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Comparison between non-Newtonian and Newtonian Lubrication of Journal Bearing Considering Cavitation using CFD

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Abstract

Based on computational fluid dynamic (CFD) and mass-conservation cavitation boundary conditions, in this paper, the non-Newtonian and Newtonian lubrication of textured journal bearing are compared. The numerical analysis is undertaken under the condition of different dimple depth and lubricant behavior. The results including the hydrodynamic pressure distribution, load support and friction force are gained and compared for optimizing dimple depth. The present results illustrate a superior performance of the Newtonian lubricant in comparison to the other lubricant types.

Keyword- Cavitation; CFD (Computational Fluid Dynamic); lubrication; non Newtonian.

INTRODUCTION

Journal bearing is mechanical element designed to carry shaft or journal which rotates freely in a supporting metals or shell. A different lubricant type leads to different flow characteristics in the lubricant film and new behavior must be considered. In the view of this nature, many researchers proposed to study the lubricant type effects on bearing performance.

Goyal and Sinhasan [1] presented of static and dynamic performance characteristics of journal bearing with non-Newtonian lubricants. Finite element method was applied to solve the Navier-Stokes and continuity equation. Their main finding was that the bearing with non-Newtonian lubricants shows a higher stability margin compared with a bearing with Newtonian lubricants. Gertzos et al. [2] examined the performance characteristics and the core formation in a hydrodynamic journal bearing lubricated with a Bingham fluid. They found that the load carrying capacity, the hydrodynamic film pressure, and the frictional force of a Bingham solid are larger than those of a Newtonian fluid. Mishra [3] studied the effect of non-Newtonian behavior of lubricant based on power law model on the performance of misaligned journal bearing. Reynolds equation and energy equation were combined to analyse this effect.

With respect to the texturing effect, numerous workers have generated much contributions to improve the lubrication performance. Yadav and Sharma [4] investigated the effect of the texturing parameters on the static and dynamic performance behavior of the bearing. In addition, they found that non-Newtonian lubricant provides the smaller value of fluid film damping coefficient than Newtonian lubricant. Lin et al. [5] investigated the non-Newtonian influences on the non-linear stability boundary of short journal bearings through the transient non-linear analysis. Pratomo et al. [6] investigated the combined effect of single texturing and boundary slip on hydrodynamic pressure based on modified Reynolds equation for lubrication with non-Newtonian power-law fluid. Later, Khatri and Sharma [7] presented a comparative study between textured and non-textured journal bearing configurations. In their study, the analysis was conducted based on Reynolds theory which may be questionable. In the following work, Tauviqirrahman et al. [8] investigated hydrodynamic lubrication performance of single-textured bearing using a mass conserving numerical approach for modelling cavitation.

Based on literature survey, very few studies carried out the investigation of journal bearing lubricated with non-Newtonian lubricant. Following this insight, in the present paper, the comparison of the non-Newtonian and Newtonian lubricant with respect to the lubrication performance of the journal bearing will be explored. In addition, the mass conserving cavitation model which was often neglected in the previous studies will also be considered to obtain more realistic results.

NUMERICAL ANALYSIS

A. Governing Equations

In the present work, the lubrication problem is analysed by the Navier-Stokes equations which 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,

$$\rho(\mathbf{u} \bullet \nabla)\mathbf{u} = -\nabla p + \eta \nabla^2 \mathbf{u} \tag{1}$$

$$\nabla \bullet \mathbf{u} = 0 \tag{2}$$

For following computation, the scheme of a power-law fluid model for modelling the shear stress–strain relation of lubricant is adopted. It means that the shear stress τ is a function of some power of shear strain rate $\dot{\gamma}$ and its mathematical expression reads [7]:

$$\tau = m\left(\dot{\gamma}\right)^{n} \tag{3}$$

where m and n are consistency and flow behavior index, respectively.

In the present work, the turbulent model of realizable k-ε is adopted. For modelling cavitation, the Zwart-Gelber-Belamri is chosen due to their capability (less sensitive to mesh density, robust and converge quickly) [9]. In this model, the liquid-vapor mass transfer (evaporation and condensation) is governed by the vapor transport equation [9]:

$$\frac{\partial}{\partial t}\left(\alpha_v \rho_V\right) + \nabla.\left(\alpha_v \rho_V \mathrm{v}\right) = R_g - R_c \tag{4}$$

where α_v is vapor volume fraction and ρ_v is vapor density. R_g and R_c account for the mass transfer between the liquid and vapor phases in cavitation. For Zwart-Gelber-Belamri model, the final form of the cavitation is as follow [9]:

$$\text{If } p \le p_v,\ R_g = F_{evap} \frac{3\alpha_{\mathrm{nuc}}\left(1-\alpha_{\upsilon}\right)\rho_{\upsilon}}{R_{\mathrm{B}}} \sqrt{\frac{2}{3}\frac{\mathrm{P}_{\upsilon}-\mathrm{P}}{\rho_{\ell}}} \tag{5}$$

$$\text{If } p \ge p_v,\ R_c = F_{cond} \frac{3\alpha_{\upsilon}\rho_{\upsilon}}{R_{\mathrm{B}}} \sqrt{\frac{2}{3}\frac{\mathrm{P}-\mathrm{P}_{\upsilon}}{\rho_{\ell}}} \tag{6}$$

where F_{evap} = evaporation coefficient = 50, F_{cond} = condensation coefficient = 0.01, R_B = bubble radius = 10^{-6} m, α_{nuc}= nucleation site volume fraction= $5\mathrm{x}10^{-4}$, ρ_l = liquid density and p_v = vapor pressure.

B. CFD Model

Schematic representation of textured journal bearing is depicted in Fig. 1, while Table 1 reflects the operational conditions of the bearing. As a note, the no-slip condition is assumed for both surfaces. The automatic mesh feature is adopted as provided by ANSYS. In this work, the non-Newtonian lubricant is modeled as power law with index n of 0.50. The density of the fluid ρ is 960 kg/m3, while consistency index m is 0.04.

In the present study, in order to improve the lubrication performance, the textured journal bearing is used. The texturing considered here is similar to what described by Cupillard et al. [10]. A series of five dimples on the surface of a two-dimensional bearing, with geometry identical to the central cross section of the three-dimensional smooth bearing is applied (Fig. 1). The circular shape of the dimples is assumed due to the ease of manufacture. A dimple is characterized by its width (w) and depth (d) as shown in Fig. 1. In the present study, the dimple depth is varied, while the dimple length is constant. The detail of the dimple geometry is reflected in Table II.

Table I. Properties of the Studied Bearing

Diameter (mm)		Clearance (mm)	Eccentricity (mm)	Rotational speed (rpm)
Journal	Bearing			
42.7	50	0.2	3.4	120

Table II. Geometry of the dimple

dimple depth	d	0.25 and 0.75
dimple length	w	2 mm
Angle	α, θ	69.39°, 30°

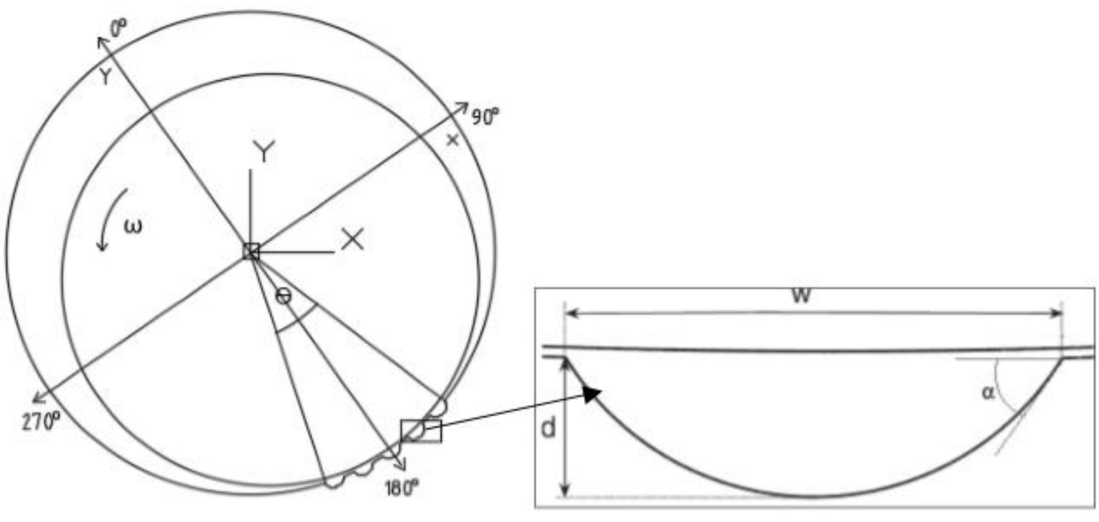


Figure 1. Schematic representation of textured journal bearing. (Inset: Dimple cell geometry). Note: c = Clearance; R_b = Radius of bearing; R_j = Radius of journal; O_b = Centre of bearing; O_j = Centre of journal; ω = Radial speed; and θ = Angular coordinate

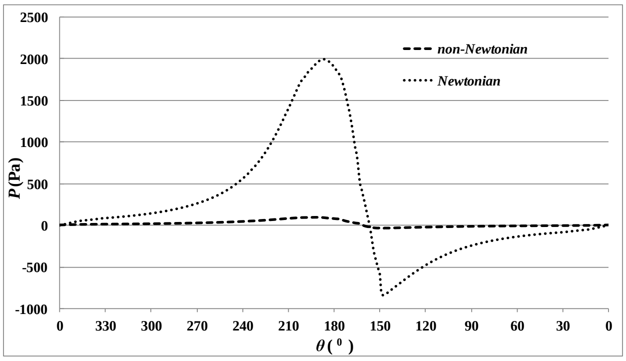
RESULTS AND DISCUSSIONS

Simulation results are shown with respect to the hydrodynamic pressure (and thus the load support) and the streamlines. As a note, the load support is defined by integrating the calculated hydrodynamic pressure field along the surface contact. The main focus of this work is to analyze the comparison between Newtonian and non-Newtonian characteristics.

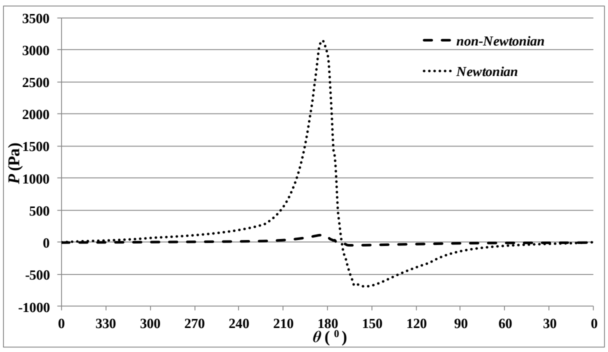
A. Hydrodynamic Pressure Profile

Figure 2 shows the comparison of the hydrodynamic pressure between non-Newtonian and Newtonian varying the dimple depth. It can be observed that with respect to the Newtonian model, increasing the dimple depth will increase the peak of the hydrodynamic pressure. For example, the peak of pressure for the case of d = 0.75 is about 3,100 Pa or 35 % larger than that of d = 0.50. As a consequence, the load support generated is high. It is also shown that high gradient pressure occurs at the texturing region.

Based on Figure 2, it can be found that the Newtonian lubricant yields the larger pressure profile compared to the non-Newtonian lubricant for all value of dimple depth considered here. The most possible explanation is that the non-linear viscosity has a little effect on the generation of the hydrodynamic pressure distribution. It is also confirmed that non-Newtonian lubricant does not seems to be very sensitive to the variation of the dimple depth.



(a)



(b)

Figure 2. Circumferential pressure distributions both for analysis non-Newtonian and Newtonian considering cavitation for (a) d = 0.25; (b) d= 0.75.

Table III shows the lubrication performance of the bearing for two cases (Newtonian versus non-Newtonian) varying the dimple depth. It is clear that with respect to the non-Newtonian scheme, the load support for the case of d = 0.75 is 1.11, or 24 % larger than the case of d = 0.25. In terms of friction force, the bearing with deeper texture gives lower friction. So, two advantages resulted from the textured bearing with dimple depth of 0.75 which is lubricated by Newtonian is the wanted thing to achieve, that is, high load support but low friction. Other interesting finding with respect to the dimple depth is that higher dimple gives more positive effect compared to the lower dimple depth for two cases (Newtonian and non-Newtonian).

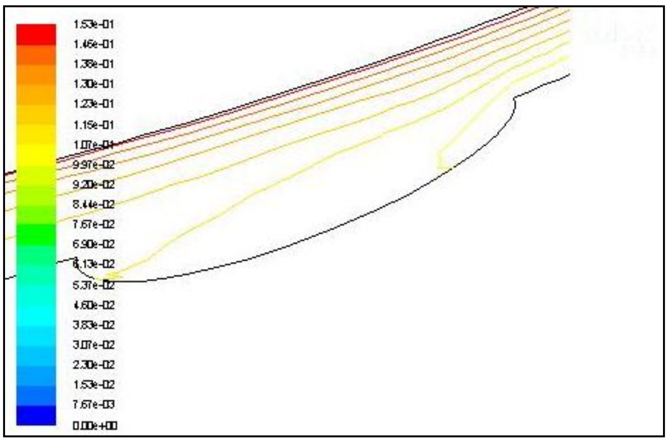
Table III. Lubrication Performance of the Bearing Studied Here

	n = 0.5 (non-Newtonian)		n = 1 (Newtonian)	
	Load Support [N]	Friction [N]	Load Support [N]	Friction [N]
d = 0.25	0.84	0.62	12.77	6.74
d = 0.75	1.11	0.54	26.07	0.17

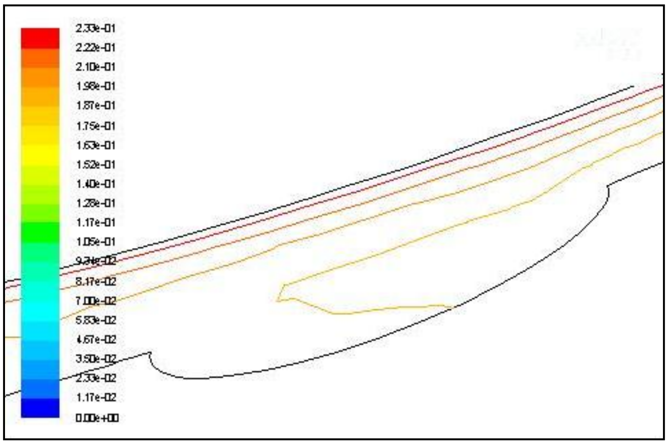
B. Streamline

Streamlines are defined as the scalar stream function ψ and along a streamline ψ = constant. Derivations of ψ refer to the corresponding velocities, i.e. $u = \partial\psi / \partial y$ and $v = \partial\psi / \partial x$. Streamlines depicts a qualitative idea about how the lubricant behaves if there are any vortexes or irregularities.

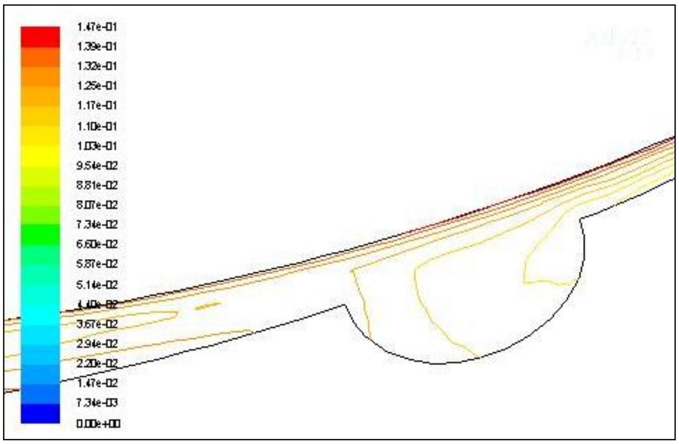
Figure 3 depicts stream functions for textured bearing geometry for different dimple depth d for two cases (Newtonian and non-Newtonian). It can be seen that the higher dimple depth of textured bearing increases the vortex development. This behavior will be more clear in the Newtonian case. This is the reason why the high dimple depth in the texture bearing with Newtonian lubricant produces best performance as mentioned in the earlier section.



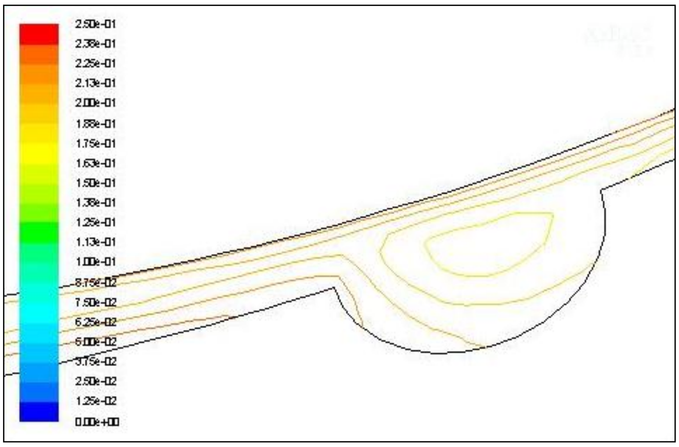
(a) non-Newtonian case with d = 0.25



(b) Newtonian case with d = 0.25



(c) non Newtonian case with d = 0.75



(d) Newtonian case with d = 0.75

Figure 3. Stream function around the dimple varying the dimple depth for two lubricant types

CONCLUSION

The comparison of hydrodynamic lubrication behavior of the textured journal bearing between non-Newtonian lubricant and Newtonian lubricant was studied for incompressible, isothermal, and steady conditions. The conclusions are summarized as follows:

1. Newtonian lubricant gives better performance compared to non-Newtonian lubricant in terms of the load support and the friction force.
2. The higher the dimple depth, the better the lubrication performance.

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