

Numerical and experimental investigation of texture shape and position in the macroscopic contact

A. Codrignani^a, B. Frohnäpfel^a, F. Magagnato^a, P. Schreiber^b, J. Schneider^b, P. Gumbsch^{b,c}

^a*Institute of Fluid Mechanics, Karlsruhe Institute of Technology, Karlsruhe, Germany*

^b*Institute for Applied Materials and μ TC, Karlsruhe Institute of Technology, Karlsruhe, Germany*

^c*Fraunhofer IWM, Freiburg, Germany*

Abstract

In this work, the influence of operating conditions on the shape parameters of surface texture is investigated by means of both numerical and experimental investigations. The analysed texture consists of micro-dimples obtained through laser surface texturing on a pin-on-disc configuration. From the numerical point of view, particular attention is paid to the faithful representation of the 2D surface of the experimental set-up and to modelling cavitation phenomena through a mass conserving algorithm. As results, the dimple depth shows a higher relevance than diameter in determining the optimal texture shape (both in terms of friction reduction and load carrying capacity). Moreover, the dimple depth, corresponding to the minimal friction, is found to scale with the square root of the Sommerfeld number in agreement with the experimental results. Finally, it is found that a numerical approach with the present hydrodynamic model cannot account for friction reduction obtained experimentally with different orientation of the texture.

Keywords:

surface texture, optimal texture parameter, numerical and experimental analysis, mass conserving cavitation

1. Introduction

Using textured surfaces is a widely spread stratagem in nature in order to improve specific performance in the interaction between surfaces and their surrounding environment [1]. Taking inspiration from this fact, a huge interest has emerged in the last decades on the applications of such surfaces for tribological purposes [2]. The great potential of such engineered surfaces was tested in the early works of Hamilton [3], Anno [4] and subsequently with experimental and numerical investigations by Etsion's group for various kinds of industrial applications such as parallel sliders [5], mechanical seals [6] and piston rings [7, 8].

Among the unlimited ways to realize surface textures, non-communicating textures like grooves and dimples have drawn most of the attention, thanks to the great improvement of Laser Surface Texturing techniques (LST) [2, 9]. In the struggle to identify the condition under which dimpled surfaces bring actual benefits, the following main mechanisms have been detected. In the boundary lubrication regime, dimples can reduce static friction mainly thanks to a contact area reduction [10]. In mixed lubrication, dimpled surfaces can better entrap debris, hence reducing the wear by minimizing the third-body abrasion [11, 12]. Their ability to act as a lubricant reservoir improves the contact wettability under starved lubrication [13]. Moreover, dimples can shift the transition from the mixed lubrication regime to the hydrodynamic one to lower velocities [14]. Finally, regarding the hydrodynamic regime, dimples are responsible for a pressure build-up which consequently leads to a reduction of the tangential stress through a thickening of the fluid film and thus to a reduction of the overall friction coefficient.

From the numerical point of view, the underlying mechanisms of the above mentioned effects have been extensively studied over the last years for various applications. In case of low convergence bearings, dimples are deemed to be effective thanks to the hydrodynamic lift which results from the asymmetrical pressure distribution when cavitation [15] occurs. The physical mechanism behind this pressure build-up has been explained by Fowell *et al.* as an "inlet suction" effect due to the reduced pressure in the dimples [16]. More generally for other kind of geometries, the effects introduced by dimples can be interpreted in terms of density changes or by considering the coupling with thermodynamics [17]. Non-linear effects also play a role in generating a non-symmetrical pressure distribution, which may

lead to beneficial as well as detrimental effects, as found in many works [18, 19, 20]. For more complex geometries, an explanation of the pressure build-up has been provided by Cupillard *et al.* by analysing how energy is transferred from the moving wall to the fluid and converted into pressure at the beginning of the texture [21].

The difficulty in the thorough comprehension of the physics behind textured surfaces is complicated by the large variety of texture design parameters such as texture location, pattern and density, as well as details of the texture shape (*e.g.* dimple depth and diameter). In order to cast light on the design and optimization of dimples many studies adopted a systematic approach in describing the influence of each of the above mentioned design parameters on the overall performance of typical industrial applications. Different texture shapes are analysed by Adjemount *et al.* [22], who show that cylindrical and spherical dimples have the most positive influence. The role of partial texturing is investigated, among others, by Fillon's group, showing that a partial texturing can lead to an increase in load carrying capacity if compared to a fully textured case [23, 24]. In respect to texture density, a disagreement existed originally between numerical and experimental analysis, since most of numerical works [25] overestimated the experimentally determined optimal value found between 10% and 20% [26]. An explanation to this discrepancy is proposed by Wang *et al.* [27] by considering the influence of roughness and contact mechanics.

Among the above mentioned design parameters, the diameter and the depth of dimples have risen the biggest interest in the research community. Numerical analysis revealed the importance of texture depth on texture performances [28, 29]. In particular, Ramesh *et al.* [30] and Fowell *et al.* [31] present a direct correlation between dimple depth and gap height for a 2D geometry and point out that the optimal depth increases with higher viscosity. A systematic experimental investigation of this trends is carried out by Braun *et al.* [14] with a pin-on-disk set-up. In this work, different dimple diameters ranging from $15\mu\text{m}$ to $800\mu\text{m}$ are considered at constant texture density and depth-to-diameter ratio. The results, based on various sliding velocities and viscosities, indicate that the Stribeck curve of the optimal texture scales with the Hersey number $\frac{\mu\Omega}{W}$, where μ is the viscosity of the lubricant, Ω the rotational speed and W the average contact pressure. With respect to the influence of the texture pattern and the optimal orientation angle on friction reduction, experimental and numerical studies come to different conclusions [26, 32].

In this work we intend to deepen the physical understanding of the experimental results from Braun *et al.* [14] by numerically investigating the same geometry of the experiments. The numerical approach allows to analyse the scaling of optimal dimple parameters with respect to the operating conditions; particularly viscosity, velocity and gap height. At first we focus on the numerical and experimental investigation of the sensitivity of the load carrying capacity with respect to the position of the dimple on the macro-geometry. After this prior analysis an exhaustive parametric study of a 2D textured surface as extension of the 1D works of Fowell and Rahmani [31, 33, 34] is performed. Lastly, we discuss the effects of the texture orientation based on a comparison of numerical and experimental results.

2. Numerical approach

Among the different effects that textures have on tribological performance we consider from the numerical point of view only those which are related to the hydrodynamic regime; an approach taken in the vast majority of literature on this topic [2]. In contrast to some previous studies with 1D textures [31, 34, 35], a realistic 2D surface, which corresponds to the one employed in the corresponding experiments, is considered for the numerical parameter studies in the present work. In order to enable such a parametric study at reasonable computational cost, particular attention is paid to an efficient numerical implementation.

2.1. Governing equation

We model the shear flow of a Newtonian lubricant between two sliding walls through the incompressible Reynolds equation. The employed equation considers also cavitation phenomena through a mass conserving algorithm as presented by Woloszynski *et al.* in [36]

$$\nabla \cdot (h^3 \nabla p - 6\mu \vec{V} h (1 - \theta)) = 0 \quad (1)$$

where $h(x, z)$ describes the gap height distribution, μ is the dynamic viscosity, $\vec{V} = \{U, W\}$ is the upper wall velocity. The cavity fraction θ is defined through a reference density ρ_{ref} as follows:

$$\theta = 1 - \frac{\rho}{\rho_{ref}}. \quad (2)$$

number of cells in each direction	normal force [N]	tangential force [N]
65	1178.379057	5.029480
129	1188.524679	5.037692
257	1193.755952	5.045461
513	1196.401560	5.049900
1025	1197.470182	5.051418
2049	1198.008821	5.052289
4097	1198.288155	5.052802

Table 1: Convergence of the Reynolds solver with respect to the normal and tangential force. The results corresponds to the textured pin geometry A with a dimple diameter $D = 40\mu m$.

The pressure p and the cavity fraction θ satisfy the complementarity constrain $(p - p_{cav})\theta = 0$, in which the relative pressure $p - p_{cav}$ and the cavity fraction θ are always positive. The cavitation pressure is kept at a realistic constant value of $P_{cav} = 80000Pa$, in order to keep the investigated parameter space in a reasonable dimension. The flow is considered isoviscous and isothermal.

As shown in [36] and subsequently also in [37], equation (1) can be directly coupled with the complementarity constrain in a non-linear unconstrained system, whose solution requires only few steps of the Newton-Raphson scheme, making this algorithm tremendously faster than other traditional approaches based on constrained solution of the Reynolds equation (such as, for example, the $p - \theta$ algorithm by Elrod and Adams [38]). For the present work, the non-linear unconstrained system is iteratively solved until the residuum on pressure and cavity fraction drops below 10^{-6} .

The Reynolds equation is discretized with a finite volume method, which is based on its weak formulation [39], allowing to increase the solution precision in the presence of high geometry discontinuities like in the considered set-up. Moreover, the finite volume method is a good compromise between computational performance and easiness of implementation [18], the achieved convergence for one of the test cases is presented in table 1.

2.2. Geometry description

The geometry of the untextured pin is extracted from the experimental set-up based on optical profiler measurements of the employed pellets. It should be noted that the experimental campaigns (see section 3) employed two different pin geometries which are both considered numerically. The 3D numerical representation of the first pin (A) is shown in figure 1 for a case in which a computer generated partial texture is imposed on the surface topography of the untextured pin. In the experiments, the pellets are textured with LST resulting in very similar shapes as the dimples that are considered numerically [14, 26]. The surface of the pin is not perfectly flat, but presents a perceptible curvature which is shown through a magnification in figure 2 which corresponds to a cut through the computational domain at $z = 0$. This curvature is the main cause for the creation of a pressure distribution along the gap which is perturbed by the presence of the texture. We define the reference gap height H in the centre of the pin surface where the distance between the two surfaces has its minimum. The second pin (B) is also cylindrical, but its surface is much flatter ($< 0.1\mu m$) than the one of pin A. The average profile of pin B is shown through a further magnification in figure 2.

A constant ambient pressure $p|_{\partial A} = p_{amb}$ is prescribed at the domain boundaries (see figure 2) where the cavity fraction is set to $\theta|_{\partial A} = 0$ since the whole pellet is submerged in oil in the pin-on-disc experiments. In conformity to the experimental set-up, the height of the edge of the pin is set to $2.5mm$. In this way the inlet height h_{in} is at least one order of magnitude larger than the gap height between the pin surface and the upper wall. This leads to a flat pressure distribution outside the pin surface and thus the pressure distribution over the pin surface becomes independent from the boundary conditions [21]. The upper wall moves in the x -direction with velocity U . A comparative analysis of the untextured case with both Navier-Stokes and Reynolds equations has shown that inertial effects are negligible despite the gap height discontinuity at the edge of the pin.

The texture consists of spherical dimples which can be fully characterized by two parameters: diameter and depth. The distance between the center of two adjacent dimples can vary according to their diameter in order to keep the

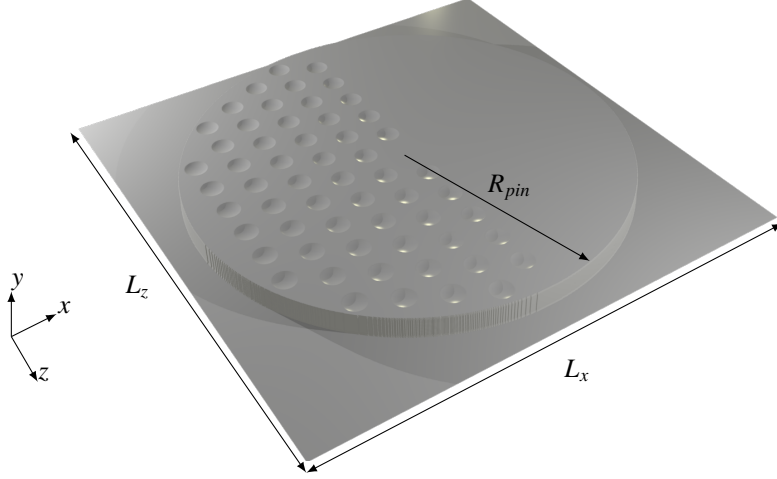


Figure 1: Representation of the geometry of pin A. The partial texture shown here corresponds to a dimple diameter $D = 400 \mu m$, $Depth = 40 \mu m$ and a surface density of $\rho_{tex} = 10\%$. The surface is in relative motion to the upper rotating disc so that a mean flow in positive x -direction is generated in the gap. In the numerical simulation, only the first half of the surface is structured with dimples for reasons discussed in section 4.2.2. The domain size is $L_x = L_z = 9mm$ while the radius of pin A is $R_{pin} = 8mm$.

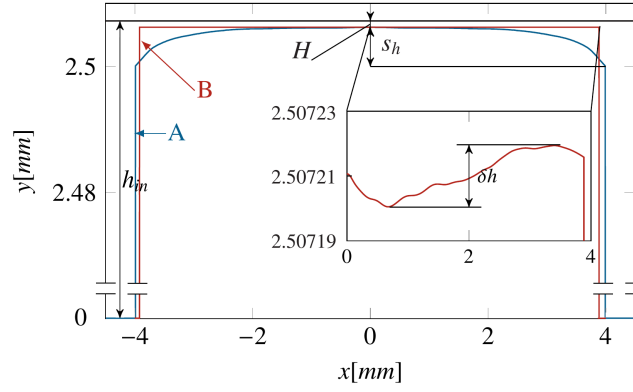


Figure 2: Profile of the pins as measured from the experimental set-up, please note that the y axis is 1000 times magnified. The gap height H is defined at the center, where the distance between the two plates is minimal. A further magnification of the surface profile of pin B is provided, axis values are expressed in mm , the maximal measured height variation is $\delta h = 0.047 \mu m$. The domain boundaries are located at $(x, \pm \frac{L_z}{2})$ and $(\pm \frac{L_x}{2}, z)$.

texture density at the constant value $\rho_{txt} = \frac{A_{txt}}{A_{pin}} = 10\%$. This value of texture density corresponds to the one used in the experimental campaigns (see section 3).

Thanks to the good convergence (see table 1) a mesh of 2049x2049 can be considered for all the results presented in this work, allowing a maximal resolution of $4.3 \mu m$ in both x and z direction with a computational time in the range of some minutes. The convergence rate reported in table 1 corresponds to the geometry of pin A with dimples of diameter $D = 40 \mu m$ and $Depth = 4 \mu m$. This case is the numerically most challenging since the small dimple size requires the highest resolution. Similar convergence results are obtained also with pin B. Note that, although a first order spatial convergence is achieved by the chosen cavitation algorithm, a very low number of iterations (around 15) is required, such that the overall computational effort is limited to a minimum.

3. Experimental setup

Three experimental campaigns are considered in this work. The results of the first two are already published [14, 26], while the third one, concerning the investigation of texture positioning, is reported here for the first time.

All above mentioned tribo-experiments are pin-on-disc set-ups, realized on two different tribometers. The experiments presented in [14] and [26] were conducted on a Plint TE-92 HS tribometer (Phoenix Tribology, Kingsclere, UK) sketched in figure 3. This apparatus consists of a rotating disc with 70mm diameter, upon which a pin (8mm, height 2.5mm) is pressed with a normal force of 150N. The tribo-contact was submerged in an additive free PAO-18-oil bath at controlled temperatures during the tests. The discs were made of hardened and tempered steel 100Cr6 (AISI 5210), with hardness values of about 800HV. The roughness of the fine-ground discs varied in the range from $Ra = 0.080$ to $0.120 \mu m$. The pins were mounted in a self-aligning holder and pressed on the disc from below at sufficient distance from the center of rotation, in order to reduce the velocity gradient effects [40]. In these first two experimental campaigns [14, 26], the pin was made of normalized steel C85 with homogeneous pearlitic grain structure (400HV) and the surface was ground and subsequently polished using a $3 \mu m$ and $1 \mu m$ diamond suspension. The pin geometry employed in the first two experimental campaigns correspond to pin A shown in figure 2.

The third experimental campaign was conducted on a CSEM pin-on-disc tribometer (CSEM, Peseux, Switzerland). While the discs used in these experiments were prepared as described above, pins made of cemented carbide (WC-Ni, 1450HV) were employed. These pins were fine-ground, resulting in very low waviness of the surfaces with roughness values of $Ra = 0.025$ to $0.035 \mu m$ (see pin B in figure 2). The experiments on the CSEM tribometer were conducted at room temperature ($20^\circ \pm 2^\circ C$) and a low viscous automotive Shell V-Oil1404 was employed. According to the manufacturer, the kinematic viscosity at $20^\circ C$ is $3.8 mm^2/s$ while the density at $15^\circ C$ is $826 kg/m^3$. The normal force was kept constant at 15N, applied via dead weights. The sampling of friction values for different sliding speeds was realized by a stepwise reduction of the rotational speed of the disc, for a total number of nine speed steps. Each step was hold for 300 seconds and the whole test sequence was repeated five times during one experiment. In order to eliminate run-in effect, only the last three data sets are considered in the data evaluation.

Experiments for each test configuration were repeated at least twice with a fresh pin and disc in order to ensure the reproducibility of the measurements. In all presented cases, the dimpled surfaces for the experiments were obtained through laser surface texturing [14]. Particular attention was paid to the removal of laser ablation debris by means of an additional polishing procedure or grinding step for the steel and cemented carbide pins, respectively.

4. Results and discussion

Both experimental and numerical investigations are based on a systematic variation of the design parameters and operating conditions with the goal to identify combinations of operating conditions and texture design which lead to the minimal friction or the maximal load carrying capacity. Similar studies in literature, which carry out such systematic investigations, mainly describe either 1D slider bearings [31, 6, 33, 41], 1D or 2D parallel thrust bearings [34, 22, 42, 43], 2D seals [44] or 2D journal bearing [45, 46]. In contrast, the present work focuses on a pin-on-disc tribometer and its 2D numerical representation. To the authors' knowledge such a numerical parametric study of a 2D surface of the pin of a pin-on-disc tribometer is not yet available in the literature.

The obtained results are grouped in four sections in the following. First, the new experimental results are introduced along with the ones, which were already presented [14, 26]. Second, the result of a preliminary study is

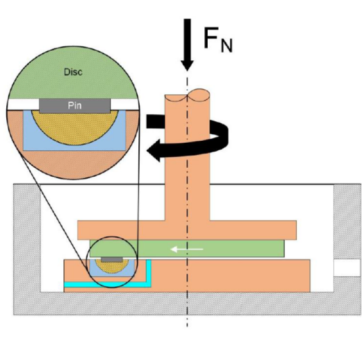


Figure 3: Experimental set-up of the Plint TE-92 HS tribometer form Phoenix Tribology, from Braun *et al.* [14]

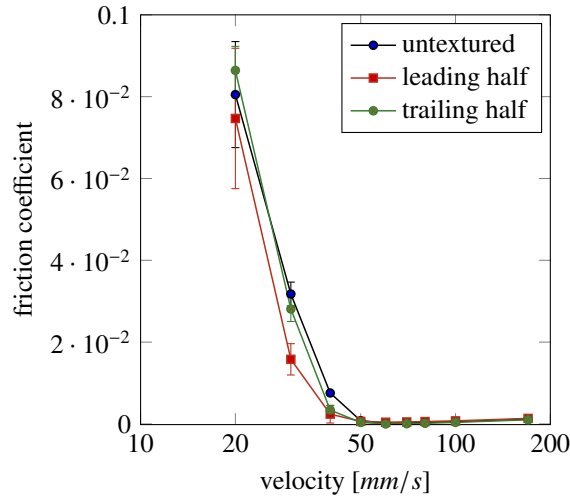


Figure 4: Experimental Stribeck curve of the partially textured surface in comparison to the untextured case. Both textured cases have the same dimple shape and texture area as in the numerical simulations shown in figure 11.

presented, which addresses the differences between the experimental set-up and its numerical representation. Subsequently, the scaling of the optimal dimple shape parameters is analysed with respect to the operating conditions. Lastly, the influence of different texture arrangements is considered and the related contradictions between experimental and numerical findings are discussed.

4.1. Experimental results

The first aspect of the experimental campaign analysed in this work concerns the influence of partial texturing in order to investigate, in parallel with the numerical analysis, which part of the pin surface delivers the best performance if textured. Two different extensions of the textured area are considered here: partial texturing on the front part of the pin and on the rear part. Figure 4 shows the experimental Stribeck curve obtained with the three texture extensions and the untextured reference case. The texturing of the leading half of the pin clearly leads to a friction reduction in comparison with the untextured case and represents a better improvement than the case with texture on the trailing half. This result is in agreement with previous studies in literature [35, 47, 48, 49] and is numerically further analysed in section 4.2.2. The experimental results reveal the biggest difference between the textured and the reference curves in the mixed lubrication region. In the hydrodynamic region of the Stribeck curve low signal-to-noise ratio does not allow to identify appreciable differences in the friction coefficient.

The second aspect of the experimental results considered in this work concerns the study of Braun *et al.* [14] about the effects of temperature (*i.e.* viscosity) on the determination of the dimple diameter which leads to the lowest

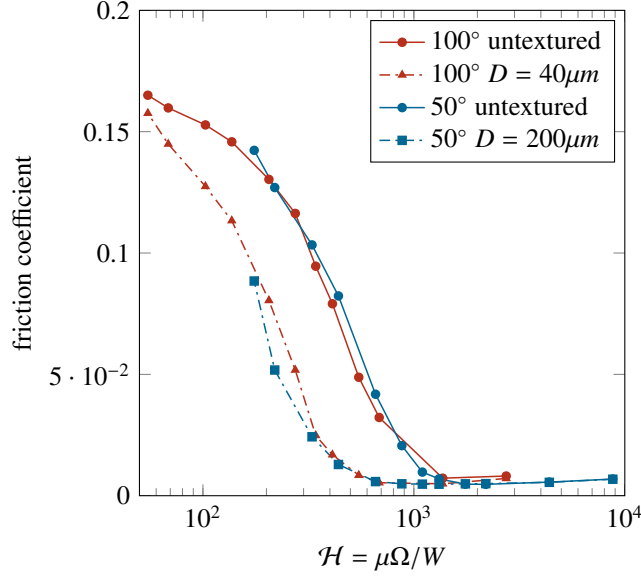


Figure 5: Experimental Stribeck curve as function of the Hersey number at two different temperatures, comparison between two untextured cases and the corresponding ones with optimal texture. Adapted from [14].

friction coefficient. In this experimental study, the Stribeck curve of pellets with different dimple diameters (ranging from $15\mu\text{m}$ to $800\mu\text{m}$) were tested at two different temperatures, $T = 50^\circ\text{C}$ and $T = 100^\circ\text{C}$, under the constrain of constant normal load. Figure 5 shows the experimental Stribeck curves from [14] for the untextured and the optimally textured cases at two different temperatures. The dimple shape is spherical and aspect ratio is kept constant to $\lambda = \text{Depth}/D = 0.1$. It is interesting to note that the experimental curve obtained with the optimal texture scales very well with the Hersey number $\mathcal{H} = \mu\Omega/W$, where μ is the dynamic viscosity, Ω the rotational speed of the upper disc in the pin-on-disc tribometer and W is the normal load. The numerical comparison with this experimental analysis is addressed in section 4.3.

The last aspect based on [26], concerns the impact of the dimple arrangement. The dimple pattern was changed by imposing a progressive shift s_z in the rows of dimples so that the overall pattern could change between a quadratic one and a pseudo hexagonal one. This shift can be depicted in terms of the angle between dimples from two consecutive rows as shown in figure 6. The measured Stribeck curve is shown in figure 7. As can be seen, for pseudo hexagonal disposition of dimples, $\alpha = 60^\circ$ (i.e. $s_z/l_z = 0.5$), the textured surface shows a higher friction reduction compared to the case with $\alpha = 45^\circ$ and $\alpha = 55^\circ$. Only the angle between consecutive rows of the dimple distribution is changed, while all other dimple design parameters such as diameter and depth are kept constant. In particular, the texture density is fixed to $\rho_{\text{txt}} = 10\%$, the dimple diameter to $D = 40\mu\text{m}$ and the depth to $\text{Depth} = 4\mu\text{m}$.

4.2. Preliminary considerations of the numerical representation of the experiments

The typical pressure and cavity fraction distributions over the textured pin surface displayed in figure 1 are shown in figures 8a and 8b. In the first half of the pin surface the convergent height distribution induces a consistent increase of pressure which subsequently reduces in the second half where cavitation occurs and therefore density diminishes. The pressure distribution is flat in the proximity of the domain boundaries, indicating that the domain is big enough in order to have a solution which is independent of the boundary conditions.

4.2.1. Comparison between 1D and 2D textured surface

Despite the increase in computational cost, we based our numerical analysis on the realistic 2D surface of the experimental set-up. This choice stems from the rationale that 1D textures, although extensively studied, lead to pressure distributions which can be misleading if compared to 2D results. Figure 9 shows the net pressure distribution in the centerline of the pin due to the placement of a single dimple for both 1D and 2D simulations. The pressure

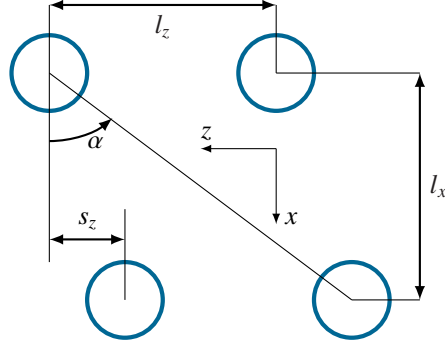


Figure 6: Schematic representation of the dimple pattern. The upper wall slides in positive x -direction. The distances between dimples in x and z -direction are set to $l_z = l_x$ so that a constant texture density $\rho_{txd} = 10\%$ is kept for each value of s_z . When $s_z = 0$ ($\alpha = 45^\circ$) the square pattern is obtained. The pseudo-hexagonal pattern corresponds to $\alpha = 60^\circ$ and $s_z = L_z/2$.

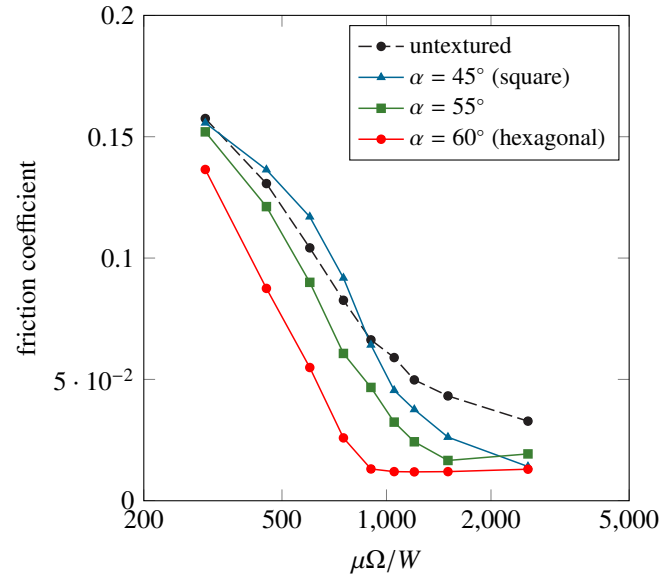


Figure 7: Experimental Stribeck curve as function of the Hersey number with different orientations of the textured surface. Dimple aspect ratio $\lambda = 0.1$, $Diam = 40 \mu m$, $\rho_{tex} = 10\%$. The experiments were run with the same specimens as in [14] but with the CSEM tribometer and the load/speed parameters of the newest experiments.

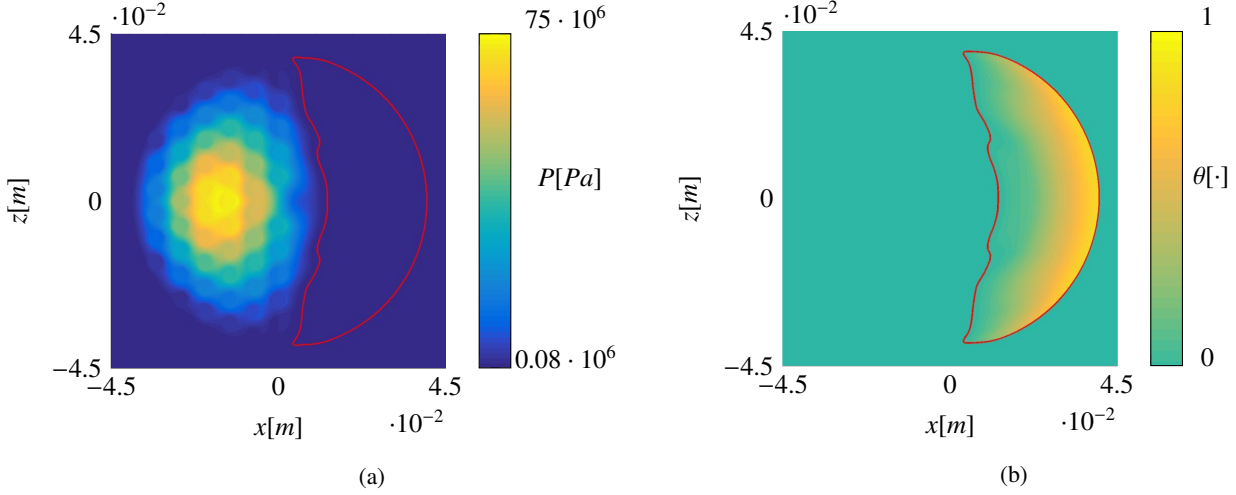


Figure 8: Pressure (a) and cavity fraction (b) distribution over the textured pin, $D = 400 \text{ } [\mu\text{m}]$, $\text{Depth} = 40 \text{ } [\mu\text{m}]$, $\rho_{\text{LST}} = 10\%$. In both figures the cavitation region is delimited by the red line. The cavity fraction, defined in equation 2 is non zero in most of the divergent part of the pin surface, where cavitation occurs. The upper wall slides from left to right with velocity $U = 1 \text{ m/s}$, viscosity $\mu = 0.1871 \text{ Pas}$ and the gap height $H = 1 \mu\text{m}$.

perturbation due to the presence of the dimple clearly propagates differently in both cases. In particular, the 2D pressure perturbation is smaller and decays faster than in the 1D case. The pressure perturbations disappear in the rear region of the pin where cavitation occurs. This difference might be attributed to the fact that in the 1D case the texture does not actually represent a spherical dimple but rather a channel perpendicular to the sliding direction. Moreover, the presence of a pressure gradient in the z -direction in the Reynolds equation additionally smooths the pressure distribution.

Previous analyses concerning the influence of the number of considered dimensions were carried out in literature [50, 51, 52]. However, only the differences between the 3D and 2D Navier-Stokes approach for single texture elements was investigated, coming to the conclusion that little or no difference is noticed by switch from 3D Navier-Stokes results to 2D ones. The present case requires the use of the 2D Reynolds equation (see Eq.1).

Figure 10 shows the normalized net pressure distribution due to the presence of a single dimple on the pin surface (geometry A). As one can see, the influence of the single dimple decays very fast and becomes smaller than 1% within 10 diameters from the center of the dimple. This implies that in realistic 2D surfaces, dimples influence a restricted area, whereas in 1D simulations the corresponding pressure perturbations can reach the domain boundaries.

4.2.2. Setting of the numerical parametric study

Numerical simulation should ideally mimic the same operating conditions under which the tribological tests are carried out. In the experiment, the employed tribometer runs under a constant normal force until it converges to a steady state with an unknown gap height. In contrast, the Reynolds equation requires a prescribed height distribution as input and delivers the value of normal force as output. Simulations with prescribed constant normal force can be carried out iterating on the gap height until the equilibrium solution is found. However, this procedure is computationally expensive and thus difficult to realize for parametric studies.

Moreover, as we can see in figure 3, the pin is mounted on a support which can automatically pitch in order to ensure the ideal formation of a flat-on-flat contact during the tribological tests. This feature of the tribometer implies that an additional parameter should be considered in the simulations, since the asymmetry in the pressure distribution generates an angular momentum which might make the pin pitch. The pitch angle becomes then the results of an additional bisection algorithm nested into the one on the gap height. The consideration of both loops would represent a significant increase in the computational costs which would make a parametric study with 2D textured surfaces infeasible.

These two additional degrees of freedom of the geometry, namely the gap height and the pitch angle, are deemed to play also a role together with the extension of the texture on the pin surface. As a matter of fact, the experimental

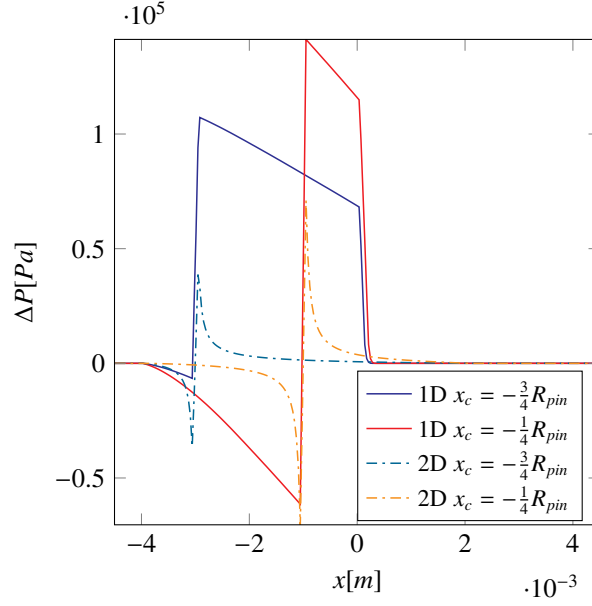


Figure 9: Comparison between 1D and 2D centerline net pressure distribution $\Delta P = P_{txt} - P_{untxt}$ due to the presence of a single dimple. Two different dimple positions are considered for completeness. These simulations are carried out with the following parameters: $H = 10\mu m$, $\mu = 0.18Pas$, $U = 0.1m/s$.

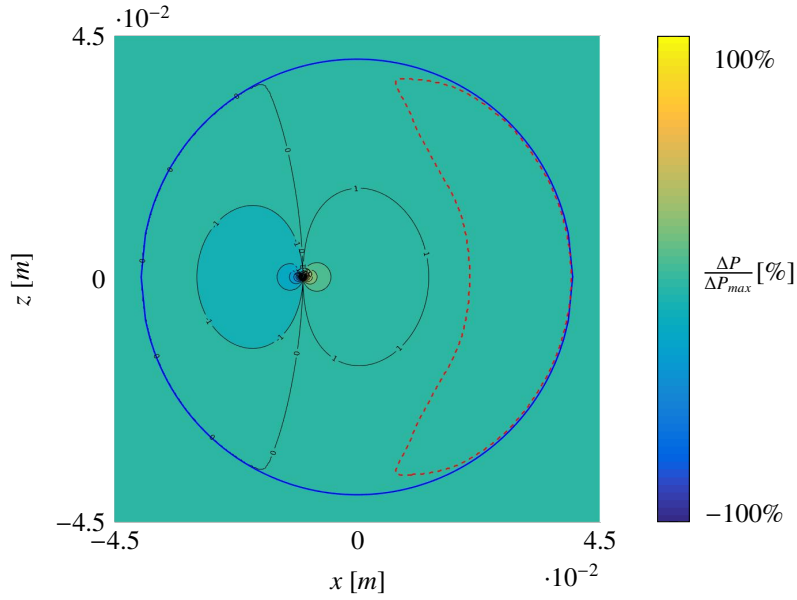


Figure 10: Influence area of a single dimple in terms of normalized net pressure distribution. The contour lines represent the normalized net pressure distribution computed as $\frac{\Delta P}{\Delta P_{max}}$ whereas $\Delta P = P_{txt} - P_{untxt}$. The red dotted line encloses the region in which cavitation occurs. This figure is obtained with the same parameters as in figure 9 for the case with $x_c = -\frac{3}{4}R_{pin}$.

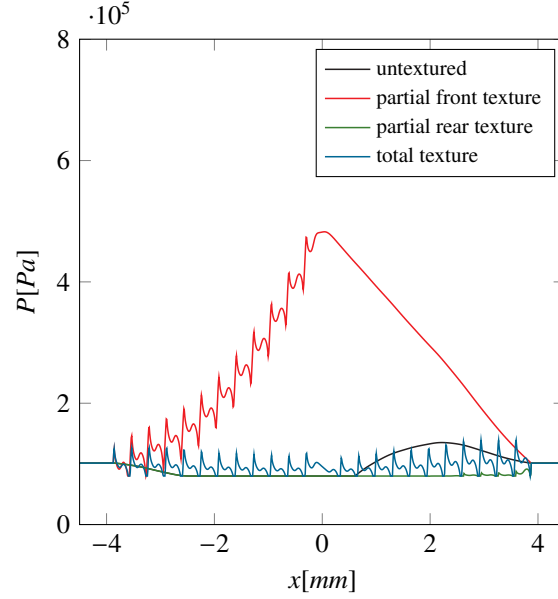


Figure 11: Centerline pressure distribution for four cases: front and rear partial texture, full texture and untextured as reference. These results are obtained with the pin geometry B. All textured cases have the same dimple shape, with $D = 60\mu\text{m}$ and $\text{Depth} = 4.5\mu\text{m}$. Texture density is 10% of the pin area. The gap height is set to $H = 10\mu\text{m}$, $U = 0.1\text{m/s}$, $\mu = 0.18\text{Pas}$

results presented in [14, 26] are obtained with fully textured pins, while numerical analysis mostly leads to the result that only a partial texturing is effective in improving the tribological performances [23, 24, 53]. This apparent difference between experiments and numerics is probably related to the fact that experiments are carried out in the mixed lubrication regime, while our numerical approach describes the tribological interaction from the hydrodynamic point of view only.

The effect of partial texturing in the numerical simulations is discussed in the following. Figure 11 shows the numerical results for different texturing of the pin surface and their comparison with the untextured reference case. As one can see from the pressure distribution in the centerline, only the partial texture of the front part can increase the load carrying capacity, whereas the fully textured case and the one with a texture only on the rear half present a pressure distribution which is lower than the untextured case, hence a lower load carrying capacity. This has implications also on the friction coefficient, since, in most of the cases, a higher load carrying capacity is directly related to lower friction values. The improved behaviour of partial front texture is found numerically for all operating parameters tested in the present study, and is confirmed experimentally as shown in section 4.1 and figure 4. Although these results are obtained in case of a geometry based on a pin-on-disc tribometer, the particular flatness of the pin (pin B shown in figure 2) makes the pressure distribution similar to the one of a parallel bearing. For such a geometry, several studies in literature have confirmed that a partial front texture has a much better impact on the friction reduction than full or rear texturing [47, 48, 49].

In the experiments, the partial rear textures shows for some velocities a lower friction coefficient as seen in the numerical simulations. This could be due to the fact that the numerical simulations do not consider the probable inclination of the pin as well as mixed lubrication phenomena. In the following numerical simulations, only the case with partial texturing in the front half of the pin is numerically considered and we leave the analysis with self-aligning pins to further studies.

4.2.3. Sensitivity to positioning location of single dimple

In this section we assess the sensitivity of the pressure distribution with respect to the dimple position on the surface, in order to understand how a single dimple interacts with the macro-geometry. This is particularly important in view of the consideration expressed in section 4.2.1 about the propagation of the pressure perturbations due a dimple on 2D surfaces. For such analysis the position of a single probing dimple is varied in order to establish on which part

of the surface the overall normal force is increased. Furthermore, such simulations are carried out for different values of operating parameters (viscosity, gap height and velocity of the upper wall) as well as design parameters such as diameter and depth.

The variation of operating parameters can be represented in a more generic fashion through the Sommerfeld number. The Sommerfeld number is derived as the relevant non-dimensional parameter when equation 1 is written in dimensionless form. It is given by

$$S = \frac{\mu UL}{H^2 W_r} \quad (3)$$

where μ , U , L are the viscosity, wall velocity and length, respectively; H is the gap height and W_r the load applied on the upper surface. In this analysis the reference length L corresponds to the domain length and therefore is kept constant; the reference load W_r is the one of the untextured case, since the presence of a single dimple affects the load by an almost negligible amount. Therefore, the other three parameters can be varied in order to study their impact on the pin regions in which the presence of a dimple leads to an increase of the load carrying capacity.

Figure 12 shows the isolines of the net normal force for different positions of the probing dimple on the pin surface. For symmetry reason only the lower half of the pin is shown. The shape of the probing dimple is kept constant in this first case with $D = 100 \mu m$ and $Depth = 10 \mu m$. Twelve different isolines are depicted in three coloured groups, each group represents a different gap height ranging from 1 to $10 \mu m$, while each line style portrays a different value of the Sommerfeld number S obtained through different combination of velocity and viscosity.

The contour of the area with positive net normal force lies always in the first half of the pin surface. This suggests that, in case of full texture, only the front part would contribute in generating a positive lift, while dimples located in the rear part would have a detrimental effect. In particular, in the rear half of the pin, dimples have almost no influence since that region is dominated by the cavitation pressure. This finding is again in agreement with previous studies, which have pointed out how partial texturing can be more effective than a fully textured case [23, 24, 47, 49, 48]. Moreover, in case of curved geometries, dimples are deemed to become more effective when placed close to the center of the geometry where the pressure reaches its maximum [21].

Isolines with same gap height but different values of the Sommerfeld number (hence with different viscosity and velocity) almost collide, while those with different gap height show a more prominent distance between each other. This hints to the fact that gap height plays a more important role than viscosity and velocity, meaning that the area of positive normal force is extremely sensitive to changes of the distance between the two walls. In other words, if the changes in the Sommerfeld number are simply due to a different combination of viscosity and velocity, the area of positive normal force will remain almost unaffected. Furthermore, for small values of gap height, the area in which dimples have positive effects shrinks considerably. This trend is likely to be related to the increasing relative gap height variation (slope) with reduced gap height. A similar trend has been observed in slider bearings by Murthy *et al.* [41], where a reduction of texture effectiveness is shown for highly slanted geometries.

4.3. Scaling of the optimal dimple shape

In order to analyse the scaling of the optimal dimple shape (in terms of friction reduction or increase of load carrying capacity), two parametric studies of the dimple diameter and depth with respect to the operating parameter are considered. The first one concerns simulations with prescribed gap height, which, due to the lower computational cost, allows studying a richer parametric space. The second one is based on the simulations with constant normal force, so that a closer comparison with experimental data can be obtained.

4.3.1. Analysis of the optimal dimple depth

In this study we vary viscosity and velocity in a range which is broad enough to represent most of the operating conditions in real applications. Table 2 resumes the range of these parameters while the gap height ranges from $0.5 \mu m$ to $20 \mu m$. For each point of this parametric space we consider the influence of different dimple diameters and depths on the load carrying capacity and on the friction coefficient. These two quantities are computed, according to [36] and [37], through the following definitions:

$$F_N = \int_A (P - P_{amb}) dA \quad (4)$$

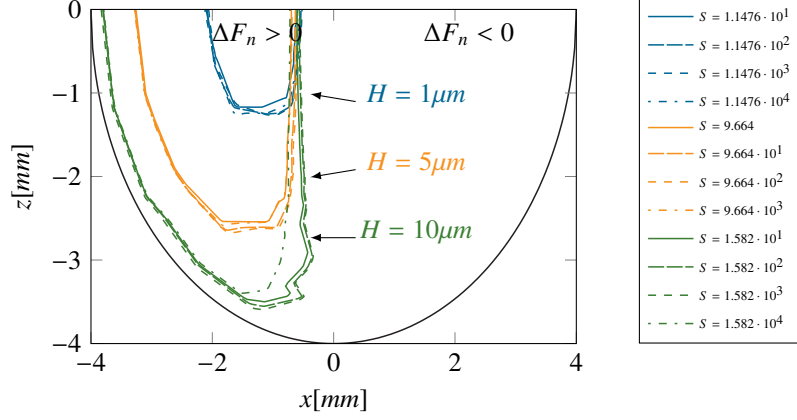
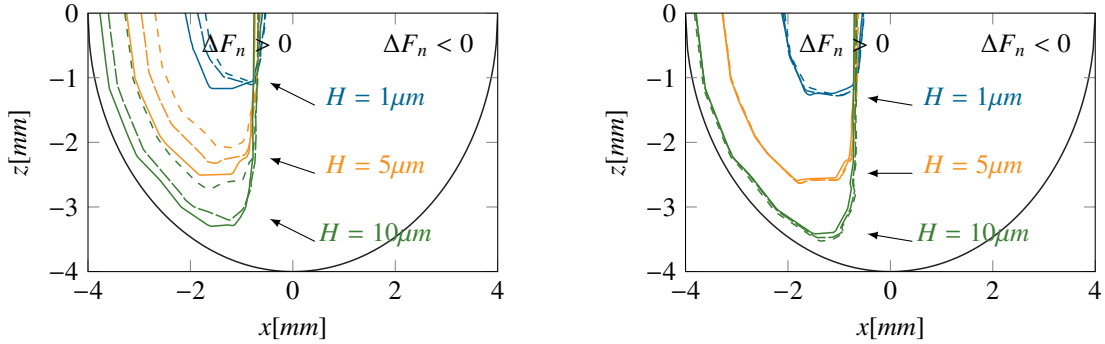


Figure 12: Zero-isolines of the net normal force $\Delta F_n = F_{n_{ext}} - F_{n_{unl,xl}}$ obtained through a variation of velocity, viscosity and gap height. Only one half of the pin (geometry A) is shown due to symmetry. The flow is from left to right. The Sommerfeld number is computed according to the definition in equation 3. The normal load W is a result of the simulations which are carried out at prescribed gap height H . The values of viscosity and velocity are varied in the following range: $0.0187 < \mu < 1.871 \text{ Pas}$ and $0.01 < U < 1 \text{ m/s}$.



(a) Constant dimple aspect ratio $\lambda = 0.1$. Different dimple diameters — $D = 100 \mu\text{m}$, - - - $D = 200 \mu\text{m}$, . . . $D = 400 \mu\text{m}$.

(b) Constant dimple depth $\text{Depth} = 10 \mu\text{m}$. Different dimple diameters — $D = 100 \mu\text{m}$, - - - $D = 200 \mu\text{m}$, . . . $D = 400 \mu\text{m}$.

Figure 13: Zero-isolines of the net normal force $\Delta F_n = F_{n_{ext}} - F_{n_{unl,xl}}$ plotted on half of the pin (geometry A). Variation of dimple shape and gap height H . Viscosity and velocity are kept constant in this plots: $\mu = 0.187 \text{ Pas}$ and $U = 0.1 \text{ m/s}$

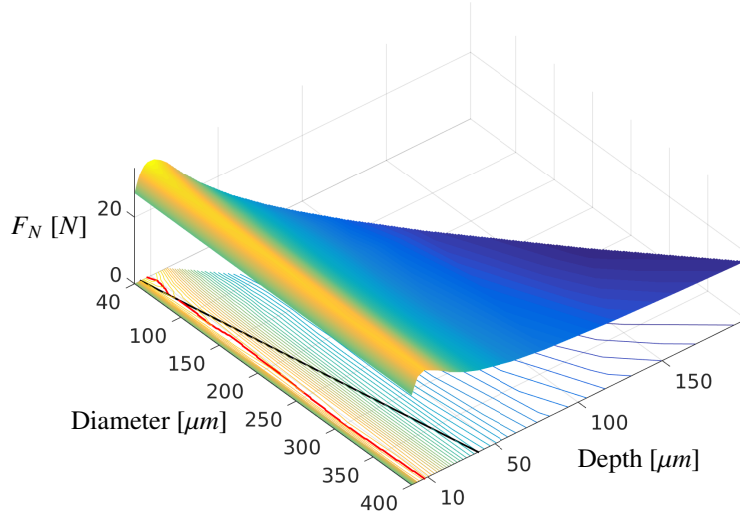


Figure 14: Distribution and contour lines of the normal force in the depth-diameter parameter space. The red line indicates the position of the optimal depth. The black one represents the aspect ratio $\lambda = 0.1$ that was employed in the experimental parameter study [14]. The portrayed case is obtained with the following operating parameters: $U = 1\text{m/s}$, $\mu = 0.187\text{Pas}$, $H = 7\mu\text{m}$

$$F_T = \int_A \left[(1 - \theta) \mu \frac{U}{h} + (1 - \theta) \frac{h}{2} \frac{\partial P}{\partial x} \right] dA \quad (5)$$

$$C_f = \frac{F_T}{F_N}. \quad (6)$$

As explained in section 3, only the first half of the pin surface is textured. The texture density is kept constant at $\rho_{\text{txt}} = 10\%$ as in the corresponding experiments [14, 26]. The dimple diameter varies between $40\mu\text{m}$ and $400\mu\text{m}$ while the depth varies between 0 (untextured case) and $200\mu\text{m}$. Since the dimples have a spherical shape, the maximum aspect ratio $\lambda = \frac{\text{depth}}{\text{diameter}}$ cannot exceed the value 0.5. As shown by Dobrica and Fillon [19], for values of $\lambda > 0.1$ the applicability of the Reynolds equation may become arguable in case of square 1D dimples, nonetheless, thanks to the less sharp shape of the 2D spherical dimples non-linear effects are less likely to occur. The validity of the Reynolds equation is also assured by the very low Reynolds number, which is given by $Re < 0.01$ for every combination of the operating parameters.

Figure 14 shows the distribution of the normal force for different values of dimple diameter and depth. It can clearly be seen that the normal force is strongly governed by the dimple depth and depends on the dimple diameter in a much weaker way. This means that an optimal depth value exists, which is insensitive to the diameter. The experimental investigation [14] addresses the role of the dimple diameter at constant aspect ratio $\lambda = 0.1$, hence, the optimal diameter is located at the intersection between the optimal depth line (red) and the line that links diameter and depth for that specific aspect ratio (black).

It is interesting to notice that the topology of the normal force distribution in the depth-diameter space does not change for different values of viscosity and velocity although the absolute force value is directly proportional to them according to the Sommerfeld number (see Eq. 3). The optimal depth can shift only if the prescribed gap height changes. This indicates that geometrical parameters such as the gap height play a much bigger role than viscosity and velocity in the definition of the optimal depth, which is in agreement with the findings discussed in section 4.2.3.

This fact becomes even more evident in figure 15, where the value of the optimal depth, obtained with the process just described, is plotted against the gap height for several different values of viscosity and sliding velocity. Each single point in this figure corresponds to a parametric investigation of the normal force in the diameter-depth space for a specific combination of velocity and viscosity, as shown in table 2. It can be seen that the dependency of the optimal depth on the gap height is almost perfectly linear, regardless of the combination of viscosity and velocity. This linear dependency is also in agreement with the findings of 1D parametric studies such as those by Fowell *et al.* [31] and Ramesh *et al.* [30].

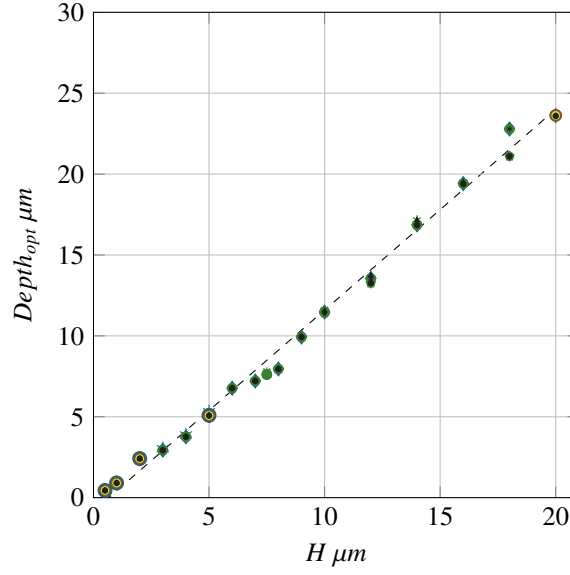


Figure 15: Scaling of the optimal dimple depth with respect to the gap height. Each point represents the optimal depth extrapolated from the process explained with respect to figure 14 for a certain combination of viscosity and velocity as shown in table 2.

$\mu[Pa\cdot s]$ \ $U[m/s]$	0.01	0.05	0.1	0.5	1	5	10
0.0187			■		●		◆
0.0935					●		
0.1871	■	*	■	×	●	◆	◆
0.9357					●		
1.8714	■		■		●		◆

Table 2: Velocity and viscosity range of the parametric study with prescribed gap height used in the figure 15. The texture density is kept constant at $\rho_{txt} = 10\%$, while the gap height ranges between 0.5 and $20\mu m$. The overall range of Sommerfeld number based on the gap height H is $172 < S < 14.3 \cdot 10^3$.

4.3.2. Scaling between dimple depth and the Sommerfeld number

Even though the above found linear relation between optimal dimple depth and gap height is not affected by viscosity changes, a dependency of the optimal depth on viscosity can be observed when the tests are carried out at imposed normal load. This is in agreement with the experimental results, where the pin-on-disc tribometer are run under the input of constant normal force. In these experiments, the performance of different dimple diameters with constant aspect ratio $\lambda = 0.1$ is tested at two different temperatures (see figure 5) and a viscosity dependence is identified.

As explained in section 4.3, an iterative procedure is required in order to represent the experimental procedure numerically. In these simulations, just like in the experiment, the normal force is assigned as input while the gap height is obtained as a result. It is known that for geometries such as slider bearings and the current one, the resulting load is directly proportional to viscosity, velocity and reference length and at the same time inversely proportional to the square of the gap height according to the relation $W = W_r \frac{\mu LU}{H^2}$ [54, 55, 56]. Where the W_r is the non dimensional load which can be arbitrarily chosen to a constant. Consequently, the variations of gap height at imposed normal load are inversely proportional to the applied load as follows:

$$H = \sqrt{\frac{\mu LU W_r}{W}} \quad (7)$$

Therefore, any change in the viscosity μ influences the gap height H , which is now free to vary in order to find the equilibrium position at which the pressure distribution balances the constant load.

With these simulation settings, we focus again on the diameter-depth space as done in section 4.3.1 (figure 15). Since the normal force is prescribed, the search for the optimal depth is carried out with respect to the lowest friction coefficient. As in the previous findings shown in figure 14, the minimal friction coefficient can be found for a constant value of dimple depth and it is not affected by the diameter. The values of optimal depth are plotted against viscosity variation in figure 16. In this scenario, the optimal depth (which corresponds to the minimal friction coefficient) can change through the variation of gap height induced by different values of viscosity. The plot also includes the experimental optimal depth values previously shown in figure 5. As one can see, the slope of the numerical curves is the same as the one observed in experiments, indicating that the computed scaling of the optimal depth is in agreement with the experimental results. Nonetheless, a shift is evident between the numerical curves and the experimental ones. This is most likely due to the fact that the experimental results are obtained in the mixed lubrication regime while the numerical ones, being based on the pure Reynolds equation, can only describe the hydrodynamic regime. In the simulation results, a higher normal force yields smaller optimal depths. This is in agreement with equation 7, because under higher normal load the system finds its equilibrium at a smaller gap height, hence the optimal dimple depth will be smaller.

As a further confirmation of the linear dependency of the optimal depth on the gap height, figure 17 represents the optimal depth values computed at constant normal force with respect of the gap height found as output for every single simulation point in figure 16. It becomes obvious that this linear relation is independent from the approach used to carry out the simulations, namely either with prescribed gap height or prescribed normal force.

As final outcome the linear dependency can be explicitly formulated by recalling, at first, a more general description of the relation between the optimal depth and other parameters, $Depth_{opt} = Depth_{opt}(\mu, U, H, D, W)$, where μ is the viscosity, U the sliding velocity, H the gap height, D the dimple diameter and W the applied load. Thanks to the above mentioned scaling considerations and to the fact that the optimal depth does not depend on the dimple diameter (see figure 14), one can simplify this generic formulation into the following one:

$$Depth_{opt} = kH(\mu, U, W) \quad (8)$$

where k is a constant which depends on the shape of the macro-geometry. By substituting equation 7 into equation 8 we obtain

$$Depth_{opt} = k \sqrt{\frac{\mu LU W_r}{W}} = K \sqrt{\frac{\mu LU}{W}} \quad (9)$$

where the reference dimensionless load W_{ref} can be grouped outside of the square root together with k into the constant K . Therefore, the optimal depth is proportional to the square root of viscosity, velocity and reference length, and inversely proportional to the square root of the normal load.

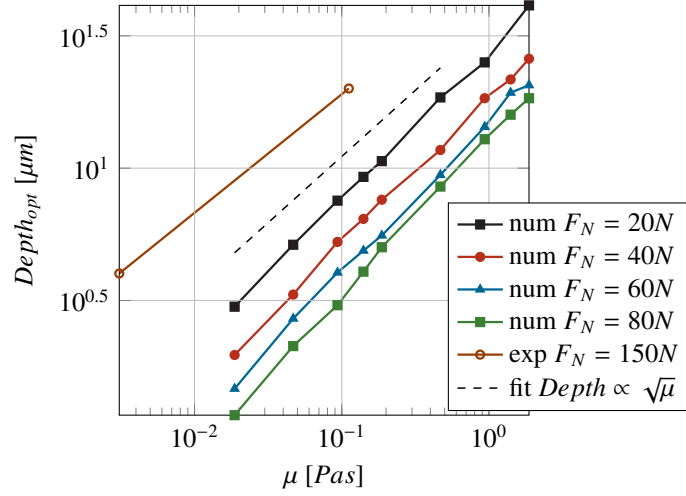


Figure 16: Comparison of the scaling of the optimal depth with respect to viscosity for a set of simulations with constant normal force and experimental results [14]. The fitting dashed line represents the scaling expressed in equation 9.

The relation expressed in equation 9 can be brought into a non-dimensional formulation through the introduction of the Sommerfeld number. For geometries such as slider bearings and the current one, Hamrock *et al.* [54] and Raimondi and Boyd [55], propose a definition of the Sommerfeld number based on the shoulder height s_h (*i.e.* the inlet-outlet difference between of the slider height, see figure 2). This is because, unlike in journal bearings, the clearance cannot be defined *a priori*. For the geometry of the pin A, this shoulder height s_h is measured as the difference between the height at the center of the pin and the one at the edge, so $s_h = 6.22\mu\text{m}$. Thanks to this definition we can reformulate relation 9 as follows:

$$Depth_{opt} = k s_h \sqrt{S} = K \sqrt{\frac{\mu U L s_h^2}{W s_h^2}} \quad (10)$$

which can finally express the relation between the non-dimensional optimal depth and the Sommerfeld number.

$$\frac{Depth_{opt}}{s_h} = K \sqrt{S} \quad (11)$$

This relation suggests that the non dimensional optimal depth is proportional to the square root of the Sommerfeld number times a constant K which depends on the shape of the macro-geometry and the arbitrary reference dimensionless load W_{ref} . This relation is particularly suitable for the comparison between experimental and numerical results since the gap height which is unknown in case of the experiments is not a parameter anymore. Figure 18 shows the numerical results from figure 16 in terms of non-dimensional depth versus the Sommerfeld number S . It can clearly be seen that all data collapse onto a single curve indicating that this scaling holds indeed. The experimental results reveal the same slope but different absolute values in this plot, which is probably due to the fact that they correspond to the mixed lubrication regime. Nonetheless, the good agreement in terms of slope might indicate that the scaling, which was observed experimentally in the mixed lubrication region, is likely due to hydrodynamic mechanisms.

4.4. Effects of the dimple distribution

The third parametric investigation concerns the pattern of the dimpled texture. As shown in section 4.1, dimpled surface textures can have a very different impact on the friction reduction depending on the orientation of the texture.

The corresponding numerical parametric study is set by varying the parameter s_z for several cases with different viscosity, velocity and gap height. In addition, the design parameters such as diameter and depth are varied, while the texture density is kept constant at $\rho_{txt} = 10\%$ like in the experiments. The typical results of such a study is shown in figure 19, where the friction coefficient is plotted against the normalized shift parameter s_z/l_z . As one can see,

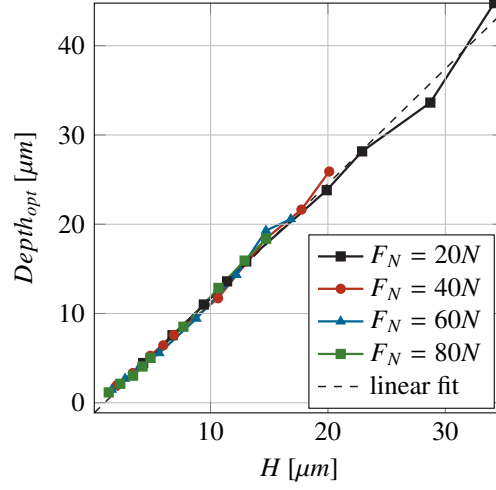


Figure 17: Optimal depth with respect to the gap height computed at which the optimum is found. The values of optimal depth are the same as those shown in figure 16.

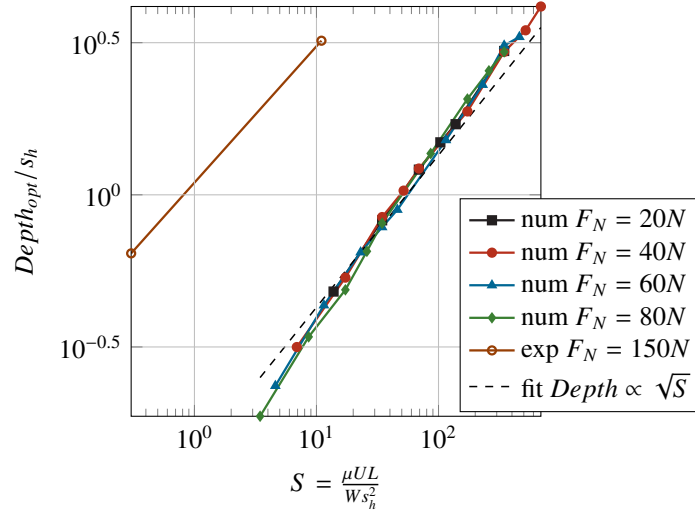


Figure 18: Comparison of the scaling of the non-dimensional optimal depth with respect to Sommerfeld number for a set of simulations with constant normal force and experimental results by Braun *et al.* [14]. The fitting dashed line represents the scaling expressed in equation 11.

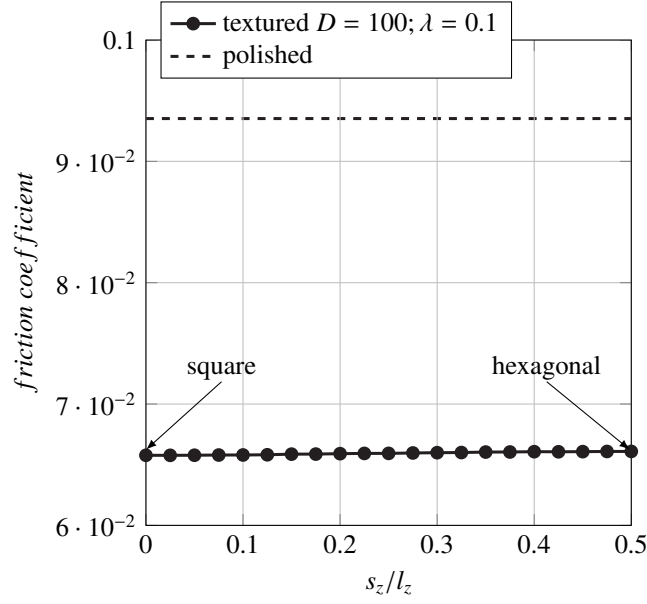


Figure 19: Friction coefficient with respect to the shift parameter of the dimple pattern. The shift parameter is defined in figure 6.

besides a vertical shift due to the fact that the textured surface has a lower friction coefficient, no clear difference can be noticed for different values of s_z . Such behaviour is encountered for all operating and design parameters of the present study and the very small difference, that one can identify in figure 19, remains always below the numerical uncertainty.

This result implies that the Reynolds equation, as used in the present study, does not take into account the relevant physical phenomena which lead to the experimental observation that the hexagonal disposition of dimples shows better results than any other kind of dimple arrangement. The reason behind this discrepancy is likely to lie either in the absence of contact mechanics modelling (as already indicated in section 4.3.2) or in the neglect of the non-linear term [18, 24].

Consequently, the use of the simple Reynolds equation, although with mass conserving cavitation algorithm, cannot explain the effects of different dimple arrangements.

5. Conclusion

A joint numerical and experimental analysis of textured surfaces in tribological contacts is carried out in this work focusing on the relations and scalings between operating parameters and the optimal shape of dimples. The numerical investigation is based on the incompressible Reynolds equation with mass conserving cavitation, whereas particular attention is paid on the computational performance of the solver in order to carry out parametric studies based on high-resolution surface representations of experimental set-ups. The experimental campaigns treated in this work, concerns both, known literature results [14, 26] as well as new tribological tests about partial texturing and texture orientation. The main findings of this work can be summarized as follows:

- The numerical analysis of the sensitivity of the load carrying capacity with respect to the dimple placing points out that the gap height plays a relevant role in defining which part of the surface benefits from the presence of a texture. This numerical evidence leads to the conclusion that a partial texturing on the front half of the pin can improve the tribological performance much better than textures either on the whole or on the second half of the pin. This is in agreement with the experimental investigation carried out in this work and also with literature results.

- Thanks to an extensive parametric study, the effects of operating parameters on dimple diameter and depth are investigated. As results we find that the load carrying capacity has a maximum in the diameter-depth space which is independent of the diameter value, therefore an optimal depth exists which is shown to be a linear function of the gap height. In addition, it is shown that other operative parameters such as viscosity and sliding velocity play a minor role compared to the gap height itself.
- The linear relation between optimal dimple depth and gap height can be expressed through the following relation:

$$Depth_{opt} = kH = K \sqrt{\frac{\mu LU}{W}} \quad (12)$$

or consequently in its non-dimensional formulation:

$$\frac{Depth_{opt}}{s_h} = K \sqrt{S}. \quad (13)$$

This scaling is in agreement with the experimental results [14]. The quantitative shift between experimental and numerical results can be linked to the fact that the numerical analysis describing the hydrodynamic lubrication regime while experiments are carried out in the mixed lubrication regime. Nonetheless, the good agreement in terms of slope between the experimental and the numerical results in figure 16 and 18 suggests that hydrodynamic mechanics induced by dimples play a role also in the mixed lubrication region.

- The experimental investigations concerning the role of texture orientation show that the pseudo-hexagonal arrangement of dimples performs better than a square arrangement or intermediate dispositions. In contrast, the incompressible Reynolds equation alone leads to results which are insensitive to the pattern orientation for all the different combination of operational and design parameters which are tested, indicating that the relevant physical mechanisms is not captured by this model formulation.

6. References

- [1] B. Bhushan. Biomimetics: lessons from nature—an overview. *Philos Transactions Roya Society London A*, 367:1445–86, 2009.
- [2] D. Gropper, L. Wang, and T. J. Harvey. Hydrodynamic lubrication of textured surfaces: A review of modeling techniques and key findings. *Tribology International*, 94:509–529, 2016.
- [3] D.B. Hamilton, J.A. Walowit, and C.M. Allen. A theory of lubrication by microirregularities. *Journal of Basic Engineering*, 88(1):177–185, 1966.
- [4] J. N. Anno, J.A. Walowit, and C.M. Allen. Microasperity lubrication. *Journal of Lubrication Technology*, 90(2):351–355, 1968.
- [5] I. Etsion, G. Halperin, V. Brizmer, and Y. Kligerman. Experimental investigation of laser surface textured parallel thrust bearings. *Tribology Letters*, 17(2):295–300, 2004.
- [6] V. Brizmer, Y. Kligerman, and I. Etsion. A laser surface textured parallel thrust bearing. *Tribology Transactions*, 46(3):397–403, 2003.
- [7] I. Etsion. Improving tribological performance of mechanical components by laser surface texturing. *Tribology Letters*, 17(4):733–737, 2004.
- [8] Y. Kligerman, I. Etsion, and A. Shinkarenko. Improving tribological performance of piston rings by partial surface texturing. *Journal of Tribology*, 127(3):632–638, 2005.
- [9] I. Etsion. State of the art in laser surface texturing. *Journal of tribology*, 127(1):248–253, 2005.
- [10] C. Greiner, M. Schäfer, U. Popp, and P. Gumbsch. Contact splitting and the effect of dimple depth on static friction of textured surfaces. *ACS applied materials & interfaces*, 2014.
- [11] A. Kovalchenko, O. Ajayi, A. Erdemir, and G. Fenske. Friction and wear behavior of laser textured surface under lubricated initial point contact. *Wear*, 271(9):1719–1725, 2011.
- [12] M. Varenberg, G. Halperin, and I. Etsion. Different aspects of the role of wear debris in fretting wear. *Wear*, 252(11):902–910, 2002.
- [13] K. Li, Z. Yao, Y. Hu, and W. Gu. Friction and wear performance of laser peen textured surface under starved lubrication. *Tribology international*, 77:97–105, 2014.
- [14] D. Braun, C. Greiner, J. Schneider, and P. Gumbsch. Efficiency of laser surface texturing in the reduction of friction under mixed lubrication. *Tribology International*, 77:142–147, 2014.
- [15] R. Ausas, P. Ragot, J. Leiva, M. Jai, G. Bayada, and G. C. Buscaglia. The impact of the cavitation model in the analysis of microtextured lubricated journal bearings. *Journal of Tribology*, 129(4):868–875, 2007.
- [16] M. Fowell, A.V. Oliver, A.D. Gosman, H.A. Spikes, and I. Pegg. Entrapment and inlet suction: two mechanisms of hydrodynamic lubrication in textured bearings. *Journal of Tribology*, 129(2):336–347, 2007.
- [17] Y. Kligerman and I. Etsion. Analysis of the hydrodynamic effects in a surface textured circumferential gas seal. *Tribology Transactions*, 44(3):472–478, 2001.
- [18] T. Woloszynski, P. Podsiadlo, and G. W. Stachowiak. Evaluation of discretization and integration methods for the analysis of finite hydrodynamic bearings with surface texturing. *Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology*, 229(4):465–477, 2015.

- [19] M.B. Dobrica and M. Fillon. About the validity of reynolds equation and inertia effects in textured sliders of infinite width. *Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology*, 223(1):69–78, 2009.
- [20] S. Cupillard, S. Glavatskih, and M.J. Cervantes. Inertia effects in textured hydrodynamic contacts. *Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology*, 224(8):751–756, 2010.
- [21] S. Cupillard, M. J. Cervantes, and S. Glavatskih. Pressure buildup mechanism in a textured inlet of a hydrodynamic contact. *Journal of Tribology*, 130(2):021701, 2008.
- [22] M. Adjemout, N. Brunetiere, and J. Bouyer. Numerical analysis of the texture effect on the hydrodynamic performance of a mechanical seal. *Surface Topography: Metrology and Properties*, 4(1):014002, 2015.
- [23] C.I. Papadopoulos, L. Kaiktsis, and M. Fillon. Computational fluid dynamics thermohydrodynamic analysis of three-dimensional sector-pad thrust bearings with rectangular dimples. *Journal of Tribology*, 136(1):011702, 2014.
- [24] M.B. Dobrica, M. Fillon, M.D. Pascovici, and T. Cicone. Optimizing surface texture for hydrodynamic lubricated contacts using a mass-conserving numerical approach. *Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology*, 224(8):737–750, 2010.
- [25] X. Wang, K. Kato, K. Adachi, and K. Aizawa. Loads carrying capacity map for the surface texture design of sic thrust bearing sliding in water. *Tribology International*, 36(3):189–197, 2003.
- [26] D. Braun J. Schneider and C. Greiner. Laser textured surfaces for mixed lubrication: Influence of aspect ratio, textured area and dimple arrangement. *Lubricants*, 2017.
- [27] X. Wang, J. Wang, B. Zhang, and W. Huang. Design principles for the area density of dimple patterns. *Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology*, 229(4):538–546, 2015.
- [28] I. Etsion, Y. Kligerman, and G. Halperin. Analytical and experimental investigation of laser-textured mechanical seal faces. *Tribology Transactions*, 42(3):511–516, 1999.
- [29] T. H. Yu and F. Sadeghi. Groove effects on thrust washer lubrication. *Journal of Tribology*, 123(2):295–304, 2001.
- [30] A. Ramesh, W. Akram, S. P. Mishra, A. H. Cannon, A. A. Polycarpou, and W. P. King. Friction characteristics of microtextured surfaces under mixed and hydrodynamic lubrication. *Tribology International*, 57:170–176, 2013.
- [31] M.T. Fowell, S. Medina, A.V. Olver, H.A. Spikes, and I.G. Pegg. Parametric study of texturing in convergent bearings. *Tribology International*, 52:7–16, 2012.
- [32] H. Yu, W. Huang, and X. Wang. Dimple patterns design for different circumstances. *Lubrication Science*, 25(2):67–78, 2013.
- [33] R. Rahmani, I. Mirzaee, A. Shirvani, and H. Shirvani. An analytical approach for analysis and optimisation of slider bearings with infinite width parallel textures. *Tribology International*, 43(8):1551–1565, 2010.
- [34] R. Rahmani, A. Shirvani, and H. Shirvani. Optimization of partially textured parallel thrust bearings with square-shaped micro-dimples. *Tribology Transactions*, 50(3):401–406, 2007.
- [35] M. B. Dobrica, M. Fillon, M. D Pascovici, and T. Cicone. Texturing effects in plane-inclined slider bearings. In *Proceedings of the ASME/STLE International Joint Tribology Conference*, pages 22–24, 2007.
- [36] T. Woloszynski, P. Podsiadlo, and G. W. Stachowiak. Efficient solution to the cavitation problem in hydrodynamic lubrication. *Tribology Letters*, 58(1):18, 2015.
- [37] C. Gu, X. Meng, Y. Xie, and D. Zhang. Mixed lubrication problems in the presence of textures: An efficient solution to the cavitation problem with consideration of roughness effects. *Tribology International*, 103:516–528, 2016.
- [38] H. G. Elrod. A cavitation algorithm. *ASME J. Lubr. Technol.*, 103:350, 1981.
- [39] M. Arghir, A. Alsayed, and D. Nicolas. The finite volume solution of the reynolds equation of lubrication with film discontinuities. *International journal of mechanical sciences*, 44(10):2119–2132, 2002.
- [40] C. Greiner, T. Merz, D. Braun, A. Codrignani, and F. Magagnato. Optimum dimple diameter for friction reduction with laser surface texturing: the effect of velocity gradient. *Surface Topography: Metrology and Properties*, 3(4):044001, 2015.
- [41] A. N. Murthy, I. Etsion, and F. E. Talke. Analysis of surface textured air bearing sliders with rarefaction effects. *Tribology Letters*, 28(3):251–261, 2007.
- [42] V.G. Marian, M. Kilian, and W. Scholz. Theoretical and experimental analysis of a partially textured thrust bearing with square dimples. *Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology*, 221(7):771–778, 2007.
- [43] Y. Qiu and M.M. Khonsari. Performance analysis of full-film textured surfaces with consideration of roughness effects. *Journal of tribology*, 133(2):021704, 2011.
- [44] N. Brunetière and B. Tournerie. Numerical analysis of a surface-textured mechanical seal operating in mixed lubrication regime. *Tribology International*, 49:80–89, 2012.
- [45] S. Kango, R.K. Sharma, and R.K. Pandey. Comparative analysis of textured and grooved hydrodynamic journal bearing. *Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology*, 228(1):82–95, 2014.
- [46] H. Zhang, G. Dong, M. Hua, F. Guo, and K. S. Chin. Parametric design of surface textures on journal bearing. *Industrial Lubrication and Tribology*, 67(4):359–369, 2015.
- [47] S. Vlădescu, A. Ciniero, K. Tufail, A. Gangopadhyay, and T. Reddyhoff. Optimisation of pocket geometry for friction reduction in piston-liner contacts. *Tribology Transactions*, (just-accepted):00–00, 2017.
- [48] N. Tala-Ighil, P. Maspeyrot, M. Fillon, and A. Bounif. Effects of surface texture on journal-bearing characteristics under steady-state operating conditions. *Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology*, 221(6):623–633, 2007.
- [49] M.D. Pascovici, T. Cicone, M. Fillon, and M.B. Dobrica. Analytical investigation of a partially textured parallel slider. *Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology*, 223(2):151–158, 2009.
- [50] F. Sahlin, S. B. Glavatskih, T. Almqvist, and R. Larsson. Two-dimensional cfd-analysis of micro-patterned surfaces in hydrodynamic lubrication. *Transactions of the ASME-F-Journal of Tribology*, 127(1):96–102, 2005.
- [51] M. Arghir, N. Roucou, M. Helene, and J. Frene. Theoretical analysis of the incompressible laminar flow in a macro-roughness cell. *Journal of tribology*, 125(2):309–318, 2003.
- [52] H. Yong and R. Balendra. Cfd analysis on the lubrication behaviours of journal bearing with dimples. pages 1279–1284, 2009.

- [53] A.R. Gherca, P. Maspeyrot, M. Hajjam, and A. Fatu. Influence of texture geometry on the hydrodynamic performances of parallel bearings. *Tribology Transactions*, 56(3):321–332, 2013.
- [54] B. J. Hamrock, S. R. Schmid, and B. O. Jacobson. *Fundamentals of fluid film lubrication*. CRC press, 2004.
- [55] A.A. Raimondi and J. Boyd. Applying bearing theory to the analysis and design of pad-type bearings. *Trans. ASME*, 77(3):287–309, 1955.
- [56] A. Codrignani, D. Savio, F. Magagnato, and B. Frohnäpfel. A scaling parameter for pressure losses and thermal effects in lubricant flows with viscous dissipation. *Tribology International*, 113:238–244, 2017.