The texture effect on the static characteristics of hydrodynamic journal bearings

N. Tala-ighil, A. Brick Chaouche and A. Mokhtari

***Abstract*— Understanding the influence of surface texturation on journal bearing performances inevitably involves experimental investigations pending being followed by a numerical modeling of the problem. This work consists to model and to understand the evolution of journal bearing characteristics with and without presence of textures on the bearing surface. The numerical approach based on the finite difference method is used in this analysis. The results of our simulations are in good agreement with those obtained experimentally by Xiaobin.**

***Index Terms*— Cavitation, friction, hydrodynamics, journal bearings, lubrication, numerical analysis, surface texture, thin films, tribology.**

I. INTRODUCTION

Nowadays, there is a strong need to make machines more efficient by looking for power losses within the machine and to reducing them. The most important losses in a machine come from the bearings [1]. These bearings have several advantages such as low friction and wear, good heat dissipation through the oil, and reduction of noise and vibrations. Their lubrication is really important because the contact between surfaces would cause rapid wear [2].

The deterministic roughness that is known as surface texture was introduced deliberately on the bearings with the help of micro-fabrication techniques. Surface texturing is claiming progressively more attention and is expected to be a major component in future bearing structure design as demonstrated by the authors [3, 4]. By means of the new technology as the laser surface texturing [5], chemical etching [6], novel dressing technique [7]… It is now possible to produce controlled micro-geometries (textures) on journal bearing surfaces to improve the overall tribological performance including the friction reduction, the reliability improvement, the severity conditions increase, and the energy consumption lowering. Some other and recent studies [8–14] have established that the surface texture geometry such as texture depth, width, number of textures, and location of textures influence the bearing performance.

The author Xiaobin [15] has presented experimental results on the effect of dimples on the Stribeck curve. Load, oil type, dimple size, depth and shape are varied to explore their influence on the friction characteristics. In recent work, the authors show that the most significant characteristics can be improved through an appropriate arrangement of the textured area on the contact surface [16].

II. THEORY

In a hydrodynamic lubrication problem, the governing equations in a full hydrodynamic lubrication region can be described by the well-known Reynolds' equation [17]. For Cartesian coordinates, when the thickness of the lubricant film h is in the direction of the y axis (Figure 1a), the pressure in the lubricating film for a journal operating at steady state, is governed by the following equation :

$$\frac{\partial}{\partial x}\left(\frac{h^3}{12\mu}\frac{\partial P}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{12\mu}\frac{\partial P}{\partial z}\right)=\frac{u_2-u_1}{2}\frac{\partial h}{\partial x}+\frac{\partial h}{\partial t} \quad (1)$$

P is the lubricant pressure, h is the film height, μ is the dynamic viscosity and u_1, u_2 are the velocities of the journal and the bearing respectively. The bearing geometry is shown in Figure 1.

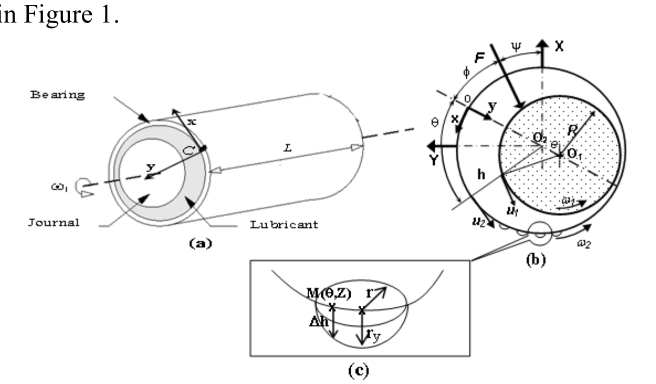

Fig. 1. Hydrodynamic journal bearings (a) Bearing geometry, (b) Right section of the bearing, and (c) Texture geometry.

When using the variables: $Z=z/L$, $\theta=x/R$ and $(u_2-u_1)=R.(\omega_2-\omega_1)$, we have :

$$\frac{\partial}{\partial \theta}\left(h^3\frac{\partial P}{\partial \theta}\right)+\left(\frac{R}{L}\right)^2\frac{\partial}{\partial Z}\left(h^3\frac{\partial P}{\partial Z}\right)=6\mu R^2\left[(\omega_2-\omega_1)\frac{\partial h}{\partial \theta}\right] \quad (2)$$

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R is the radius of the journal, L the length of the bearing and ω_1, ω_2 are respectively, the angular velocities of the journal and the bearing.

The film thickness h can be written:

$$h = C(1 + \varepsilon \cos\theta) + \Delta h\ (\theta,Z) \tag{3}$$

In the equation above, $\Delta h(\theta,Z)$ is the film thickness variation due to the textured surface, textured surface, ε the relative eccentricity of the journal and C the radial clearance bearing.

The boundary conditions, known as Reynolds boundary conditions, are used to determine the rupture zone of the film. They consist in ensuring that $\partial P/\partial\theta = \partial P/\partial Z = 0$ and $P=0$ at the rupture limits of the film lubricant.

The bearing is operating under steady state conditions; the applied load F is constant and its direction is vertical ($\psi=0$). The total load W (supported by the contact) is calculated by integrating the pressure field along the surface contact of the journal bearing, than the attitude angle ϕ is obtained (figure 1b).

The figure 2 shows the texture shape. The centre of the texture is located on the surface of the bearing, making $y_c=0$. The depth at point M on the surface bearing situated on the texture geometry is defined by Δh (Figure 1c).

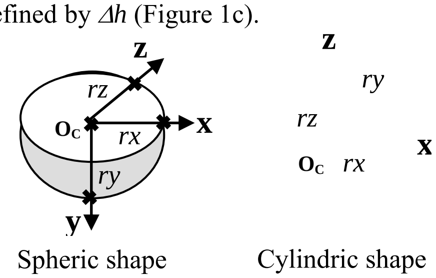


Fig. 2. Texture shape.

The spherical texture geometry is defined by,

$$\frac{(x - x_C)^2}{r^2} + \frac{(\Delta h - y_C)^2}{r_y^2} + \frac{(z - z_C)^2}{r^2} = 1 \tag{4}$$

In the case of spherical geometry $r_x=r_z=r$, where r is the radius of the circle on the bearing surface. Finally,

$$\Delta h(\theta, Z) = \frac{r_y}{r}\sqrt{r^2 - (x - x_C)^2 - (z - z_C)^2} \tag{5}$$

III. Problem resolution

The bearing surface is stationary and the journal is moving. Only one-half of the journal bearings system is studied because of the symmetry of the bearing and because refined uniform meshes are used.

TABLE I
GEOMETRICAL PARAMETERS AND OPERATING CONDITIONS FOR THE JOURNAL BEARING STUDIED [15].

External force F (N)	667
Journal speed ω (rpm)	80 to 5000
Shaft diameter (m)	0.024625
Bearing length L (m)	0.0254
Radial clearance C (m)	0.000085
Initial temperature T_0 (°c)	40
Initial viscosity μ_0 (Pa.s)	0.081496
Initial density ρ (kg/m^3)	876.3
Mesh: $N_\theta \times N_Z$ (nodes)	929 x 305

The data cited in table 1 are used in our simulations. The imposed precisions for the calculations of the pressure P is $\varepsilon_P=10^{-4}$ and the load W is $\varepsilon_W=10^{-5}$. The mesh size used is $N_\theta=929$ and $N_Z=153$.

Three cases are studied: the case of smooth bearing without texture (Conventional bearing) and the two cases of textured bearing (the entire surface of the bearing is textured, from 0 to 360°). The considered texture geometry in the study is spherical with dimensions [15]:

- Texture 1 : diameter of 2mm and depth of d=0.165mm
- Texture 2 : diameter of 4mm and depth of d=0.448mm

The determination of the pressure field in the lubricant film consists of the numerical resolution of (2) by using the Finite Difference Method. The most usual resolution method is that of Christopherson [18] and is used here. The resolution of linear systems obtained after discretisation is obtained by the iterative method of Gauss-Seidel. The use of an iterative method for the resolution is justified by the application of the Reynolds boundary conditions. The analysis only leaves pressure as the unknown to be solved, while the eccentricity is given (e.g. according to the load difference between the given value F and the computational one W, at the previous iteration step). It implies that only (2) is used to form the final equation system for obtaining P, while a cavitation conditions should also be satisfied. For a steady-state regime, the computational procedure consists of giving initial values to the eccentricity e. The pressure field, at each nodal point under a steady external loading F (shown in Figure 1b) is obtained, verifying the pressure convergence condition $\left|\Delta P_i\right| / \left|P_i\right| \le \varepsilon_P$ at each node i of the bearing surface mesh.

The supported load W and bearing attitude angle ϕ are calculated. The calculated load W and fixed load F are compared; the process stopping after the load convergence condition $\left|F - W\right| / \left|F\right| \le \varepsilon_W$ is satisfied. If this error control is not satisfied, the eccentricity value is updated and the process of calculation begins.

IV. Results

In this section, we validate the numerical model by comparing our numerical results with experimental results obtained by Xiaobin [4].

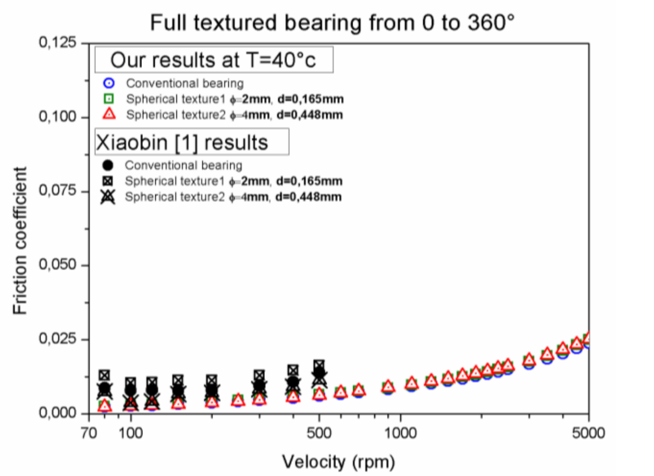
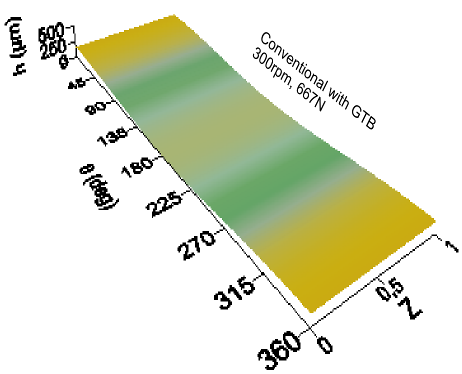


Fig. 3. Friction coefficient as a function of velocity for Xiaobin **[4]** results and our simulation results.

Figure 3 shows a comparison between our numerical results (speed range of 80-5000 rpm) and those obtained experimentally by Xiaobin [15] (speed range of 80-500 rpm). The results of the two studies are in good agreement. The friction coefficient gradually increases with the increase of the journal speed and it varies in the range of 0,001-0,025. The case of texture 1 gives a higher coefficient of friction (unfavorable case) than the conventional case. A completely textured bearing (from 0 to 360°) with a case of texture 2 gives a low friction coefficient (favorable case) compared to the cases of conventional and texture 1.



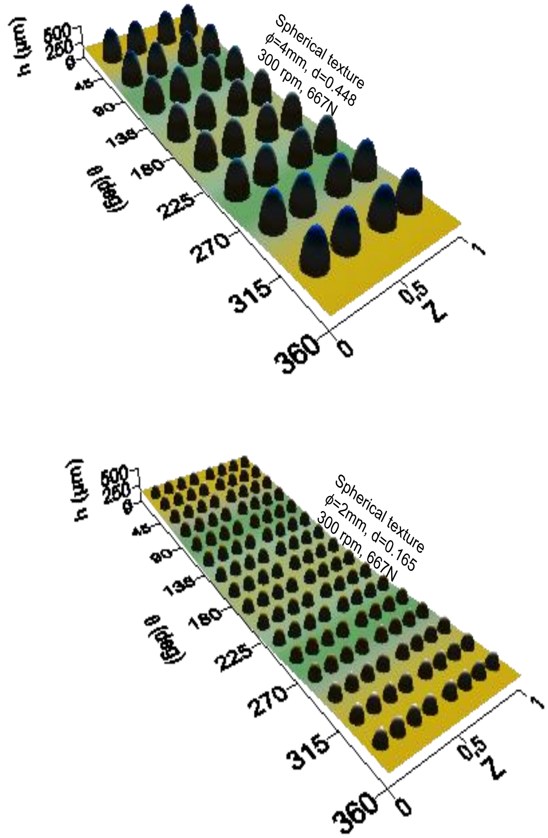
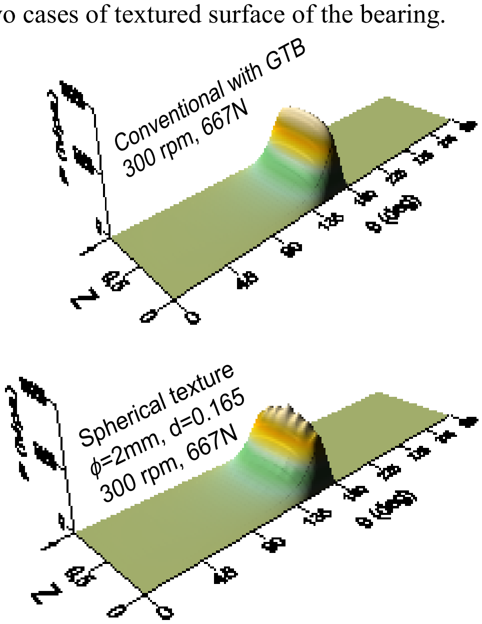


Fig. 4. Evolution of film thickness on the contact for textured and untextured surface.

Figure 4 shows the evolution of the lubricant film thickness in the contact for the three studied cases (conventional bearing, texture 1 and texture 2) at speed of 300 rpm. It can be noted the distribution of textures on the surface of the bearing in the two cases of textured surface of the bearing.



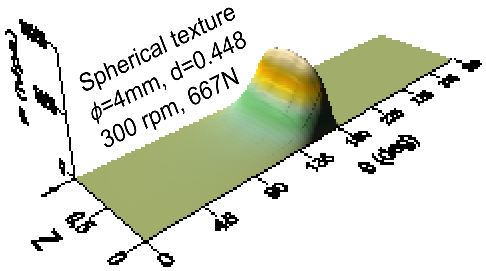


Fig. 5. Evolution of pressure field on the contact for textured and untextured surface.

Figure 5 shows the evolution of the pressure field and the lubricant film thickness in the contact for the three studied cases (conventional bearing, texture 1 and texture 2) at speed of 300 rpm. It can be noted the effect of the geometry and size of the texture on the shape of the pressure field, particularly near the position of 180 degrees.

V. Conclusion

As a conclusion, we note that the results of our simulation are in good agreement with those obtained experimentally by the author [15].

The presence of textures over the entire surface of the bearing does not necessarily improve the hydrodynamic characteristics (friction reduction, increased thickness of the fluid film, improved hydrodynamic lift ...) in a journal bearing contact.

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