Performance evolution of fully and partially textured hydrodynamic journal bearings lubricated with two lubricants

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Abstract. This study investigates the evolution of the main bearing performance of partially and fully textured hydrodynamic journal bearing. The viscosity effect is also analysed by the mean of numerical simulations for two types of oil: the oil 1 (ISO VG 32, 31.3 cSt at 40 °C) has a lower viscosity than oil 2 (ISO VG 100, 93 cSt at 40 °C). Reynolds equation is solved by finite difference and Gauss-Seidel methods with over-relaxation for various operating conditions. It is shown that, under hydrodynamic lubrication regime, the improvement of the most important characteristics (the friction coefficient and minimum film thickness) of a textured journal bearing depend strongly on the lubricant viscosity and the journal rotational speed. The fully textured journal bearing is highly favorable at very low speeds while the partially textured journal bearing is more suitable for slightly higher speeds. The gain in bearing performance due to the texturing of the bushing disappears at a critical speed of the journal and then, for higher rotational speeds, the presence of textures becomes detrimental.

1. Introduction
The modern theory of lubrication was introduced by the great work of Osborne Reynolds (1842-1912). The excellent hydrodynamic lubrication method permits to attain a low friction coefficients close to 0.001 [1]. Under ideal conditions, Reynolds showed that the lubricant pressure was sufficient to separate the two bodies in motion and the friction in the contact is due to the viscous resistance of the lubricant. The distance between the contact surfaces reduces with higher carrying loads, small speeds, and less viscous lubricants. In the past, the choice of lubricant was essentially based on experience and knowledge [2]. Today, because of the extreme operating conditions and increasingly stringent of machines (run faster, longer and hotter), this approach is no longer reliable.

Today's lubricants must adapt to extreme operating conditions of modern machines. The minimum film thickness of the lubricant increases with the velocities increase of the contact surfaces and decreases as the load increases [3]. 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 [4]. Microtextures act as micro- hydrodynamic bearings enhance load carrying capacity and increase film thickness, which also leads to lower friction compared to untextured surfaces.
Recently, experimental results concerning the dimples effect on the Stribeck curve have been presented by Lu and Khonsari [5]. 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.

2. Theory

In a hydrodynamic lubrication problem, the governing equations for a full hydrodynamic lubrication region can be described by the known Reynolds’ equation [6].

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

(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:

\[
h = C(1 + \varepsilon \cos \theta) + \Delta h(\theta,Z)
\]

(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.

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

\[
P_{i,j} = \left( I - \Omega \right) P_{i,j} + \Omega \left[ A_1 P_{i+1,j} + A_2 P_{i-1,j} + A_3 \left( P_{i,j+1} + P_{i,j-1} \right) + A_4 \left( P_{i,j+1} - P_{i,j-1} \right) + A_5 \right] (4)
\]

(4)

\( P_{i,j} \) is the pressure value at the mesh node \((i,j)\). \( A_1, A_2, A_3, A_4, A_5 \) and \( \Omega \) 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 [7]. 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 \(|\Delta P|/|P| \leq \varepsilon_0\), 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
After calculating the pressure field, the static characteristics are calculated. The global process stops after the load convergence condition $|F-W|/|F| \leq \varepsilon_W$ is satisfied. The used precisions for the calculation are: $\varepsilon_p = 10^{-7}$ and $\varepsilon_W = 10^{-5}$. 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.

### 3. Results and discussion

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 µm. 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.

| Oil                  | Viscosity (cSt) | Specific Gravity at 15 °C |
|----------------------|-----------------|---------------------------|
| **Oil 1 (ISO VG 32)**| 31.30           | 5.25                      |
| **Oil 2 (ISO VG 100)**| 93.00           | 10.80                     | 0.877

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 texture geometry in the study is cylindrical with diameter of 4 mm and depth of 0.130 mm.

As shown in figure 2 and for a constant applied load of 667 N, the friction coefficient (figures 2a1 and 2a2) and the minimum film thickness (figures 2b1 and 2b2) are increasing with the journal speed increase. 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 those with the lighter oil 1. One can observe that, for a fully textured journal bearing and for a partially (only the first one half of the bearing surface is textured) journal bearing, the positive effect of surface texturing on the carrying capacity (improvement of the minimum film thickness) and on the friction (reduction of the friction coefficient) is effective for the low shaft rotational speeds up to a critical speed which depends on the textured area.

As seen on figures 2b1 and 2b2, the full texturing is very favorable at very low speeds below 152 rpm for oil 2 and 454 rpm for oil 1, while partial texturing is more suitable for speeds between 152 and 612 rpm for the oil 2 and from 454 to 1750 rpm for the oil 1. For the configurations of fully textured bushing and above the critical speed of 298 rpm for oil 2 and 907 rpm for oil 1, the texturation effect on the journal bearing performance becomes negative. For the partially textured bushing configurations, a similar phenomenon is observed for critical speeds of 612 rpm for the oil 2 and of 1750 rpm for oil 1. For both studied oils, the transition speeds for each type of texturing correspond to constant values of the friction coefficient and minimum film thicknesses. As observed on figure 2b1, for $h_{\text{min}} = 2.6$ µm, the positive effect of partial texturing become greater than that of fully texturing at speed greater than 152 rpm for oil 2 and 454 rpm for oil 1. For $h_{\text{min}} = 4.1$ µm, the positive effect of fully texturing disappears at values of 298 rpm for oil 2 and of 907 rpm for oil 1. For $h_{\text{min}} = 7.6$ µm, the texturation effect becomes unfavorable from the speeds of 612 rpm for oil 2 and of 1750 rpm for oil 1. The highest gains in minimum film thickness are observed for lowest rotational speed (80 rpm) and for the fully textured bearing surface (0 to 360°): +35.5% and +28.7% for oil 1 and oil 2, respectively (figure 2b2). For the same configurations and operating conditions, the friction reduction is $-2.50\%$ for oil 2 (figure 2a2).
As shown in Figure 3 and for a fixed journal speed (80, 600 and 2000 rpm), the minimum film thickness is decreasing with the carrying load increase.

Figure 3. Minimum film thickness versus applied load for three speeds (80, 600 and 2000 rpm).
The minimum film thickness in the case of journal bearing (conventional and textured bearing) lubricated with oil 2 (ISO VG100) are higher than those with the lighter oil 1 (ISO VG32) for each value of the journal speed. One can observe that, for a textured journal bearing, the positive effect of surface texturing especially on the carrying capacity is effective for the low shaft rotational speeds regardless of the supported load variation.

As observed in figure 4, the main reason of the gain in bearing performance is that the presence of the dimples on the bearing surface leads to a better repartition of the pressure field for the lowest value of the journal speed. One can observes that, for a textured journal bearing, the positive effect of surface texturing especially on the carrying capacity is effective for the low shaft rotational speeds (80 rpm), an applied load of 667 N and lubricated with less viscous oil (oil 1, ISO VG 32).

![Figure 4](image-url)

**Figure 4.** Pressure versus circumferential angle at the middle of the bearing \((Z=1/2)\) for two speeds (80 and 2000 rpm) (a) for the two oils (b) for oil1 VG32 and (c) for oil2 VG100.

A higher significant hydrodynamic effect can be observed in the angular zone between 120 to 170° which also induces a significant decrease of the maximum pressure, from 14.48 MPa for the un-textured bearing configuration to 10.08 MPa for the fully texturing configuration, i.e. a reduction of 30%. For the oil 2 (ISO VG 100), the maximum pressure decreases from 9.12 MPa for the un-textured bearing configuration to 7.15 MPa for the fully texturing configuration, i.e. a reduction of 21.6%.
fact, the gain is as large as the journal eccentricity is high, i.e. the minimum film thickness is low. The partially textured bearing has a less effect and so, the reduction on the maximum pressure is 8.9% for oil 1 and 13.5% for oil 2, only.

4. Conclusion
This study shows that under a hydrodynamic lubrication regime, the improvement of the most important characteristics (friction coefficient and minimum film thickness) of a textured (dimpled) journal bearing depends strongly on lubricant viscosity and the shaft rotational speed. The type of texture is also very important.

The fully textured bearing has the best performance for the lowest rotational speed and the less viscous lubricant. The significant increase in minimum film thickness leads to the decrease in maximum pressure and to the reduction in friction, for the largest shaft eccentricities. This improvement could be also greatly influenced by the thermal effects (mainly because of a decrease of the lubricant viscosity and so, an increase in journal eccentricity) which will be taken into account in the future studies.

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