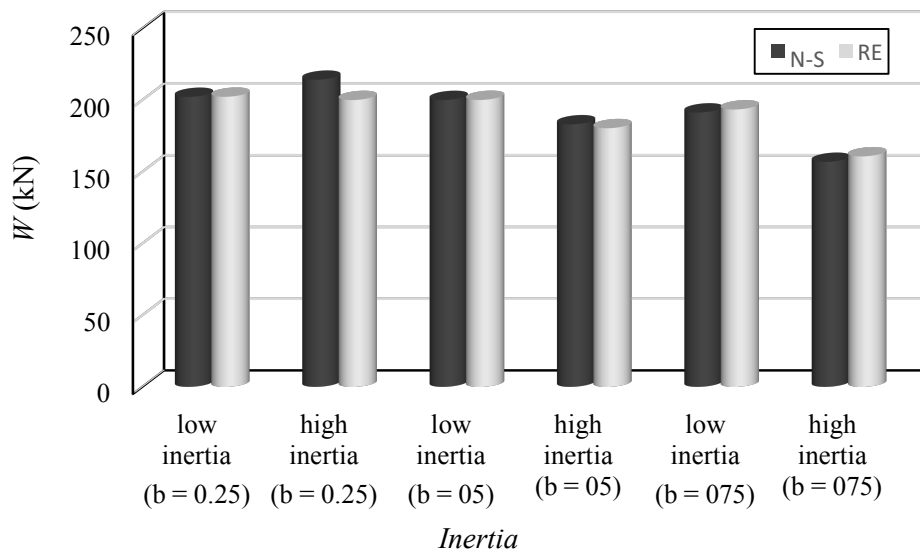


Figure 8. Comparison of the prediction of load support for various conditions.

4. Conclusions

In the present paper, two methods (i.e. Navier-Stokes and Reynolds equations) were used to investigate the effects of inertia in a textured slider bearing. The cavitation model was adopted to model the reformation boundary. Based on the discussion above, the conclusions can be highlighted as follows:

1. Inertia brings two different effects; it can improve the pressure but on the other hand, it can also reduce the pressure depending on the texture length.
2. The increase in the texture length increases the cavitation region.
3. Cavitation region increases with increasing the Reynolds number.

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