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## THE EFFECT OF OIL VISCOSITY ON THE PERFORMANCES OF A TEXTURED JOURNAL BEARING

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### ABSTRACT

This work compares the viscosity effect of two different oils used as lubricant in a textured hydrodynamic journal bearing. The textured bearing performance enhancement passes by a minimum film thickness and a friction torque improvement through an appropriate surface texturing. The used numerical approach in this analysis is finite difference method. It is found that in the hydrodynamic lubrication regime, the improvement of the performances of the textured (dimpled) journal bearing depend strongly on the journal velocity and the lubricant viscosity.

**Keywords:** Journal bearing, texture, viscosity, lubricant.

### INTRODUCTION

The modern period of lubrication began with the work of Osborne Reynolds (1842-1912). Hydrodynamic lubrication is an excellent method of lubrication since it is possible to achieve coefficients of friction as low as 0.001 (Reynolds, 1886). Under ideal conditions, Reynold showed that the lubricant pressure was great enough to keep the two bodies from having any contact and that the only friction in the system was the viscous resistance of the lubricant. The distance between the two surfaces decreases with higher loads on the bearing, less viscous fluids, and lower speeds. Lubricant selection was typically based on experience and knowledge (Lauer, 2008). Today, this approach is no longer viable due to the requirements of the current demanding environments to run faster, longer and hotter.

Today's lubricants must satisfy extreme requirements that are specific to each application. The minimum lubrication film thickness increases as the entraining velocity increases and the viscosity becomes greater, and that it decreases as the load grows (Grote, 2009). The minimum film thickness is normally greater than 1µm. However it was only recently that been such textures engineered in order to improve the machine elements tribological performance (Etsion, 1999). Microtextures act as micro- hydrodynamic bearings, enhance load support and increase film thickness, which leads to lower friction compared to untextured surfaces.

Recently, experimental results concerning the dimples effect on the Stribeck curve have been presented (Lu and Khonsari, 2007). Load, oil type, dimple size, depth and shape were varied to explore their influence on the friction characteristics. It is shown that with proper dimensions of dimples, the friction performance of journal bearings can be improved.

## THEORY

In a hydrodynamic lubrication problem, the governing equations for a full hydrodynamic lubrication region can be described by the known Reynolds' equation (Frêne, 1997).

$$\frac{\partial}{\partial \theta} \left( h^3 \frac{\partial P}{\partial \theta} \right) + \left( \frac{R}{L} \right)^2 \frac{\partial}{\partial Z} \left( h^3 \frac{\partial P}{\partial Z} \right) = 6\mu R^2 \left[ (\omega_2 - \omega_1) \frac{\partial h}{\partial \theta} \right] \quad (1)$$

$R$  is the bearing radius,  $L$  the bearing length.  $\omega_1$  and  $\omega_2$  are respectively, the rotational speeds of the journal and the bearing (Figure 1a). The film thickness  $h$  is:

$$h = C(1 + \varepsilon \cos \theta) + \Delta h(\theta, Z) \quad (2)$$

$\Delta h(\theta, Z)$  is the film thickness variation due to the dimple surface,  $\varepsilon$  the relative eccentricity of the journal and  $C$  the radial clearance bearing.

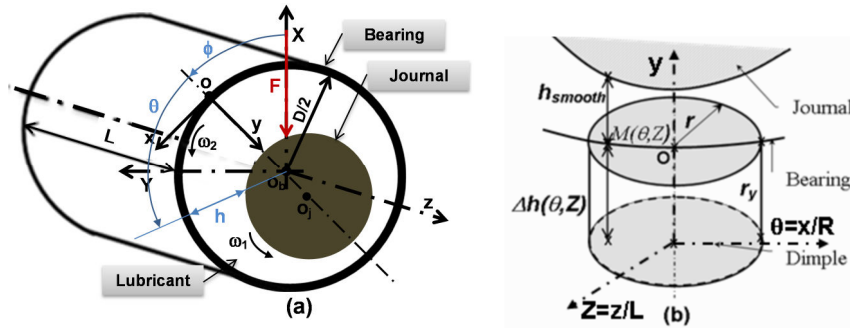


Fig.1 - (a) Hydrodynamic journal bearing geometry and (b) Cylindrical dimple shape.

In the case of cylindrical dimple geometry (fig.1b), the equation of geometry is defined by,

$$(x-x_o)^2 + (z-z_o)^2 = r^2 \quad (3)$$

The film thickness variation can be written:  $\Delta h(\theta, Z) = r_y$

The applied Reynolds boundary conditions on pressure are used to determine the film rupture zone. They consist in ensuring that  $\partial P / \partial \theta = \partial P / \partial Z = 0$  and  $P = 0$  at the rupture limits for the film lubricant defined by the rupture angle  $\theta_e$  (vary from 180 to 210°).

## RESOLUTION METHOD

The Reynolds equation (1) after applying the Finite Difference Method can be written:

$$P_{i,j} = (1 - \Omega) P_{i,j} + \Omega [A_1 P_{i+1,j} + A_2 P_{i-1,j} + A_3 (P_{i,j+1} + P_{i,j-1}) + A'_3 (P_{i,j+1} - P_{i,j-1}) + A_4] \quad (4)$$

$P_{i,j}$  is the pressure value at the mesh node  $(i,j)$ .  $A_1, A_2, A_3, A'_3$  and  $A_4$  are coefficients.  $\Omega$  is an over-relaxation parameter (in lubrication problems, the value of this parameter is generally chosen between 1.5 and 1.85). The best resolution method used to calculate the pressure field is that of (Christopherson, 1941). The Gauss-Seidel iterative method is used to solve the linear systems (4) obtained after discretization as a consequence of the Reynolds boundary conditions. The used global computational procedure contains two computational processes linked together. The first one deals with pressure  $P$  computation until convergence ( $|\Delta P_i|/|P_i| \leq \varepsilon_p$ ), while the second one concerns the

relative eccentricity  $\varepsilon$  computation until convergence on the load (the applied load  $F$  and the calculated ones  $W$  are compared  $|F-W|/F \leq \varepsilon_W$ ). After calculating the pressure field, the static characteristics are calculated. The global process stop after the load convergence condition  $|F-W|/F \leq \varepsilon_W$  is satisfied.

The used precisions for the calculation are:  $\varepsilon_P = 10^{-7}$ ,  $\varepsilon_W = 10^{-5}$  and  $\varepsilon_T = 10^{-4}$ . The meshing is chosen so that to obtain a square mesh (verified condition  $\pi.D/N_\theta \cong (L/2)/N_Z$ ) and the optimized mesh size is  $N_\theta = 929$  and  $N_Z = 153$  (only one-half of the bearing is meshed), respectively along the circumferential and the axial directions.

## RESULTS

The bearing surface is stationary ( $\omega_2=0$ ) and the journal is moving. Only one-half of the journal bearing system is studied due to the bearing symmetry and the used refined uniform meshes. The Geometrical parameters for the studied journal-bearing are: the journal diameter is 24.625 mm, the bearing length is 25.400 mm and the radial clearance is 85  $\mu\text{m}$ .

Two oils are applied. The oil 1 (ISO VG 32, 31.3 cSt at 40 °C) has a much smaller viscosity than oil 2 (ISO VG 100, 93 cSt at 40 °C). Their properties are shown in Table 1.

Table 1 - Oil properties

	Viscosity (cSt)		Specific Gravity at 15 °C
	40 °C	100 °C	
<b>Oil 1 (ISO VG 32)</b>	31.30	5.25	0.877
<b>Oil 2 (ISO VG 100)</b>	93.00	10.80	0.890

Three configuration cases are studied: one smooth bearing without texture (Conventional bearing) and two cases of textured bearing (the bearing is fully textured from 0-360° or just the first half of the bearing surface is textured 0-180°). The considered dimple geometry in the study is cylindrical with diameter of 4 mm and depth of 0.130 mm. The applied load is 667 N.

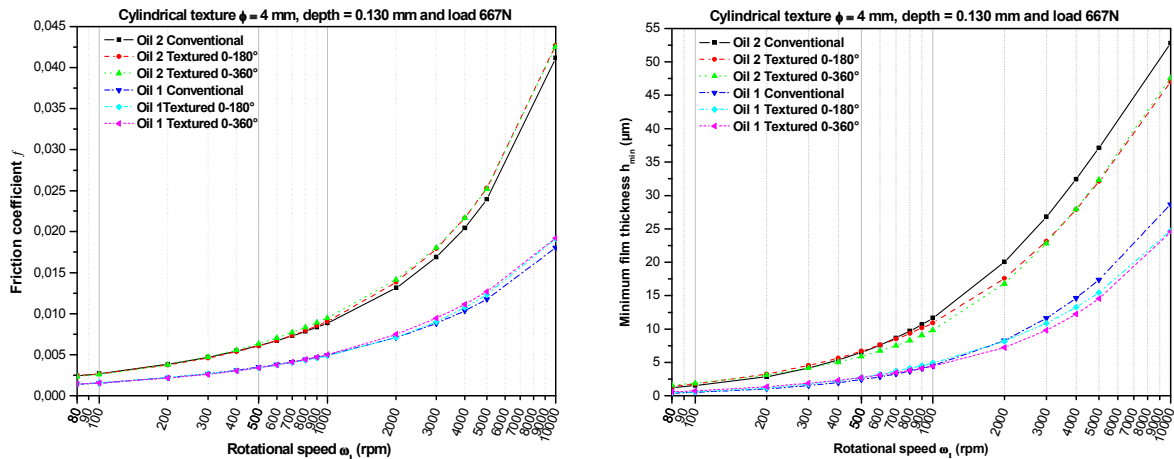


Fig. 2 - Variation of the friction coefficient and the minimum film thickness with the rotational speed of the journal

As shown in Figure 2, the friction coefficient and the minimum film thickness, increase with the journal speed increases. The friction coefficient and the minimum film thickness in the case of journal bearing (conventional and textured bearing) lubricated with oil 2 are higher than that with the lighter oil 1.

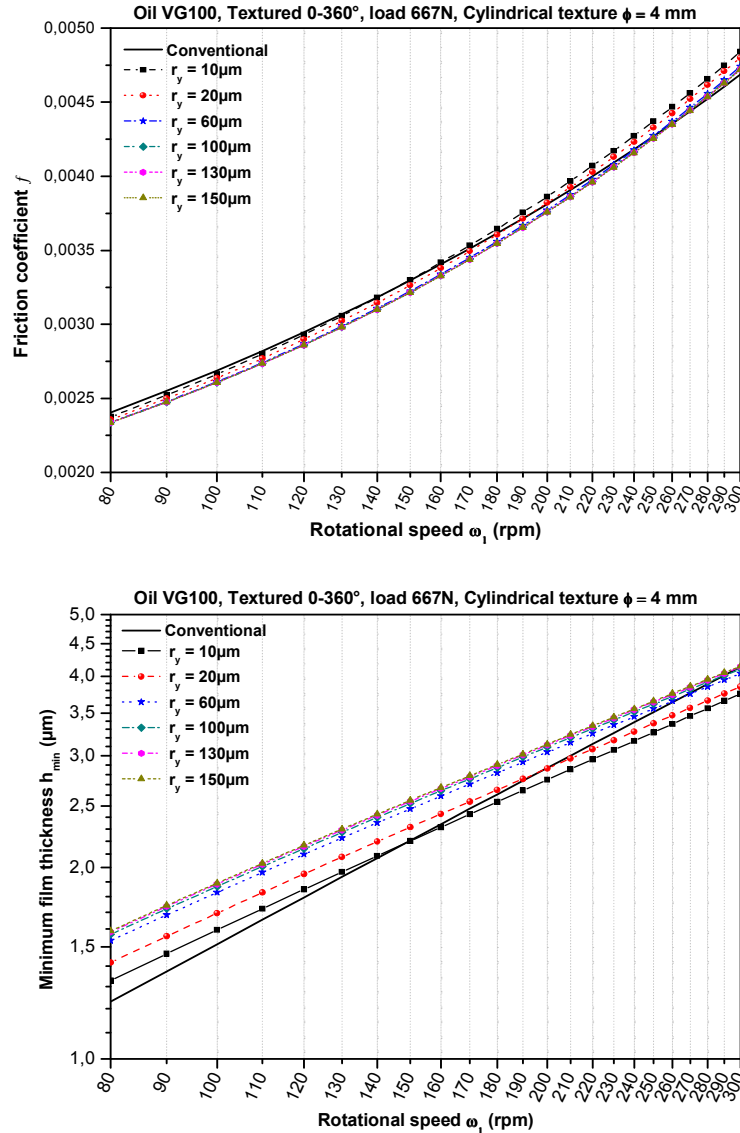


Fig. 2 - Variation of the friction coefficient and the minimum film thickness with the rotational speed of the journal for different depths of cylindrical the texture

From Figure 2, we note that for a fully textured journal (maximum density of textures on the bearing surface), the positive effect of surface texturing on the carrying capacity (improving the minimum film thickness of the fluid) and on the friction (reduction of the friction coefficient) is justified for the low shaft rotation speeds (in our case, it should be lower than a transition speed of 300 rpm for a dimple depth of 150  $\mu m$ ).

This transition speed increases with the dimple depth until reaching the limit of 300 rpm, as from the depth of 60  $\mu m$  and especially for the three greatest values of the dimple depth which are 100, 130 and 150  $\mu m$  (despite a gap of 50  $\mu m$ ) the curves of  $f$  and  $h_{min}$  tend to overlap.

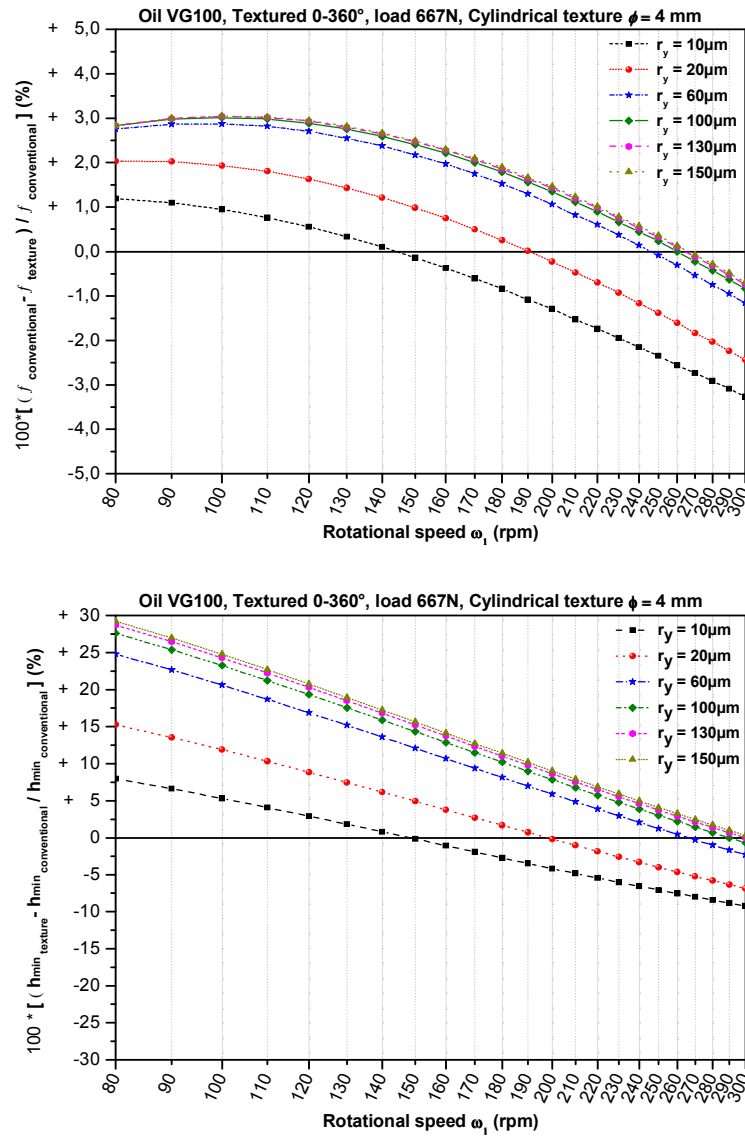


Fig. 3 - Variation of relative discrepancy for the friction coefficient and the minimum film thickness with the journal speed for different depths of the cylindrical texture

From Figure 3, one can observe that, for a fully textured bearing, the maximum positive contribution of the textures is obtained for a very lower speed (from 80 to 110 rpm) and for a specific dimple depth which in our case corresponds to the value of 150  $\mu\text{m}$ .

For a dimple depth greater than 60  $\mu\text{m}$ , it is noted that the maximum positive of the relative discrepancy on the minimum film thickness  $h_{\text{min}}$  converges to a value close to 30% corresponding to the lowest speed of 80 rpm. As for the maximum positive of the relative discrepancy on friction coefficient  $f$ , it corresponds to a value close to 3% and it is obtained for a speed of 100 rpm and a depth of 150  $\mu\text{m}$ .

## CONCLUSION

This study shows that in the hydrodynamic lubrication regime, the improvement of the most important characteristics (the friction coefficient and minimum film thickness) of a textured (dimpled) journal bearing depend strongly on the journal velocity and lubricant viscosity.

The fully texturing of a journal bearing is highly favorable at very low speeds and for a specific texture depth of 150 microns, in this case study.

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