

Static Performance of Surface Textured Magnetorheological Fluid Journal Bearings

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ABSTRACT

Previous studies of journal bearings with artificial texturing on the bearing surface show potential benefits in certain cases. These benefits are usually focused on a specific operating area of the bearing, whereas under certain operating conditions the performance of the bearing is deteriorating due to the surface texturing. Gaining control over the viscosity of the lubricant may become a useful tool in order to take advantage of the surface texturing in a wider range of loads and journal velocities. One way to achieve this control is the use of magnetorheological fluid journal bearings. Magnetorheological fluids are solutions of iron based paramagnetic particles in conventional lubricant. Under the influence of an external magnetic field, these particles form chains, they hinder the flow of the lubricant and they ultimately alter its apparent viscosity. In this work the two techniques are combined in order to optimize the behaviour of the journal bearing in as much a variety of operating conditions as possible. Different shapes applied on the surface texturing will be examined.

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1. INTRODUCTION

The use of surface modification in order to achieve performance improvement in journal bearings has been the subject of numerous works [1,2]. The main issue with the application of surface texturing techniques in journal bearings is the fact that possible benefits are limited in a specific set of operating conditions. Lu and Khonsari [3] have shown that dimples may affect positively the journal bearing performance especially under boundary lubrication regime. On the other hand, the application of artificial surface texturing in

thrust bearings and parallel sliders has shown higher potential benefits in comparison to journal bearings as shown in [4,5].

Since the benefits from artificial texturing application occur under specific operating conditions [6], making the most out of this technique requires active control over the operating conditions themselves.

Magnetorheological fluid journal bearings offer the opportunity of control over the viscosity of the lubricant. Magnetorheological fluids consist of paramagnetic iron-based particles dispersed in

conventional lubricant. A magnetic field polarizes the particles, forces them into forming chains and changes the apparent lubricant viscosity. Magnetorheological fluid journal bearings have been the object of recent research [7,8] but the potential of using the magnetorheological fluids in conjunction with artificial texturing has not been investigated previously.

The simulation of the complex flow induced in a journal bearing with artificial texturing is a task of which Navier Stokes are better suited than Reynolds since inertia effects in this case are important [9,10]. This effect is even higher in the case of magnetorheological fluids due to their high density.

In this work the capacity of the magnetorheological fluids to take advantage of the artificial texturing in journal bearing is examined. The static performance of various configurations of artificial texturing is considered.

2. THEORY

The magnetorheological fluids exhibit a rheological behaviour which can be described by the Bingham rheological model.

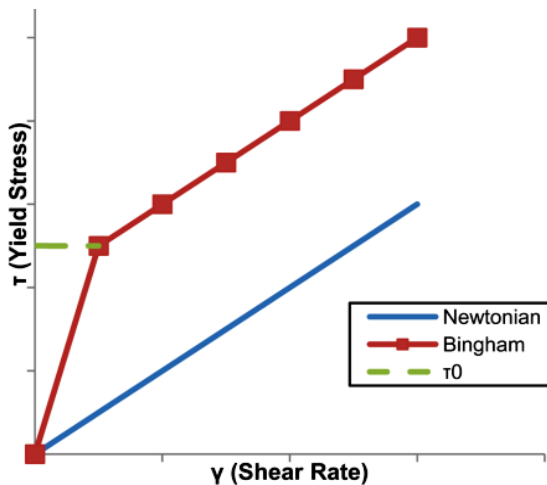


Fig. 1. The Bingham rheological model.

The Bingham viscosity model, as shown in Figure 1, describes a semi-solid material which flows under a certain value of shear stress. This behaviour describes the rheology of the magnetorheological fluid under the excitation of an external magnetic field. The apparent viscosity of the magnetorheological fluid according to Bingham fluid is described as follows:

$$\mu_a = \begin{cases} \mu_f + \frac{\tau_0(H)}{\dot{\gamma}}, & \dot{\gamma} > \frac{\tau_0(H)}{\mu - \mu_f} \\ \mu_p, & \dot{\gamma} \leq \frac{\tau_0(H)}{\mu - \mu_f} \end{cases} \quad (1)$$

where: μ_f is the Newtonian viscosity of the magnetorheological fluid, μ_p is the plastic viscosity, $\tau_0(H)$ is the yield stress under the influence of a field with intensity H and $\dot{\gamma}$ is the shear rate.

The calculation of the journal bearing performance is been accomplished using the continuity

$$\frac{\partial \rho}{\partial t} + \nabla(\rho \vec{v}) = 0 \quad (2)$$

and momentum equations:

$$\frac{\partial}{\partial t}(\rho \vec{v}) + \nabla(\rho \vec{v} \vec{v}) = -\nabla p + \nabla(\bar{\tau}) + \rho \vec{g} + \vec{F} \quad (3)$$

where: ρ is the fluid density, \vec{v} is the velocity vector, p is the pressure, $\bar{\tau}$ is the stress tensor and \vec{F} is the external forces vector. The basic geometry of the bearing considered in this work is depicted in Fig. 2.

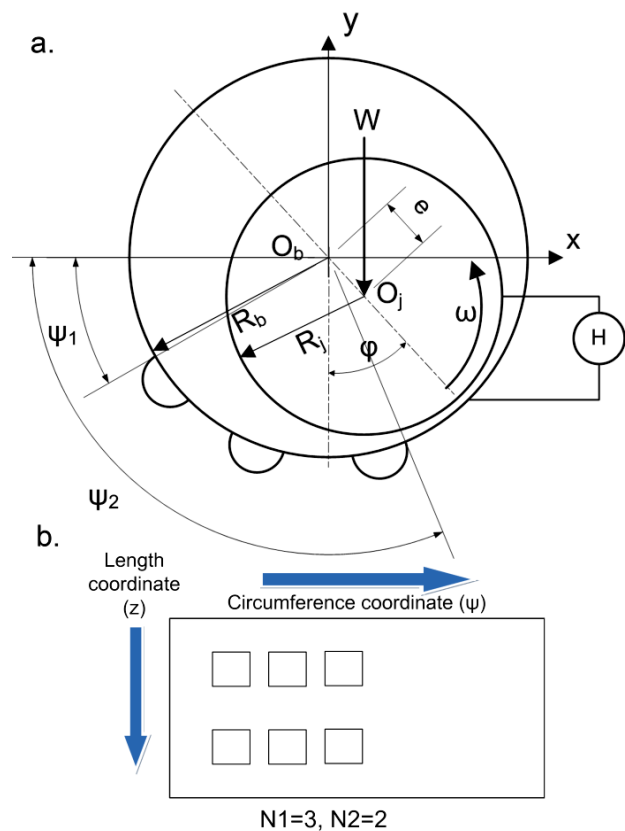


Fig. 2. Schematic of the journal bearing's geometry.

The bearing considered in the simulations has a radius $R_b = 49.999$ mm, and total length $L = 49.999$ mm. The radial clearance is $C = 85.5$ μm . The configuration of the texturing pattern is described by the number of dimples in circumferential and longitudinal directions (N_1 and N_2 , respectively). The bearing surface is only partially textured with the textured area being defined by the inlet and outlet angles ψ_1 and ψ_2 , respectively. In this work two shapes of dimples were examined: rectangular and egg-shaped dimples, as shown in Fig. 3.

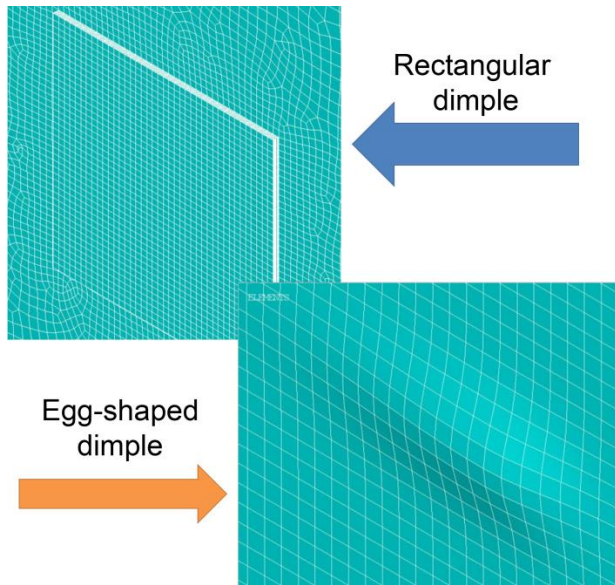


Fig. 3. The dimples' meshed models.

Rectangular dimples have a length of 1.3 mm and a width of 1.5 mm. Egg-shaped dimples where defined by the following equation:

$$r(\psi, z) = \begin{cases} R_b + d_{\text{dim}} \cdot e^{(2\sigma_1^2)} \cdot e^{(2\sigma_3^2)}, \psi < a \\ R_b + d_{\text{dim}} \cdot e^{(2\sigma_2^2)} \cdot e^{(2\sigma_3^2)}, \psi \geq a \end{cases} \quad (4)$$

where: d_{dim} is the dimple depth, a is the circumferential position of the dimple's center, b is the longitudinal coordinate of the dimple's center. The σ_1 and σ_2 parameters control the dimple overall length while σ_3 controls the dimple's width. In this work, $\sigma_1 = 1.2\text{E-}3$, $\sigma_2 = 6\text{E-}4$ and $\sigma_3 = 8\text{E-}4$.

3. RESULTS

In all cases examined in this work, the overall arc in which texturing is applied extends to 30 degrees and the dimples are uniformly

distributed within the given length of the bearing. A single configuration of $N_1 = 6$ and $N_2 = 6$ was examined for both shapes. The rotational velocity of the journal is set to 1000 rpm. The lubricant used has a density of 2950 kg/m^3 , yield stress of 25000 Pa and fluid (Newtonian) viscosity of 0.112 Pas. The journal bearing with egg-shaped texturing was modelled with 77220 hexahedral elements (89700 nodes). The journal bearing rectangular shape texturing was modelled with 260400 hexahedral elements (293088 nodes).

2.1 Validation

For the purposes of model validation, the results of the simulation of a plain bearing were compared with the results presented by Brito et al [11].

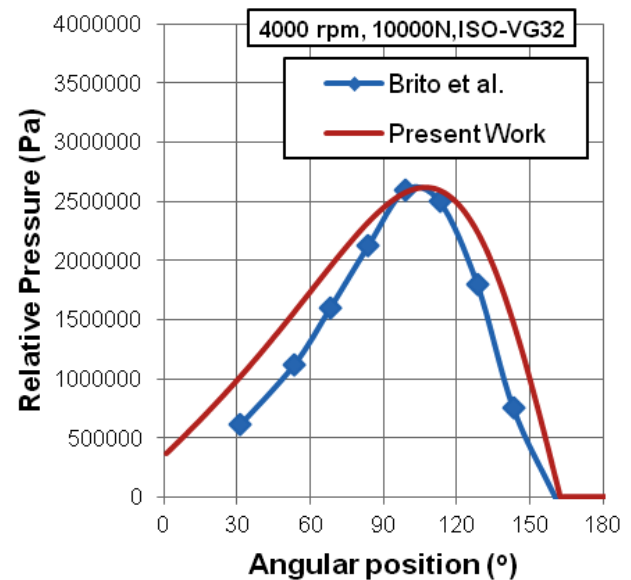


Fig. 4. Comparison of experimental with simulation results.

The bearing for which validation was performed has the same bearing radius and radial clearance as the ones used for the purposes of the simulations. The length is $L = 80$ mm and it was lubricated with ISO VG 32 lubricant. The viscosity of the lubricant was 0.0293 Pas at 40 °C.

2.2 Angular position of texturing

The angular position of the textured area, as shown in Fig. 5, has been investigated in order to establish the relationship between ψ_1 , load capacity and friction.

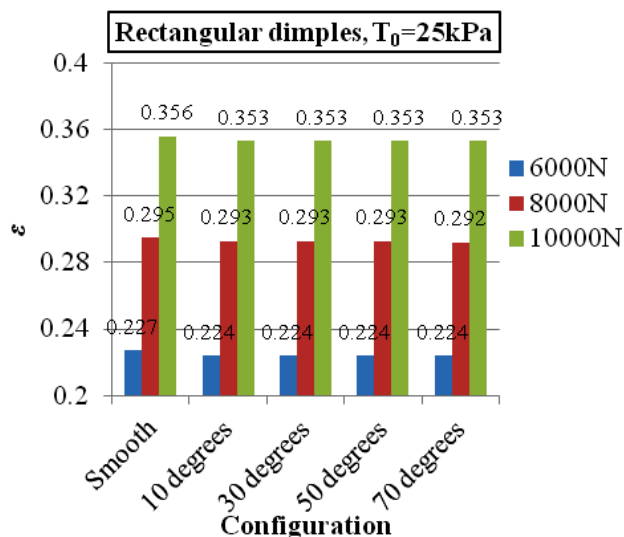


Fig. 5. The relative eccentricity for four different ψ_1 values with rectangular dimples for load capacity of 6000, 8000 and 10,000 N. The smooth bearing performance is present for comparison.

The results show minor improvement in terms of relative eccentricity with maximum improvement shown for all textured cases where the relative eccentricity drop reaches 1.32 % for the 6000 N load.

The egg-shaped texturing shows some significant improvement as depicted in Fig. 6.

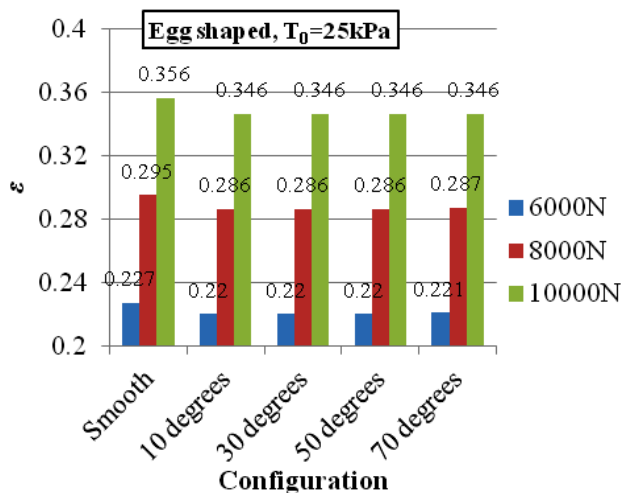


Fig. 6. The relative eccentricity for four different ψ_1 values with egg-shaped dimples for load capacity of 6000, 8000 and 10,000 N. The smooth bearing performance is present for comparison.

Although there is minor influence on the relative eccentricity of the journal bearing when the angular position of the textured area changes, there is a 3.08 % decrease in the cases between 10 and 50 degrees for a 6000 N load.

The friction coefficient is minimally affected by the angular position of the artificial texturing inside the bearing. In Fig. 7 the normalized friction coefficient is presented for a series of texturing circumferential position values. The results include the performance of the smooth bearing for comparison.

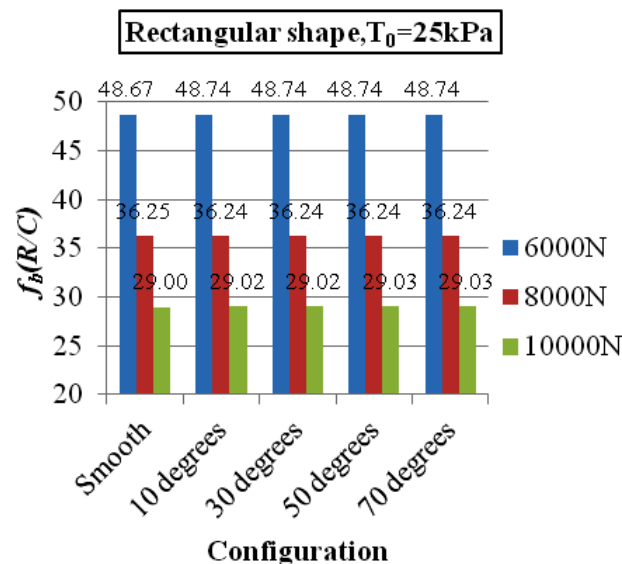


Fig. 7. The normalized friction coefficient for smooth and textured bearing with rectangular dimples for various texturing circumferential locations under 6000, 8000 and 10,000 N load.

The friction coefficient exhibits a maximum increase of 0.14 % in the case of 6000 N load. In the case of 8000 N there is a 0.05 % decrease but these changes are negligible. The same trend appears in the case of egg-shaped artificial texturing, described in Fig. 8.

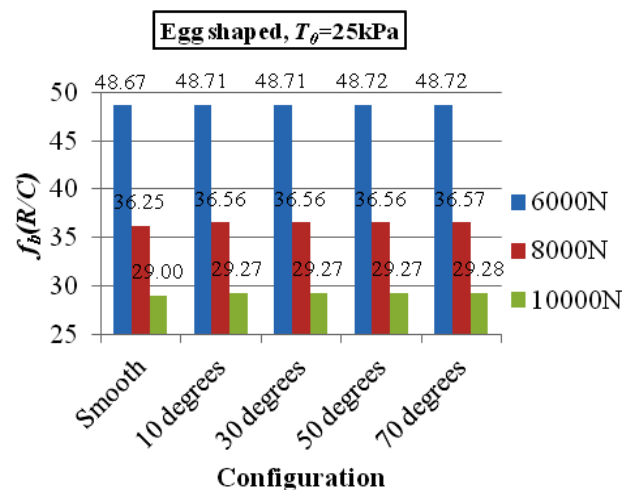


Fig. 8. The normalized friction coefficient of smooth and artificially textured journal bearing with egg-shaped dimples on four angular locations in the circumference of the bearing for loads of 6000, 8000 and 10,000 N.

The friction coefficient does not change significantly with the change of the dimple geometry. There is a maximum increase of 0.95 % of the friction coefficient for a load of 10,000 N. Overall there are negligible deviations from the values obtained for the smooth bearing.

2.3 Dimple depth

Another parameter of the overall geometry of the artificial texturing is the dimple depth. A comparison of the effect of the dimple depth on the relative eccentricity of the artificially textured journal bearing is presented in Fig. 9, for a load of 6000 N.

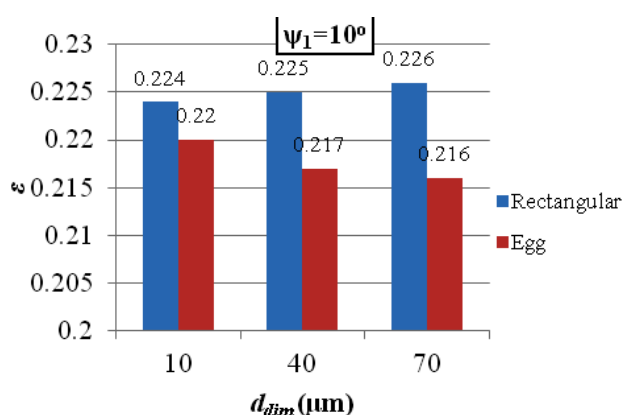


Fig. 9. The effect of dimple depth on the relative eccentricity for a load of 6000 N for both rectangular and egg-shaped dimples.

The difference of geometry between the two configurations induces different results on the performance of the artificially textured journal bearing. While the rectangular dimples depth increase has a negative impact on the relative eccentricity, the increase of the egg-shaped dimples depth improves the static performance of the bearing.

4. CONCLUSIONS

The effect of artificial texturing on the performance of the journal bearing is positive although the extent of this effect seems to be minor in absolute values. The egg-shaped texturing is although promising as the increase of depth resulted in relative eccentricity improvement of 4.8 % with a dimples density that is rather low, whereas the rectangular shaped texturing performance deteriorates. In other words there seem to be margins for

further improvement of the performance benefits that the specific artificial texturing geometry has to offer.

Acknowledgement

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