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Lubricated Textured Plan Parallel Sliding Contacts
The Effect of Boundary Slip and Cavitation

Muchammad

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The Effect of Boundary
Slip and Cavitation

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**LUBRICATED TEXTURED PLAN PARALLEL
SLIDING CONTACTS**
THE EFFECT OF BOUNDARY SLIP AND CAVITATION

DISSERTATION

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Summary

Surface texturing is an effective approach to increasing the tribological performance of lubricated mechanical components. It is known that introducing textured surfaces can significantly affect friction and load support of sliding bearings. In recent years, a great deal of effort has been employed to investigate the influence of the texture's geometric parameters. It was shown in various experiments that such texturing enhances load support and reduces hydrodynamic friction; for instance, systems with two parallel sliding surfaces, mechanical seals, and reciprocating (cylinder-liner) contacts. Recently, in addition to surface texturing, the use of an artificial boundary slip surface is also of great interest with respect to lubrication. It is worth noting that when the effect of boundary slip is of particular interest in bearing analysis, for most previously published works cavitation was ignored, and most of them used a non-mass conserving equation. As can be seen in the analytical and numerical results, cavitation has a significant effect and the exclusion of cavitation in mathematical models may lead to wrong conclusions.

As is known, cavitation in a bearing is common and directly influences the hydrodynamic pressure and, subsequently, the load support and friction of the contact. In order to address such a cavitation effect, several cavitation models have been proposed by numerous researchers. In this research, the focus is on modifying the Reynolds equation by considering the cavitation model in which the mass conservation along the lubricated contact is fulfilled. The first order of the Reynolds approach is adopted. The hydrodynamic performance of a textured contact with boundary slip is investigated through analytical and numerical simulations. In order to analyse the boundary slip, the slip is modelled using the Navier slip criterion.

In the present work, an analytical equation based on the first-order Reynolds theory is derived. In this way, the Reynolds equation is extended by considering boundary slip and cavitation. Here, boundary slip is varied with respect to the location on the lubricated contact. For a single and double pocketed slip bearing, a mathematical model including boundary slip and cavitation has been developed, from which expressions of the pressure and flow rate can be derived. Applying pressure continuity at boundary points, a set of governing equations can be obtained for the flow rate and for constants of integration in the pressure expressions. The global contact geometry in this study is that of a rectangular textured sliding surface.

The significance of the cavitation effect in the lubrication analysis based on the developed model is investigated. Analytically, the cavitation and no cavitation condition with boundary slip and no-slip on parallel slider bearings with a single texture is discussed. It is found that adding boundary slip can reduce the cavitation area, affecting the pressure and load support of the textured slider bearing. The pressure distribution for the cavitation and no cavitation condition by varying the inlet length, pocket length and pocket depth is discussed. Besides, the analysis has also been conducted for the case of double pockets.

This thesis is divided into two parts. The first part relates to the modelling aspects and the methodology used to derive a modified form of the Reynolds equation capable of including wall slip boundary conditions. The second part is devoted to the details of individual research papers. This enables the reader to obtain a clear understanding of the overall purpose of the research from the first part while the second part elucidates the details.

Samenvatting

Het aanbrengen van een oppervlaktetextuur is een effectieve manier om de tribologische prestaties van gesmeerde mechanische onderdelen te verbeteren. Het is bekend dat het introduceren van dergelijke oppervlakken een aanzienlijke invloed kan hebben op de wrijving en de belastbaarheid van glijlagers. De laatste jaren is veel inspanning geleverd om de invloed van geometrische parameters van de textuur te onderzoeken. In de literatuur is met verschillende experimenten aangetoond dat door middel van oppervlaktetexturering de belastbaarheid verbetert en de hydrodynamische wrijving vermindert in bijvoorbeeld systemen met twee parallelle glijvlakken zoals bij mechanische afdichtingen en heen en weer gaande contacten. Naast het aanbrengen van een oppervlaktetextuur is slip, tussen het oppervlak en de (smeer)vloeistof, ook van groot belang met betrekking tot de tribologische prestaties van een gesmeerd contact. Verder moet vermeld worden dat wanneer het effect van texturering en slip geanalyseerd wordt, cavitatie niet genegeerd kan worden zoals dat in veel publicaties is gebeurd naast het niet voldoen aan de massa continuïteit vergelijking. Cavitatie heeft, zoals te zien is in de analytische en numerieke resultaten, een significant effect en het negeren van cavitatie kan leiden tot verkeerde conclusies.

Zoals bekend is treedt cavitatie op in gesmeerde contacten en heeft direct invloed op de hydrodynamische druk en daarmee op de belastbaarheid en wrijving van het contact. Om het effect van cavitatie mee te nemen zijn verschillende modellen beschikbaar. In dit onderzoek ligt de focus op het wijzigen van de Reynolds vergelijking rekening houdend met cavitatie en tevens te voldoen aan massa conservering waarbij de eerste orde Reynolds benadering is toegepast. De invloed van slip op de hydrodynamische prestaties van een getextureerd contact is door middel van analytische en numerieke simulaties onderzocht. Om de invloed van slip te analyseren is gebruik gemaakt van het Navier slip criterium.

In het huidige werk is een analytische vergelijking afgeleid gebaseerd op de eerste orde Reynolds theorie rekening houdend met slip en cavitatie voor een lager met een enkele en dubbele groef. De slip is gevarieerd met betrekking tot de locatie op het oppervlak van het gesmeerd contact. Uitdrukkingen zijn verkregen voor de stroomsnelheid en druk in het contact.

Het effect van cavitatie is met behulp van het ontwikkelde model onderzocht. Cavitatie en niet cavitatie voorwaarden alsmede slip en niet slip condities zijn voor parallelle glijdende loopvlakken met een enkele groef besproken. Het is gebleken dat bij slip het cavitatiegebied kan verkleinen, waardoor de druk wordt beïnvloed en diensgevolge de belastbaarheid van het getextureerde glijlager. De drukverdeling is besproken voor de cavitatie en niet cavitatie conditie door de inlaatlengte, groeflengte en groefdiepte te variëren. Daarnaast is de analyse ook uitgevoerd voor het geval van dubbele groeven.

Dit proefschrift is opgedeeld in twee delen. Het eerste deel heeft betrekking op de modelleringsaspecten en de methodologie die gebruikt is om een gewijzigde vorm van de Reynolds vergelijking af te leiden inclusief cavitatie en slip. Het tweede deel is gewijd aan de details van individuele onderzoek artikelen. Hierdoor krijgt de lezer een duidelijk beeld van het algemene doel van het onderzoek uit het eerste deel, terwijl het tweede deel in gaat op de details.

Contents

Summary	vii

Nomenclature	xv

Part I

1. Introduction	1
1.1. Lubricated textured surfaces, cavitation and slip	1
1.2. Effect of slip on the cavitation zone	4
1.3. Summary and outline thesis	5
2. Lubrication with slip	7
2.1. The classical Reynolds equation	7
2.2. Cavitation models	8
2.2.1. Half Sommerfeld (Gumbel) cavitation model	10
2.2.2. The Swift-Stieber (or Reynolds) cavitation model	12
2.2.3. Mass conserving cavitation model.....	12
2.3. Previous studies of hydrodynamic lubrication with cavitation	15
2.4. Slip models	17
2.4.1. Slip length model (SLM)	17
2.4.2. Critical shear stress model (CSSM)	18
2.5. Modified Reynolds equation with cavitation	20
2.5.1. Load support	20
2.5.2. Friction force	24
2.6. Summary	25
3. Modelling	27
3.1. Modified Reynolds equation with slip, texture and cavitation	28
3.1.1. Theoretical modelling	34
3.1.2. Numerical approach	43
3.2. Lubrication performance	45
3.3. Slip texture consideration	46
3.3.1. Slip parameters	46
3.3.2. Surface texture	47
3.4. Summary	49

4. Results	51
4.1. Significance of the cavitation phenomena	51
4.2. Combined effects of slip and cavitation	52
4.2.1. Slip in the pocket area of bearing	55
4.2.2. Slip in the inlet area of bearing	59
4.2.3. Slip in the full area of bearing	61
4.3. Single texturing (theoretical approach)	65
4.4. Multi texturing (theoretical and numerical approach)	68
4.4.1. Theoretical approach	68
4.4.2. Numerical approach	74
4.5. Validation	78
4.6. Summary	81
5. Conclusion and future work	83
5.1. Conclusion	83
5.2. Future work	84
References	85

Part II List of Appended Papers

Paper A:

M. Muchammad, M. Tauviquirrahman, J. Jamari, D.J. Schipper, 2016, "An Analytical Approach on the Tribological Behaviour of Pocketed Slider Bearings with Boundary Slip Including Cavitation", *Lubrication Science*, Volume 29, Issue 3, pp. 133-152.
doi: 10.1002/lis.1361.

Paper B:

M. Muchammad, M. Tauviquirrahman, A.W. Pratomo, J. Jamari, D.J. Schipper, 2017, "Theoretical Investigation of Boundary Slip on the Hydrodynamic Lubrication Performance in Pocketed Bearings Including Cavitation", *International Journal of Surface Science and Engineering*, Volume 11, Issue 2, pp. 100-117.
doi: 10.1504/IJSURFSE.2017.084669.

Paper C:

M. Muchammad, M. Tauviquirrahman, J. Jamari, D.J. Schipper, 2017, "Numerical Investigation of Texturing and Wall Slip in Lubricated Sliding Contact Considering Cavitation", *Tribology Online*, Volume 12, Issue 2, pp. 67-75.
doi: 10.2474/trol.12.67.

Paper D:

M. Muchammad, M. Tauviquirrahman, J. Jamari, D.J. Schipper, 2021, "Analysis of the Effect of the Slip Pocket in Single and Double Parallel Bearing Considering Cavitation: A Theoretical Approach", *Lubricants*, Volume 9, Issue 1, Article Number 3, pp. 1-20.
doi: 10.3390/lubricants9010003

Paper E:

M. Muchammad, M. Tauviquirrahman, F. Hilmy, J. Jamari, D.J. Schipper, "Effect of a Hydrophobic Surface on the Load Carrying Capacity of a Double Pocket Lubricated Plan Parallel Sliding Contact", to be submitted: *Tribology – Materials, Surfaces & Interfaces*.

Paper F:

M. Muchammad, M. Tauviquirrahman, A.W. Pratomo, J. Jamari, D.J. Schipper, 2016, "The Beneficial Effect of Superhydrophobic Layer Addition on Lubrication Behaviour in Bearing Application", *AIP (American Institute of Physics) Conference Proceedings*, Volume 1725, Issue 020047, pp. 1-8, ISBN: 0094243X.
doi: 10.1063/1.4945501.

Paper G:

M. Muchammad, M. Tauviquirrahman, A.W. Pratomo, J. Jamari, D.J. Schipper, 2016, "The Effect of Boundary Slip and Cavitation on Hydrodynamic Pressure Generation in Pocket Bearings", *AIP (American Institute of Physics) Conference Proceedings*, Volume 1725, Issue 020048, pp. 1-7, ISSN: 0094243X.
doi: 10.1063/1.4945502.

Paper H:

M. Muchammad, F. Hilmy, M. Tauviquirrahman, J. Jamari, D.J. Schipper, 2018, "CFD Analysis of Texture Depth and Length on the Performance of Hydrodynamic Lubricated Bearing", IOP (Institute of Physics) Conference Series: Journal of Physics Conference Series, Volume 1090, Issue 012067, pp. 1-6, ISSN: 17578981.
doi :10.1088/1742-6596/1090/1/012067.

Nomenclature

Roman Symbols

a	inlet length	[m]
b	first pocket width	[m]
B_o	slider length	[m]
b'	slip length	[m]
c	length between pockets	[m]
d	second pocket width	[m]
D	universal function	[-]
e	outlet land length	[m]
F	friction force	[N]
F'	switch function	[-]
h	fluid film thickness	[m]
h_o	minimum film thickness in inlet length	[m]
h_d	height of microtexture pocket	[m]
h_{d1}	height of microtexture first pocket	[m]
h_{d2}	height of microtexture second pocket	[m]
L_C	texture cell length	[m]
L_D	groove length	[m]
L_x	length of lubricated surface in the- x direction	[m]
L_y	length of lubricated surface in the- y direction	[m]
p	hydrodynamic pressure	[Pa]
p_{atm}	atmospheric pressure	[Pa]
p_{cav}	cavitation pressure	[Pa]
q	flow rate	[m ³ /s]
T_R	texturing zone	[m]
U	sliding velocity	[m/s]
u_s	slip velocity	[m/s]
W	load support (or load carrying capacity) per unit width	[N/m]
x	coordinate in sliding direction	[m]
X_b	non cavitated fraction of first pocket breadth	[m]
X_d	non cavitated fraction of second pocket breadth	[m]
z	cross film coordinate	[m]

Greek Symbols

α	slip coefficient	[m ² s/kg]
α_h, α_s	slip coefficient at surface s (moving) and h (stationary)	[m ² s/kg]
η	dynamic viscosity	[kg/(m.s)]

μ	coefficient of friction	[-]
τ	shear stress	[Pa]
τ_c	critical shear stress	[Pa]
ρ	density of lubricant	[kg/m ³]
ρ_o	density of lubricant-gas mixture	[kg/m ³]

Sub/Superscripts

i	input
o	output
a	top surface
b	bottom surface

Abbreviations

CFD	Computational Fluid Dynamics
CSSM	Critical Shear Stress Model
HS	Half Sommerfeld
JFO	Jakobsson Floberg Olsson
MEMS	Micro Electro Mechanical System
SLM	Slip Length Model
SS	Swift-Stieber
TDMA	Tri Diagonal-Matrix-Algorithm
1D	One Dimensional
2D	Two Dimensional

Dimensionless Parameters

A	$\alpha\eta / h_o$
a^*	a/B_o
b^*	b/B_o
C_{Lb}^*	$(b-X_b)/B_o$
HD	h_d/h_o
$HD1$	h_{d1}/h_o
p^*	$ph_o^2 / \eta B_o U$
W^*	$wh_o^2 / U\eta B_o^2$
x^*	x/B_o
X_b^*	X_b/B_o

Part I

Chapter 1

Introduction

1.1. Lubricated textured surfaces, cavitation and slip

Surface texturing is an effective approach to increasing the tribological performance of lubricated mechanical components. It is known that introducing textured surfaces can affect friction and load support of sliding bearings significantly. In recent years, a great deal of effort has been deployed to investigate the influence of the texture's geometric parameters. It was shown in various experiments that such texturing enhances load support and reduces hydrodynamic friction; for instance, systems with two parallel sliding surfaces [1-2], mechanical seals [3], and reciprocating (cylinder-liner) contacts [4].

The effect of surface texturing on the tribological performance have been numerically analysed by many researchers such as Arghir et al. [5], Sahlin et al. [6], Brajdic-Mitidieri et al. [7], Kligerman et al. [8], Fowell et al. [9], Dobrica and Fillon [10] and Han et al. [11]. There are two approaches to investigating textured surfaces: using computational fluid dynamics (CFD) models based on Navier-Stokes [5-7,10-11] and using Reynolds [6,8-10]. Sahlin et al. [6] confirmed that by using CFD, the introduction of a micro-groove on the stationary surface of a parallel slider leads to a net pressure build-up and hydrodynamic load support. In their research, it was found that the effect of inertia on lubrication was dominant. Similarly, Han et al. [11] reported a three-dimensional CFD analysis of a spherical dimple on the stationary surface of a parallel sliding contact. An optimum micro dimple (depth and width) was defined that gives the greatest load support and lowest coefficient of friction. Brajdic-Mitidieri et al. [7] used the Navier-Stokes equation combined with a cavitation model to analyse the lubricant behaviour in plain pad bearings having a closed pocket. Such texture could produce a reduction in the friction coefficient and an increase in load support. This effect depends on the bearing convergence ratio (inlet over outlet film thickness). It should be pointed out that these analyses [5-7, 11] focused solely on single textured cells with centred dimples. Such configurations can lead to wrong conclusions in the prediction of hydrodynamic performance concerning the inertia effect [10].

In the case of multiple cells in hydrodynamic parallel sliding textured surfaces, Dobrica and Fillon [10] conducted a parametric study on the texturing parameters, dimension (depth and width), number, and density of the cells to improve the hydrodynamic performance. They analysed the validity of the Reynolds equation and inertia effects in parallel textured sliders of infinite width using the Reynolds equation and commercial CFD simulation software (FLUENT). One of the original conclusions of this study was that inertia was noted to have a negative effect when applied to partially textured surfaces, i.e. reduced load support which was also found for the fully textured configurations. This result contradicts the results from the works of Refs. [5-7,11] in the case of two parallel moving surfaces with a single cell texture, and the work of Kligerman et al. [8] in the case of the contact between a piston ring and cylinder liner with multiple texture cells. Using a theoretical model based on a modified Reynolds equation [8], it was demonstrated that partial texturing has an advantage over full texturing in terms of friction. A load support mechanism between parallel sliding surfaces was proposed by Fowell et al. [9]. They showed that the inlet suction mechanism in textured hydrodynamic bearings that have micro-pockets on their surface provides enhanced load support and reduced friction.

In most of the theoretical studies cited above, it has been assumed that the lubricated contact was represented by no-slip planes. Obviously, this assumption simplified the theoretical analysis and was, to some extent, applied successfully to engineering problems. However, with the continuous progress of the nano measurement techniques during recent years, a nano scale measurement is possible with respect to wall slip [12-14].

Recently, in addition to surface texturing, the use of an artificial slip surface is also of great interest with respect to lubrication. Such a surface was introduced and applied to bearings with the help of micro-fabrication techniques. On a micro scale such as in MEMS (micro-electro-mechanical system), the boundary condition, i.e. slip or no-slip situation, will play a very important role in determining the fluid flow behaviour. Control of the boundary condition will allow a degree of control over the hydrodynamic pressure in confined systems and will be important in a lubricated sliding contact. How to control the wall slip was one of the challenging research questions in recent investigations. However, in a tribological lubricated system such as in a bearing, the presence of the slip can be engineered in certain areas by modifying the contacting surfaces. Based on a literature survey, there are at least three mechanisms which lead to the presence of the slip. Firstly, the slip exists due to the effect of wetting and surface energy [15-18]. Secondly, the slip occurs due to the surface topography, in other words a low roughness value, as suggested by some researchers [17,19]. Thirdly, one can add components (additives) to the base oil to generate wall slip as investigated by Spikes and his group [20-21]. As a remark, it is clear that at the macro scale, the roughness and wetting properties have a significant role in promoting slip at the wall in lubricated systems. Also, in greased lubricated systems in which the film formation is governed by the viscosity of the base oil, the slip phenomenon can be controlled by modifying the roughness [22-23]. Several researchers such as Salant and Fortier [24], Wu et al. [25], and Ma et al. [26-27] have explored the behaviour of the sliding contact using an optimized slip zone with respect to load support. The results of all these investigations show the existence of a lift force (load support); there is even no wedge effect (two parallel sliding surfaces) using such an artificial slip zone. However, none of these studies have included surface texturing.

Very few researchers appear to have considered the interplay of surface texture and boundary slip effect on lubrication. Bayada and Meurisse [28] compared the slip/no-

slip heterogeneity with roughness. They concluded that film rupture in diverging zones can be provoked by a certain artificial slip boundary configuration which is similar to that caused by geometrical roughness. However, in their study, the combination of the effect of artificial slip and surface texture on the hydrodynamic behaviour was not conducted. Rao [29] employed artificial slip on a stationary surface having a single groove at the slider and journal bearing and calculated pressure and shear stress distributions. The modified Reynolds equation with the Navier-slip boundary condition was solved numerically with regard to the simultaneous effect of wall slip and single groove texture. In a recent publication, Aurelian et al. [30] investigated the influence of boundary slip on load support and power loss in hydrodynamic fluid bearings. A simple textured/boundary slip combination was investigated. The main conclusion of their study was that the texture/slip zone geometry should be chosen carefully because an inappropriate choice can lead to a drastic deterioration of the bearing performance, especially in relation to the load support. However, the authors focused only on the slip property with zero critical shear stress. Tauviquirrahman et al. [31-34] explored the possibility of slip and texture in order to improve the performance characteristics. Their results indicate that the combined texture/slip pattern increases the load-carrying capacity and decreases friction.

In most of the previous works, the load support generation mechanisms were demonstrated by using a simple simulation approach for a one-dimensional parallel bearing having a single cell texture without cavitation. Regarding the cavitation effect in the slip textured contact on the lubrication performance, it is noted that in recent literature only Wang and Lu [35] considered the cavitation effect for a two-dimensional sleeve bearing. In the present study, an improvement in the performance of pocketed slider bearings is examined based on the wall slip phenomenon. The flow pattern in the bearing can be altered if the wall slip is located in a deterministic way in certain regions of the pocketed bearing surface. A modified form of the Reynolds equation containing the cavitation effect is derived in which the wall slip is allowed to occur in a textured bearing.

The issue in analytical and numerical modelling of a liquid lubricated contact is the treatment of the cavitation phenomena. Dowson and Taylor [36] and Booker [37] have presented reviews of cavitation models. The cavitation usually occurs in the areas of a diverging lubricated contact gap, which implies negative pressure of lubricant. The cavitation areas can be examined by coupling pressure and fluid density rate in the cavitation area as proposed by Kumar and Booker [38], or by setting negative pressures to some fixed pressure [39]. Kumar and Booker [40] showed that mass conservation seems to be unimportant in determining journal bearing movement, except when the external load applied to the bearing is supported by an incomplete film lubricant. However, the oil flow cannot be examined because the mass is not conserved in the cavitation region. The first algorithm model using finite element cavitation was proposed by Kumar and Booker [38]. Their models do not use the assumption of liquid compressibility as Elrod's algorithm model did. The gas/vapour area was assumed compressible with zero bulk modulus. However, some problems occurred when the mesh or the integration method changed [41]. The aim of this work is to recognize the importance of the theoretical modelling of cavitation and starvation in the analysis of slip/no-slip mixed lubricated mechanisms. The classical theoretical models, the Half Sommerfeld model [42] and the Reynolds model [43] for starvation, are studied for the condition of slip/no-slip engineered contact [44].

1.2. Effect of slip on the cavitation zone

It is interesting to note that when boundary slip is of particular interest in bearing analysis, for most previously published works such as the work of some researchers [24-27, 29-34], the cavitation was ignored, and most of them used a non mass conservation equation. As can be seen in the numerical results below, the cavitation has a significant effect and the exclusion of cavitation in mathematical models may lead to a wrong conclusion. Therefore, to complement the previous findings, an analytical lubrication model with boundary slip that takes the cavitation effect into account is proposed.

The cavitation in lubricated bearings is a critical issue and becomes of particular interest, as discussed in all appended Papers (A, B, C, D, E, F, G, and H). It is worth mentioning that the concept of cavitation was also studied in the work of Fowell et al. [9]. They analytically demonstrated that the cavitation effect is significant in altering the behaviour of the bearing. On this issue, the numerical simulations predicted by the developed model here matched well with this finding (see Fig. 1.1), showing the difference of the hydrodynamic pressure profile predicted by the analysis considering cavitation and without the cavitation (in the following section, the term “no cavitation” is used), respectively, for slip and no-slip conditions.

In this context, cavitation can occur in both a single and double pocket of lubricated sliding contacts. For the case of a single pocket, more discussion can be found in the appended Papers (Paper A, B, F, G and H), while for the case of a double pocket, the characteristics of cavitation is discussed in the appended Papers C, D and E. For these papers, the effect of slip is also of particular interest with respect to the hydrodynamic pressure.

FIGURE 1.1: The pressure distribution as a function of the coordinate in sliding direction for the single pocketed bearing for slip situation (Note: slip is placed on the upper surface of the bearing as shown in the inset, with $h_d = 5.0 \times 10^{-6}$ m, $h_o = 7.0 \times 10^{-6}$ m and $\alpha = 0.02$ m²/s/kg). The detail of α is explained in Section 4 page 53.

FIGURE 1.2: *The pressure distribution as a function of coordinate in sliding direction for the single pocketed bearing for no-slip situation.*

1.3. Summary and outline thesis

This thesis discusses the effect of slip and cavitation on the lubrication performance of parallel sliding contact bearings. The inclusion of the cavitation in pocket bearing analysis is of particular interest in terms of obtaining the representation of the real conditions. Besides, the slip is introduced to reduce the cavitation area in the pocket bearing varying the placement of such a boundary. The thesis consists of Part I and Part II. Part I contains the theoretical approach of the lubrication mechanism of a parallel sliding pocket bearing by slip modification and cavitation, while Part II contains the associated supporting scientific publications.

Part I consists of five chapters. Chapter I concerns the background and a short review of texture either with slip or without slip. Chapter II presents the governing equation for slip modelling as well as cavitation modelling for lubrication analysis. Chapter III discusses the main finding the present research, i.e., the derivation of the lubrication equation based on the Reynolds approach. Cavitation is taken into account in the derived equation. Chapter IV gives the main results of the lubrication study for two main cases, namely the single pocket and the double pocket, by varying the texture geometry, i.e. the inlet length, texture depth and the texture length. The comparison of the effect of cavitation and no cavitation has been made by varying the boundary slip location. Chapter V presents the conclusions and the future work concerning the present work.

Part II contains eight Papers investigating several issues regarding the effect of cavitation and slip for parallel textured pattern. For the single pocket, the results can be found in Papers A, B, F, G, and H, while for the double pocket bearing, the results in more

detail have been presented in the appended Papers C, D, and E. In addition to texture geometry, the slip location on the textured pattern is of particular interest. Papers A and B focus on contact with the full slip. For Paper F, the slip is arranged on the textured edge. For the case in which the slip is applied on the leading edge of the contact, the discussion can be found in the appended Paper G. With regard to the case of the double pocket bearing, the cavitation effect for the case in which the slip is applied on the top edge of the double pockets can be found in Paper D. Finally, for the case in which the slip is applied at the inlet and the edge of the land between the two pockets, the results can be found in Paper E.

Chapter 2

Lubrication with cavitation

In hydrodynamic lubrication, the surfaces form a shallow, converging wedge so that as their relative motion causes lubricant entrainment into the contact, the lubricant becomes pressurized and therefore able to support the load. This way, the hydrodynamic action of the lubricant fully separates the surfaces and the load is carried totally by the lubricant film. The thickness of the film depends on the surface shapes, their speeds, and the properties of the lubricant. Generally, the film thickness of a bearing is of the order of micrometres and the supporting applied pressure in kilo to mega Pascals. This pressure is not high enough to increase the viscosity of the lubricant. It means that any piezo-viscous effects can be ignored.

As is known, cavitation in a bearing is common and directly influences the hydrodynamic pressure and, subsequently, the load support and friction across the contact. In order to address such a cavitation effect, several cavitation models have been proposed by numerous researchers. In this research, the focus is on modifying the Reynolds equation by considering the cavitation model in which the mass conservation along the lubricated contact is fulfilled.

2.1. The classical Reynolds equation

When hydrodynamic lubrication (HL) is simulated numerically, the hydrodynamic pressure generated in the lubricant film is normally modelled with the aid of the Reynolds equation [45]. The Reynolds equation, see Eq. (2.1), is the predominantly used partial differential equation for modelling fluid flow (or more accurately, the fluid pressure) in a full film lubricated contact. The equation is derived by combining two conservation equations of momentum and continuity into a single equation for the fluid pressure, assuming small film thickness relative to the contact length, non-varying pressure across the film thickness, and the dominance of certain viscous terms. When the derivation is conducted, inertia is omitted due to the low Reynolds number in combination with the thin film in the contact region, see

for example [12]. The equation obtained relates the fluid pressure to the rate of the convergence wedge, surface velocities and lubricant viscosity. Equation (2.1) shows the general formulation:

$$\frac{u_m}{2} \frac{\partial(\rho h)}{\partial x_i} - \nabla \cdot \left(\frac{\rho h^3}{12\eta} \nabla p \right) = 0 \quad (2.1)$$

where:

- ρ density of the lubricant
- h lubricant film thickness
- η dynamic viscosity of the lubricant
- p hydrodynamic pressure
- u_m average surface velocity in the direction of motion

2.2. Cavitation models

Vaporous cavitation is the result of a thermodynamic non-equilibrium event when the pressure falls below the vapour pressure of the liquid at the prevalent temperature [44]. Sommerfeld [39] implemented a trigonometric substitution known as the Sommerfeld transformation, which allows solutions of the Reynolds equation for all eccentricities and yields the now equally famous full Sommerfeld curve (Fig. 2.1). The most notable feature of this curve is its negative antisymmetric pressure in the divergent region of the bearing. Nominally, according to the Sommerfeld model, the minimum negative pressure is equal to the maximum positive one. For this model to be correct, the working fluid is assumed to be able to sustain high tensile stresses. It has been known for quite some time [43, 46] that a fluid under certain conditions can sustain elevated levels of tensile stresses. This, however, is not such a realistic expectation in a tribological environment where fluid flow conditions exhibit extreme tangential shear, high rates of heat dissipation, and possibly strongly forming sub atmospheric pressures. These conditions result in a film rupture wherein the resulting cavity, filled by either gas, vapour, or a mixture thereof, has to be factored into the ensuing model.

(a)

(b)

(c)

FIGURE 2.1: Typical pressure distribution in a journal bearing with an eccentric shaft: (a) typical two-dimensional axial circumferential pressure map distribution in a journal bearing; (b) 2D full Sommerfeld curve circumferential cross-section through the symmetry line of (a); and (c) typical circumferential pressure distribution in a convergent divergent clearance with a normal velocity $\pm V$ to the main flow direction [44].

2.2.1. Half Sommerfeld (Gumbel) cavitation model

Cavitation bubbles, whether gaseous or vaporous, may come from within the lubricating film or may be fed by the environment. Regardless, they represent a rupture in the continuity of the liquid film. This discontinuity at the bubble interface presents a challenge in the context of the solution to the Reynolds equation. The solution offered by Sommerfeld [39] did not take into account film rupture and solved for a full film around the circumference (Figs. 2.1 (a) and (b)). This approach allowed sub atmospheric and even negative pressures, no matter how low they were [47]. In 1914, Gumbel [48] was the first to account for the film ‘rupture’. According to his theory, for a steadily loaded bearing operating at constant angular velocity, the rupture originates in the immediate vicinity of the film’s minimum clearance at a predetermined pressure p_{cav} and remains constant at that value for the entire divergent region (Fig. 2.2 (a)). The HS model consists of solving the Reynolds equation with its boundary conditions and then replacing the negative pressures by zero, thus allowing definition of a full fluid zone where pressure is greater than the cavitation pressure and of a cavitated area in which the pressure of the gas liquid mixture is assumed to be zero. The major drawback of this model is that it is non-conservative [49].

FIGURE 2.2: Circumferential pressure development based on film rupture theories: (a) Gumbel [8]; (b) Swift-Stieber [42, 50]; (c) JFO and Floberg; and (d) Elrod [41], and Vijayaraghavan and Keith [51, 52].

2.2.2. The Swift-Stieber (or Reynolds) cavitation model

A better alternative to the half-Sommerfeld condition was offered by Swift [42] and Stieber [50], who worked independently of each other (Fig. 2.2 (b)). In 1932, Swift [15] stated that a zero derivative of the pressure is an appropriate condition for marking the inception of cavitation and considered it to be a ‘stability condition’. Stieber [50] published a full solution in 1933 for a 360-degree journal bearing considering cavitation for a zero tensile strength lubricant. Later, extending the work of [50], Floberg [53] offered different approaches for the zero and non-zero tensile stress during the cavitation phenomena. Stieber, like Swift, assumed a zero-pressure gradient at the start of the cavitation zone, but considered it to be a continuity condition. These two forefathers of cavitation modelling, while approaching it from different angles, came to the same conclusion regarding its inception and development. The cavitation zone formation conditions known as the Swift-Stieber conditions (Fig. 2.2 (b)) are

$$\frac{\partial p}{\partial x} = \frac{\partial p}{\partial y} = 0, \quad p = p_{cav} \quad (2.2)$$

Note that these conditions consider the entire cavitation zone at $p = p_{cav}$ and do not consider variations in pressure within the cavitation zone. The flow rate is indeed given by

$$q = -\frac{h^3}{12\eta} \nabla p + \frac{hs}{2} i, \quad \text{in the full-film zone} \quad (2.3)$$

$$q = \frac{hs}{2} i, \quad \text{in the cavitated area}$$

where i is the unitary vector in the x direction and the continuity of flow at the boundary between full film and cavitation leads to $\nabla p \cdot n = 0$ (n is a normal vector to the boundary) when h is continuous. A practical way of computing the pressure is to discretize the Reynolds equation by finite differences or finite elements schemes, and to solve the associated linear system by setting every negative pressure value to zero at each step of the iterative algorithm. In a sense, this model partially solves the non-conservative aspect of the HS model in numerous applications, especially when the only free boundary corresponds to film rupture. It must not be used for starvation problems with free boundaries of film formation or for contacts with multiple non-connected cavitation areas, for which it is not conservative.

2.2.3. Mass conserving cavitation model

The more elusive problem in dealing with film rupture has been respecting the mass continuity within the confines of the Reynolds equation. Gumbel’s condition [48] and the half Sommerfeld condition did not respect the continuity of mass; they just accounted for the fact that when and where the film ruptured, pressures became sub ambient (negative) and consequently automatically set to p_{cav} in the final iteration (SS/ Gumbel) and every step

iteration (HS). Today, the mass conservation cavitation has been used to analyse cavitation events. Elrod [1981] developed a mass conservation cavitation model. In full film conditions or cavitation conditions, the mass flowing in each bearing section is the same. For this reason, it is necessary to calculate the density in full film and cavitation conditions. Braun [44] decreases the mass conservation cavitation equation by describing the presence of gas trapped in a fluid (see Figure 2.3). In this two-dimensional space the continuity equations at the upstream (1) and downstream (2) locations of the boundary can be written as

$$\dot{m}_{x,i} dz - \dot{m}_{z,i} dx = \dot{m}_{cav} dz, \quad \text{where } i = 1, 2 \quad (2.4)$$

Ignoring the inertia terms, and considering constant properties, the simplified form of the NSE yield expressions for velocities u and w in the x - and z directions respectively. Upon the integration of these velocities across the fluid film between the moving and stationary members, one obtains the following flows

$$\begin{aligned} \dot{m}_{x1} &= \frac{U h_1}{2} - \frac{h_1^3}{12\eta} \left(\frac{\partial p}{\partial x} \right)_1 \\ \dot{m}_{z1} &= \frac{h_1^3}{12\eta} \left(\frac{\partial p}{\partial z} \right)_1 \\ \dot{m}_{cav} &= \frac{U h_1}{2} \end{aligned} \quad (2.5)$$

Introduction of the above mass flow rate expressions into the continuity equation on the bubble boundary at location 1 (which is upstream) results in the following condition

$$-\frac{h_1^3}{12\eta} \left(\frac{\partial p}{\partial x} \right)_1 + \frac{h_1^3}{12\eta} \left(\frac{\partial p}{\partial z} \right)_1 \left(\frac{dx}{dz} \right)_1 = 0 \quad (2.6)$$

Since no tensile stresses (negative pressures) are permitted in the film, to satisfy the bubble boundary continuity equation it is necessary that $\partial p / \partial x|_1 = \partial p / \partial z|_1 = 0$. These now become the boundary conditions for the inception of the cavitation bubble. Additionally, one has to specify the cavitation pressure p_{cav} that remains constant throughout the cavitation region. Following the same procedure for control volume 2 (which is downstream) (see Fig. 2.3), one can write

$$\begin{aligned}
\dot{m}_{x2} &= \frac{Uh_2}{2} - \frac{h_2^3}{12\eta} \left(\frac{\partial p}{\partial x} \right)_2 \\
\dot{m}_{z2} &= \frac{h_2^3}{12\eta} \left(\frac{\partial p}{\partial z} \right)_2 \\
\dot{m}_{cav} &= \frac{Uh_1}{2}
\end{aligned} \tag{2.7}$$

Further, applying the continuity equation for control volume 2 (which is downstream) one obtains a mass conservation condition for the reformation of the boundary of the downstream bubble.

$$-\frac{h_2^3}{12\eta} \left(\frac{\partial p}{\partial x} \right)_2 + \frac{h_2^3}{12\eta} \left(\frac{\partial p}{\partial z} \right)_2 \left(\frac{dx}{dz} \right)_2 + \frac{Uh_2}{2} = \frac{Uh_1}{2} \tag{2.8}$$

FIGURE 2.3: (a) Geometry of the oil cavity used in the JFO type theory. Typical cavitation bubbles for a zero tensile strength liquid with mass continuity control volumes at the upstream inception (1) and downstream reformation (2) and (b) Geometry is for one repetitive cell for a liquid with finite tensile strength for a finite (or infinite numbers of such cells) with the width of the cell $W = L/n$ [44].

2.3. Previous studies in hydrodynamic lubrication with cavitation

Introducing a pocket as well as boundary slip at a bearing surface is becoming popular in oil lubricated systems because it has proven to improve load support and reduce friction. It has been shown experimentally that surface texturing enhances load support for mechanical systems with parallel sliding contacts [1-2], cylinder liner contacts [4] and mechanical seals [3]. With the introduction of boundary slip, a high load support was highlighted by some

works [17, 18, 25, 28, 54]. Recently, in addition to surface texturing, the use of an artificial slip surface is also of interest with respect to lubrication. The earliest work on artificial complex slip/no-slip for lubricated parallel sliding devices was reported by Salant and Fortier [24]. They focused on the ability of artificial slip to improve the load support and reduce the friction force in the absence of the wedge effect. Subsequently, several studies were published confirming the findings of Salant and Fortier [24]. Guo and Wong [55] confirmed that the introduction of the so-called tailored boundary slip on the stationary surface of a slider leads to a net pressure build-up. Similarly, Wu et al. [25] studied the behaviour of a slider bearing with mixed slip surface condition and their results indicated that convergent, parallel, and even a divergent wedge can provide hydrodynamic load support. Rao [56] employed boundary slip on a stationary surface having a single groove at a slider and a journal bearing and then calculated the pressure and shear stress distribution. Regarding the slip modelling, Navier postulated that the magnitude of the slip velocity is proportional to the magnitude of the shear rate experienced by the fluid at the solid surface, and therefore the so-called Navier-slip model is popular now in modelling the boundary slip. Such a slip model has been adopted by numerous tribologists. Based on a survey of literature, the modified Reynolds equation with the Navier slip boundary condition (see Section 2.4) was solved numerically with regard to the effect of wall slip and a single groove texture simultaneously. In a recent publication, Tauviquirrahman et al. [31-33] explored the possibility of boundary slip and texture in order to improve the performance characteristics. Their results indicated that the combined slip/ texture pattern increased the load-carrying capacity and decreased friction.

Theoretical analysis of textured surfaces has been carried out using the Reynolds equation [8, 9, 57, 58] and CFD models based on Navier-Stokes [3, 17, 18, 28, 25, 54]. Two texture modes were used, namely micro and macro roughness [5-7, 10-11, 59]. The first mode is based on a single cell texture configuration [5-6, 9, 11, 54, 60, 61] and the second one is based on a multi cell texture pattern [8, 10, 57, 60]. For the two approaches and texturing modes mentioned, cavitation was not considered, so their results may be questionable.

Following the insight of the significance of cavitation, interesting results have been explored by some researchers, for example Fowell et al. [9], Gherca et al. [62], Mao et al. [63] and Galda et al. [64]. Fowell et al. [9], based on Reynolds theory, found that the inlet suction mechanism in textured hydrodynamic bearings with pockets reduce friction. Recent interesting work regarding the effect of texturing was performed by Gherca et al. [62]. They presented, using finite element modelling, the results of hydrodynamic thrust bearings operating both in the steady-state and the transient lubrication regime. It was found that the performance of the bearing is not the same when texturing is applied on the rotor and on the stator. Mao et al. [63] proposed an analytical model coupled with the wedge effect for predicting cavitation characteristics under sliding motion. The optimization was also performed to improve the performance of lubricated textured components.

Galda et al. [46] found based on an experimental approach that dimples improved the tribological characteristics under starved lubrication conditions at low sliding speeds. It was also found that the beneficial effect of dimple presence was more significant at a lower load.

In most of the previous works the load support generation mechanisms were demonstrated using a simple simulation approach for a one-dimensional parallel slider

bearing having a single cell texture without cavitation. Wall slip, surface texturing and cavitation have a significant effect on the bearing characteristics. Bayada and Meurisse [28] showed the importance of the cavitation model in the analysis of slip/no-slip hydrodynamic contacts. Fortier et al. [65] presented the numerical analysis of a journal bearing using a modified slip length model (see Section 2.4) and mass conserving algorithm. The results illustrated a high load support and low friction using a heterogeneous slip/no-slip bearing surface. On the basis of the modified slip length model and cavitation model, Wang and Lu [35] presented results for a two-dimensional sleeve bearing.

It is interesting to note that when the effect of boundary slip and surface texturing were studied, for example by [24-27, 30-32], cavitation was ignored and most of them used a non-mass conservation cavitation equation. As can be seen in the results below, cavitation has a significant effect and the exclusion of cavitation in the mathematical model may lead to wrong conclusions. A few publications considered texturing and slip in lubricated contacts considering cavitation. The aim of this study is to develop a novel lubrication model including cavitation based on a Reynolds approach. In this thesis, the texturing configurations are single and double grooved pockets in the inlet of bearing. The correlation between the texturing parameters with slip and cavitation on the load-carrying capacity is of particular interest.

2.4. Slip models

Early studies [24, 65] found experimental evidence of the boundary slip occurring at a liquid-solid interface. During the past decade, with the development of advanced experimental techniques, numerous experiments have shown that boundary slip can occur at both the hydrophobic surface [28, 30, 55-56] and the hydrophilic surface [66-69]. For most hydrophilic surfaces, however, no-slip occurs.

The great challenge for a hydrophobic surface, from the perspective of a numerical simulation, is choosing a model for the slip boundary. This is because the hydrodynamic behaviour of lubricated contacts is affected mainly by the boundary conditions of the lubricant that will provide lubrication. From a numerical point of view, there are two main wall slip models that have been adopted to describe the boundary slip, namely the slip length model (SLM) and the critical shear stress model (CSSM). In this section, an overview of slip models in published works is presented.

2.4.1. Slip length model (SLM)

The slip length model (SLM) predicts that the slip velocity is proportional to the local shear rate; the proportionality constant is called slip length – the distance below the solid surface where the velocity extrapolates linearly to zero (see Fig. 2.4). In other words, in the slip length model, the velocity at which the liquid slips against the solid surface u_s , the shear stress τ , and the slip length b are related by the following equation

$$\tau = \frac{u_s}{b} \eta \quad (2.9)$$

Here, the larger the value of b , the greater the slip. The SLM implies that the slip length is independent of the shear rate, which is not, however, supported by some of the experiments [28, 67-68, 70] and molecular dynamic simulations [14, 71-72]. First, it was reported that the slip velocity increases in a strong nonlinear manner with the shear rate, especially at high shear rates. In other words, the slip length model can only describe the slip behaviour when the shear rate is moderate. Second, it was found that a critical shear stress exists at the solid/liquid interface and that boundary slip occurs only after the surface shear stress exceeds the critical shear stress.

FIGURE 2.4: Couette velocity profiles between parallel sliding surfaces: (a) no-slip, the shear rate is u_w/h (b) slip at a lubricant-solid interface with slip length, b , at the stationary surface.

2.4.2. Critical shear stress model (CSSM)

The experimental manifestation of boundary slip at high shear rates shows the existence of a critical shear stress: this is the critical shear stress model. Figure 2.5 shows an ideal critical shear stress model. This concept was first proposed by Smith [73-74] for lubricants and was later confirmed by Bair and Winer [75-77] for metal (hydrophilic). The critical shear stress assumes that the boundary slip takes place only after the surface shear stress, τ , reaches the critical value, τ_c , where the surface shear stress is constant (shear rate independent) but the apparent shear rate can reach any value (the true shear rate equals the critical threshold). When the surface shear stress is below the critical shear stress, no-slip occurs.

FIGURE 2.5: Schematic of a stress-rate curve of the critical shear stress model.

The critical shear stress was found to depend on surface wettability, surface roughness, and fluid viscosity, etc. The roughness effect on the critical shear stress was not so clear. Some researchers [28, 55, 77] demonstrated that surface roughness inhibits boundary slip or increases the critical shear stress, but others [76] reported that it increases boundary slip. The critical shear stress may decrease depending on the angle of the surface contact [77]. With respect to the surface wettability property, usually the better the hydrophobicity of the surface, the lower the critical shear stress. The critical shear stress of the superhydrophobic surface is as low as 0.33 Pa [28, 77-78], which can be considered as a perfect (ideal) slip surface. This value is much lower than the reported critical shear stress of the interface for oil and steel that ranges from 0.16 to 8 MPa [79]. There are two kinds of critical shear stress criteria. The first is the so-called two-component slip model; the model proposed by Spikes & Granick [80] for slip in which a critical shear stress criterion is broadened to incorporate both a critical shear stress and a constant slip length criterion. Slip is envisaged to occur only when a critical surface shear stress is reached, and once slip begins, it takes place at a constant slip length. In this case, the shear stress occurring when the wall slip takes place is given by

$$\tau_c = \tau_{oc} + \frac{\eta}{b} u_s \quad (2.10)$$

where τ_{oc} is the critical threshold shear stress for slip, η is the dynamic viscosity of the liquid, b is the slip length (once slip begins), and u_s is the slip velocity. This model is quite similar to the concept proposed by Salant and Fortier [24]. They assumed that fluid slip occurs when the critical shear stress is exceeded and the resulting slip velocity is proportional to the difference between the shear stress and the critical value.

The second type of critical shear stress model is the slip model as a function of local fluid pressure. This model was investigated by several groups [73-74, 81, 82-85] and given as

$$\tau_c = \tau_{oc} + kp \quad (2.11)$$

where τ_c is the critical shear stress ($\tau_c = \eta(\partial u/\partial z)_c$ for a Newtonian fluid), τ_{oc} is the initial shear strength when the film pressure equals zero, p is the fluid pressure, and k is the critical, local, interfacial friction coefficient (sometimes quoted as the proportionality coefficient). The parameters τ_{oc} and p depend on the chemical composition of the fluid and the temperature. The proportionality coefficient, k , has been investigated in several ways and by a few researchers [83-86] with results that range from about 0.007 to 0.15 and the variation is found to be temperature dependent. This type of CSSM was used for flow analysis in lubrication mechanics based on the difference method [87] and the finite element method [26, 75, 78-80, 88-90].

It should be noted that the critical shear stress model is considered as a computational time consuming model, but with a clear conclusion, as discussed by Wu and his group [25, 82-83, 89-90] and Tauviqirrahman et al. [32-33]. For this reason, in the present study, the lubrication theory including slip adopts the Navier-slip model instead of the critical shear stress model.

2.5. Modified Reynolds equation with cavitation

In this research, the main basis of lubrication theory comes from the Reynolds theory. There are numerous works regarding the lubrication analysis with the main point of cavitation, the published work of Fowell et al. [9] for example. As a note, the authors have proposed the modified Reynolds equation pointing out the inclusion of the cavitation phenomena for the main case of the no-slip condition in the case of a sliding contact with micro-cavity (i.e. single texture cell). Below is the explanation of the lubrication model described by Fowell et al. [9] that is used as the basis of the present research.

2.5.1. Load support

Figure 2.6 depicts the schematic representation of the parallel pocketed bearing used by Fowell et al. [9]. In this configuration, cavitation is allowed to occur in the pocketed region. The cavitation is modelled by adding cavitation pressure in part of the zone during the derivation of the Reynolds equation.

FIGURE 2.6: Schematic diagram of a parallel sliding pocketed bearing.

From the first-order Reynolds equation, the flow is given by

$$q = \frac{Uh}{2} - \frac{h^3}{12\eta} \frac{dp}{dx} \quad (2.12)$$

For regions of the bearing having parallel surfaces, the film thickness h is constant, so from Eq. (2.12) the dp/dx must also be constant, i.e. the pressure variation must be linear. Equation (2.12) can thus be modified for each region of the bearing to:

$$q_{AB} = \frac{Uh_o}{2} - \frac{h_o^3}{12\eta} \frac{dp}{dx} = \frac{Uh_o}{2} - \frac{h_o^3}{12\eta} \left(\frac{p_2 - p_{atm}}{a} \right) \quad (2.13)$$

$$q_{BC} = \frac{Uh_p}{2} - \frac{h_p^3}{12\eta} \frac{dp}{dx} = \frac{Uh_p}{2} - \frac{h_o^3}{12\eta} \left(\frac{p_3 - p_2}{b} \right) \quad (2.14)$$

$$q_{CD} = \frac{Uh_o}{2} - \frac{h_o^3}{12\eta} \frac{dp}{dx} = \frac{Uh_o}{2} - \frac{h_o^3}{12\eta} \left(\frac{p_{atm} - p_3}{c} \right) \quad (2.15)$$

where, by continuity of flow, $q_{AB} = q_{BC} = q_{CD}$. Equating the flows in the inlet and outlet lands, $q_{AB} = q_{CD}$

$$\frac{Uh_o}{2} - \frac{h_o^3}{12\eta} \left(\frac{p_2 - p_{atm}}{a} \right) = \frac{Uh_o}{2} - \frac{h_o^3}{12\eta} \left(\frac{p_{atm} - p_3}{c} \right) \quad (2.16)$$

$$\left(\frac{p_2 - p_{atm}}{a} \right) = \left(\frac{p_{atm} - p_3}{c} \right) \quad (2.17)$$

It is now necessary to consider the two possible cases shown in Fig. 2.6: (i) when the pressure is at the pocket inlet, p_2 , remains above the cavitation pressure p_{cav} and (ii) when the pressure is at the entry to the pocket, (location B) falls to the cavitation pressure so as to produce a finite zone in the pocket having pressure p_{cav} .

2.5.1.1. No cavitation in the pocket

In the non-cavitating case, the pressure p_2 at B is unknown. Equating the flow in the inlet land and pocket, $q_{AB} = q_{BC}$

$$\frac{Uh_o}{2} - \frac{h_o^3}{12\eta} \left(\frac{p_2 - p_{atm}}{a} \right) = \frac{Uh_p}{2} - \frac{h_p^3}{12\eta} \left(\frac{p_3 - p_2}{b} \right) \quad (2.18)$$

Combining Eqs. (2.17) and (2.18) to eliminate p_3 gives

$$p_2 = p_{atm} + \frac{6U\eta(h_p - h_o)ab}{(ah_p^3 + ch_p^3 + bh_o^3)} \quad (2.19)$$

and thus, from Eq. (2.17)

$$p_3 = p_{atm} + \frac{6U\eta(h_p - h_o)cb}{(ah_p^3 + ch_p^3 + bh_o^3)} \quad (2.20)$$

Since the pressure variations in the inlet and outlet lands and in the pocket are linear, each of the three regions of the bearing has a triangular pressure distribution that can be simply integrated to determine the load support. The total normal force is thus given by

$$W = W_{AB} + W_{BC} + W_{CD} - p_{atm}(a + b + c) \quad (2.21)$$

$$W = \frac{6U\eta(h_p - h_o)}{h_p^3} \frac{b}{2} \left(\frac{c^2 - ab - a^2 + bc}{a + c + b \frac{h_o^3}{h_p^3}} \right) \quad (2.22)$$

2.5.1.2. Cavitation in the pocket

If Eq. (2.19) predicts that p_2 is less than the cavitation pressure, cavitation will occur in the bearing and a slightly different analysis is needed. For this case, $p_2 = p_{cav}$, so equating flow in the inlet land and pocket, $q_{AB} = q_{BC}$ gives

$$\frac{Uh_o}{2} - \frac{h_o^3}{12\eta} \left(\frac{p_{cav} - p_{atm}}{a} \right) = \frac{Uh_p}{2} - \frac{h_p^3}{12\eta} \left(\frac{p_3 - p_{cav}}{X_b} \right) \quad (2.23)$$

where X_b is the fraction of the pocket filled with lubricant, i.e., the region $B'C$ (see Fig. 2.6). Since $p_2 = p_{cav}$, Eq. (2.17) becomes

$$\left(\frac{p_{cav} - p_{atm}}{a} \right) = \left(\frac{p_{atm} - p_3}{c} \right) \quad (2.24)$$

which gives

$$p_3 = p_{atm} + \frac{c}{a} (p_{atm} - p_{cav}) \quad (2.25)$$

Equations (2.23) and (2.25) can then be combined to determine X_b

$$X_b = \frac{(p_3 - p_{cav})h_p^3}{h_o^3 \frac{p_{cav} - p_{atm}}{a} + 6U\eta(h_p - h_o)} \quad (2.26)$$

For the cavitating case, pressure has to be separately integrated over the cavitated and non-cavitated regions of the pocket, so the total load support is given by

$$W = W_{AB} + W_{BB'} + W_{B'C} + W_{CD} - p_{atm}(a + b + c) \quad (2.27)$$

$$W = \frac{1}{2a} \left[c^2 - \frac{h_o^3}{h_p^3} (X_b)^2 - a^2 - 2ab \right] (p_{atm} - p_{cav}) + \quad (2.28)$$

$$\frac{6U\eta(X_b)^2}{2h_p^3} (h_p - h_o)$$

where

$$X_b = \frac{\left[\left(1 + \frac{c}{a} \right) (p_{atm} - p_{cav}) \right] h_p^3}{6U\eta(h_p - h_o) - h_o^3 \frac{p_{atm} - p_{cav}}{a}} \quad (2.29)$$

2.5.2. Friction force

The shear stress at the lower surface is found from the product of the velocity gradient at this surface ($z = 0$) and the viscosity. This gives the standard expression [4],

$$\tau_{z=0} = -\frac{U\eta}{h} - \frac{h}{2} \frac{\partial p}{\partial x} \quad (2.30)$$

For each part of the bearing this equation can be integrated to determine the friction force. This is very straightforward for parallel surfaces since h and dp/dx are constant. Thus over the inlet land

$$F_{AB} = -\left(\frac{U\eta}{h} + \frac{h}{2} \frac{\partial p}{\partial x}\right)a \quad (2.31)$$

The signs of both terms are negative since the Couette term always opposes the motion of the lower surface, as does the Poiseuille term when dp/dx is positive.

2.5.2.1. No cavitation in the pocket

The friction force F over the surface for the case in which cavitation is not considered, can be expressed as follows:

$$F = F_{AB} + F_{BC} + F_{CD} \quad (2.32)$$

$$F = -U\eta \left[\left(\frac{a}{h_o} + \frac{c}{h_o} + \frac{b}{h_p} \right) + \left(\frac{3(h_p - h_o)^2 b(a+c)}{ch_p^3 + ah_p^3 + bh_o^3} \right) \right] \quad (2.33)$$

2.5.2.2. Cavitation in the pocket

The friction force F over the surface for the case in which cavitation is considered, can be expressed as follows:

$$F = F_{AB} + F_{BB'} + F_{B'C} + F_{CD} \quad (2.34)$$

Each friction component can be described as shown in Eq. (2.34) In the cavitated region, the friction is assumed to be zero. Thus

$$F = -\left[U\eta \left(\frac{a}{h_o} + \frac{c}{h_o} + \frac{b}{h_p} \right) + \left(\frac{h_p}{2} - \frac{h_o}{2} \right) \right] \left[\left(1 + \frac{c}{a} \right) (p_o - p_c) \right] \quad (2.35)$$

2.6. Summary

In this section, the lubrication theory regarding the mathematical equation based on the Reynolds theory is elaborated. Modifications of the Reynolds theory are discussed in more detail by introducing slip and cavitation. The analytical equations for the lubrication performance of the bearing viz load support and friction force are presented.

Chapter 3

Modelling

A lot of attention has recently been paid to hydrodynamic lubricated sliding contacts with a surface texture and boundary slip. Many attempts have been conducted by researchers to propose the optimal design for a step bearing leading to a high load support, but low friction. Regarding slip, Choo et al. [17] showed that liquid slip can be used to considerably reduce friction under full film, hydrodynamic lubrication. Tauviqirrahman et al. [32] demonstrated that partial texturing gives even better results than full texturing. However, in comparison with a flat complex slip surface, a partially textured surface is still less efficient to enhance load support, even if this textured configuration is combined with a slip condition. Rahmani et al. [58] examined the step bearing analytically by considering the variations of hydrostatic pressure at the bearing borders in order to find circumstances in which an optimum Rayleigh step can be introduced concerning load capacity, friction force, friction coefficient, and lubricant flow rate. Shukla and Deheri showed [91] that for a better performance of the bearing system, the slip velocity should be minimized. Naduvinamani and Siddangouda [92] showed that a negatively skewed surface roughness pattern increases the load-carrying capacity and decreases the coefficient of friction, whereas the opposite effects were found for the positively skewed surface roughness pattern. Rao et al. [93] stated that partially textured slip is effective in the case of a parallel slider and a journal bearing with respect to load capacity. Yu et al. [94] studied the theoretic influence of recess depth and velocity on the lift-of hydrodynamic pressure of a bearing according to the lubrication theory.

In most of the above-mentioned works, the hydrodynamic analysis ignored cavitation, which may significantly affect the results. In the present study, an improvement in the performance of slider bearings is examined based on the slip phenomenon by taking cavitation into account. The modified form of the Reynolds equation containing the cavitation effect is derived, in which the wall slip is allowed to occur in a textured bearing.

3.1. Modified Reynolds equation with slip, texture and cavitation

In the present study, a lubrication model is proposed based on the Reynolds equation. The isoviscous one-dimensional Reynolds equation is derived from the Navier-Stokes equation which assumes a laminar flow by ignoring inertia effects:

$$\begin{aligned}\frac{\partial p}{\partial x} &= \frac{\partial}{\partial z} \left(\eta \frac{\partial u}{\partial z} \right) \\ \frac{\partial p}{\partial z} &= 0\end{aligned}\tag{3.1}$$

Equation (3.1) must be integrated to obtain the fluid velocity. Before integration, the boundary conditions need to be defined. Slip will occur only in areas where the surface has been treated and when the shear stress exceeds the critical shear stress τ_c . When these two requirements are met, the resulting slip velocity is proportional to the difference between the value of shear stress and critical shear stress, with the proportionality factor α_h for the stationary surface and α_s for the moving surface. Assuming that the critical shear stress is zero, the boundary conditions are:

$$\begin{aligned}\text{at } z=0, u &= U + \alpha_s \eta \left. \frac{\partial u}{\partial z} \right|_{z=0} \\ \text{at } z=h, u &= -\alpha_h \eta \left. \frac{\partial u}{\partial z} \right|_{z=h}\end{aligned}\tag{3.2}$$

Equation (3.1) is integrated twice in x -direction,

$$\begin{aligned}\frac{\partial^2 u}{\partial z^2} &= \frac{1}{\eta} \frac{\partial p}{\partial x} \\ u &= \frac{1}{2\eta} \frac{\partial p}{\partial x} z^2 + C_1 z + C_2\end{aligned}\tag{3.3}$$

The boundary condition:

➤ at $z = 0$,

$$u = U + \alpha_s \eta \left. \frac{\partial u}{\partial z} \right|_{z=0}$$

$$U + \alpha_s \eta \frac{\partial u}{\partial z} = \frac{1}{2\eta} \frac{\partial p}{\partial x} z^2 + C_1 z + C_2$$

$$U + \alpha_s \eta \left(\frac{1}{\eta} \frac{\partial p}{\partial x} z + C_1 \right) = \frac{1}{2\eta} \frac{\partial p}{\partial x} z^2 + C_1 z + C_2$$

$$U + \alpha_s z \frac{\partial p}{\partial x} + \alpha_s \eta C_1 = \frac{1}{2\eta} \frac{\partial p}{\partial x} z^2 + C_1 z + C_2$$

$$U + \alpha_s (0) \frac{\partial p}{\partial x} + \alpha_s \eta C_1 = \frac{1}{2\eta} \frac{\partial p}{\partial x} (0)^2 + C_1 (0) + C_2$$

$$U + \alpha_s \eta C_1 = C_2$$

➤ at $z = h$,

$$u = -\alpha_h \eta \frac{\partial u}{\partial z} \Big|_{z=h}$$

$$-\alpha_h \eta \frac{\partial u}{\partial z} = \frac{1}{2\eta} \frac{\partial p}{\partial x} z^2 + C_1 z + C_2$$

$$-\alpha_h \eta \left(\frac{1}{\eta} \frac{\partial p}{\partial x} z + C_1 \right) = \frac{1}{2\eta} \frac{\partial p}{\partial x} z^2 + C_1 z + C_2$$

$$-\alpha_h z \frac{\partial p}{\partial x} - \alpha_h \eta C_1 = \frac{1}{2\eta} \frac{\partial p}{\partial x} z^2 + C_1 z + C_2$$

$$-\alpha_h h \frac{\partial p}{\partial x} - \alpha_h \eta C_1 = \frac{h^2}{2\eta} \frac{\partial p}{\partial x} + h C_1 + U + \alpha_s \eta C_1$$

$$-\alpha_h \eta C_1 - h C_1 - \alpha_s \eta C_1 = \left(\frac{h^2}{2\eta} + \alpha_h h \right) \frac{\partial p}{\partial x} + U$$

$$-(h + \eta(\alpha_h + \alpha_s)) C_1 = \left(\frac{h^2}{2\eta} + \alpha_h h \right) \frac{\partial p}{\partial x} + U$$

$$-C_1 = \left(\frac{h^2 + 2\alpha_h \eta h}{2\eta} \frac{\partial p}{\partial x} \frac{1}{h + \eta(\alpha_h + \alpha_s)} \right) + \frac{U}{h + \eta(\alpha_h + \alpha_s)}$$

$$C_1 = - \left(\frac{h}{2\eta} \frac{\partial p}{\partial x} \frac{h + 2\alpha_h \eta}{h + \eta(\alpha_h + \alpha_s)} + \frac{U}{h + \eta(\alpha_h + \alpha_s)} \right) \quad (3.4)$$

Determination of C_2 value:

$$C_2 = U + \alpha_s \eta C_1$$

$$\begin{aligned}
 C_2 &= U + \alpha_s \eta \left[- \left(\frac{h}{2\eta} \frac{\partial p}{\partial x} \frac{h + 2\alpha_h \eta}{h + \eta(\alpha_h + \alpha_s)} + \frac{U}{h + \eta(\alpha_h + \alpha_s)} \right) \right] \\
 &= U - \frac{h}{2\eta} \frac{\partial p}{\partial x} \frac{\alpha_s \eta (h + 2\alpha_h \eta)}{h + \eta(\alpha_h + \alpha_s)} - \frac{U \alpha_s \eta}{h + \eta(\alpha_h + \alpha_s)} \\
 &= \frac{U (h + \eta(\alpha_h + \alpha_s)) - U \alpha_s \eta}{h + \eta(\alpha_h + \alpha_s)} - \frac{h}{2\eta} \frac{\partial p}{\partial x} \frac{\alpha_s \eta (h + 2\alpha_h \eta)}{h + \eta(\alpha_h + \alpha_s)} \\
 &= \frac{Uh + U \alpha_h \eta + U \alpha_s \eta - U \alpha_s \eta}{h + \eta(\alpha_h + \alpha_s)} - \frac{h}{2\eta} \frac{\partial p}{\partial x} \frac{\alpha_s \eta (h + 2\alpha_h \eta)}{h + \eta(\alpha_h + \alpha_s)} \\
 &= \frac{Uh + U \alpha_h \eta}{h + \eta(\alpha_h + \alpha_s)} - \frac{h}{2\eta} \frac{\partial p}{\partial x} \frac{\alpha_s \eta (h + 2\alpha_h \eta)}{h + \eta(\alpha_h + \alpha_s)} \\
 &= U \frac{h + \alpha_h \eta}{h + \eta(\alpha_h + \alpha_s)} - \frac{h}{2\eta} \frac{\partial p}{\partial x} \frac{\alpha_s \eta (h + 2\alpha_h \eta)}{h + \eta(\alpha_h + \alpha_s)} \tag{3.5}
 \end{aligned}$$

Substituting Eqs. (3.4) and (3.5) in Eq. (3.3) so that the flow velocity in the x -direction is obtained

$$\begin{aligned}
 u &= \frac{1}{2\eta} \frac{\partial p}{\partial x} z^2 - \left(\frac{U}{h + \eta(\alpha_h + \alpha_s)} + \frac{h}{2\eta} \frac{\partial p}{\partial x} \frac{h + 2\alpha_h \eta}{h + \eta(\alpha_h + \alpha_s)} \right) z \\
 &\quad + U \frac{h + \alpha_h \eta}{h + \eta(\alpha_h + \alpha_s)} - \frac{h}{2\eta} \frac{\partial p}{\partial x} \frac{\alpha_s \eta (h + 2\alpha_h \eta)}{h + \eta(\alpha_h + \alpha_s)} \tag{3.6}
 \end{aligned}$$

the flow rate per unit width in the x -direction is

$$q_x = \int_0^h u \, dz$$

$$\begin{aligned}
&= \int_0^h \left\{ \frac{1}{2\eta} \frac{\partial p}{\partial x} z^2 - \left(\frac{h}{2\eta} \frac{\partial p}{\partial x} \frac{h+2\alpha_h\eta}{h+\eta(\alpha_h+\alpha_s)} + \frac{U}{h+\eta(\alpha_h+\alpha_s)} \right) z \right. \\
&\quad \left. + U \frac{h+\alpha_h\eta}{h+\eta(\alpha_h+\alpha_s)} - \frac{h}{2\eta} \frac{\partial p}{\partial x} \frac{\alpha_s\eta(h+2\alpha_h\eta)}{h+\eta(\alpha_h+\alpha_s)} \right\} dz \\
&= \frac{1}{6\eta} \frac{\partial p}{\partial x} z^3 - \frac{U}{2} \frac{z^2}{h+\eta(\alpha_h+\alpha_s)} - \frac{h}{4\eta} \frac{\partial p}{\partial x} \frac{h+2\alpha_h\eta}{h+\eta(\alpha_h+\alpha_s)} z^2 \\
&\quad + U \frac{h+\alpha_h\eta}{h+\eta(\alpha_h+\alpha_s)} z - \frac{h}{2\eta} \frac{\partial p}{\partial x} \frac{\alpha_s\eta(h+2\alpha_h\eta)}{h+\eta(\alpha_h+\alpha_s)} z \Big|_0^h \\
&= \left(\frac{h^3}{6\eta} \frac{\partial p}{\partial x} - \frac{h^3}{4\eta} \frac{\partial p}{\partial x} \frac{h+2\alpha_h\eta}{h+\eta(\alpha_h+\alpha_s)} - \frac{h^2}{2\eta} \frac{\partial p}{\partial x} \frac{\alpha_s\eta(h+2\alpha_h\eta)}{h+\eta(\alpha_h+\alpha_s)} \right) \\
&\quad + \left(U \frac{h^2+h\alpha_h\eta}{h+\eta(\alpha_h+\alpha_s)} - \frac{U}{2} \frac{h^2}{h+\eta(\alpha_h+\alpha_s)} \right)
\end{aligned}$$

$$\int_0^h u dz = a + b, \text{ where:}$$

$$\begin{aligned}
a &= \left(\frac{2h^3}{12\eta} - \frac{3h^3}{12\eta} \frac{h+2\alpha_h\eta}{h+\eta(\alpha_h+\alpha_s)} - \frac{6h^2}{12\eta} \frac{\alpha_s\eta(h+2\alpha_h\eta)}{h+\eta(\alpha_h+\alpha_s)} \right) \frac{\partial p}{\partial x} \\
b &= \left(U \frac{h^2+h\alpha_h\eta}{h+\eta(\alpha_h+\alpha_s)} - \frac{U}{2} \frac{h^2}{h+\eta(\alpha_h+\alpha_s)} \right)
\end{aligned}$$

➤ Determination of (a) value

$$\begin{aligned}
a &= \frac{h^3}{12\eta} \frac{\partial p}{\partial x} \left\{ 2 - \frac{3(h+2\alpha_h\eta)}{h+\eta(\alpha_h+\alpha_s)} - \frac{6\alpha_s\eta(h+2\alpha_h\eta)}{h(h+\eta(\alpha_h+\alpha_s))} \right\} \\
&= \frac{h^3}{12\eta} \frac{\partial p}{\partial x} \left\{ \frac{2[h(h+\eta(\alpha_h+\alpha_s))] - 3h(h+2\alpha_h\eta) - 6\alpha_s\eta(h+2\alpha_h\eta)}{h(h+\eta(\alpha_h+\alpha_s))} \right\} \\
&= \frac{h^3}{12\eta} \frac{\partial p}{\partial x} \left\{ \frac{2h^2 + 2h\eta\alpha_h + 2h\eta\alpha_s - 3h^2 - 6h\eta\alpha_h - 6h\alpha_s\eta - 12\eta^2\alpha_h\alpha_s}{h(h+\eta(\alpha_h+\alpha_s))} \right\}
\end{aligned}$$

$$\begin{aligned}
&= \frac{h^3}{12\eta} \frac{\partial p}{\partial x} \left\{ \frac{-h^2 - 4h\eta\alpha_h - 4h\eta\alpha_s - 12\eta^2\alpha_h\alpha_s}{h(h+\eta(\alpha_h+\alpha_s))} \right\} \\
&= -\frac{h^3}{12\eta} \frac{\partial p}{\partial x} \left\{ \frac{h^2 + 4h\eta(\alpha_h+\alpha_s) + 12\eta^2\alpha_h\alpha_s}{h(h+\eta(\alpha_h+\alpha_s))} \right\}
\end{aligned}$$

➤ Determination of (b) value

$$\begin{aligned}
b &= U \left(\frac{h^2 + h\alpha_h\eta}{h+\eta(\alpha_h+\alpha_s)} - \frac{h^2}{2(h+\eta(\alpha_h+\alpha_s))} \right) \\
&= U \left(\frac{2h^2 + 2h\alpha_h\eta - h^2}{2(h+\eta(\alpha_h+\alpha_s))} \right) \\
&= \frac{U}{2} \left(\frac{h^2 + 2h\alpha_h\eta}{h+\eta(\alpha_h+\alpha_s)} \right)
\end{aligned}$$

$$\int_0^h u dz = a + b$$

$$\begin{aligned}
\int_0^h u dz &= - \left(\frac{h^3}{12\eta} \frac{\partial p}{\partial x} \frac{h^2 + 4h\eta(\alpha_h+\alpha_s) + 12\eta^2\alpha_h\alpha_s}{h(h+\eta(\alpha_h+\alpha_s))} \right) \\
&\quad + \frac{U}{2} \frac{\partial}{\partial x} \left(\frac{h^2 + 2h\alpha_h\eta}{h+\eta(\alpha_h+\alpha_s)} \right)
\end{aligned} \tag{3.7}$$

For $q_x = -hu \frac{\partial h}{\partial x}$

$$\begin{aligned}
u &= \frac{1}{2\eta} \frac{\partial p}{\partial x} h^2 - \left(\frac{h}{2\eta} \frac{\partial p}{\partial x} \frac{h+2\alpha_h\eta}{h+\eta(\alpha_h+\alpha_s)} + \frac{U}{h+\eta(\alpha_h+\alpha_s)} \right) h \\
&\quad + U \frac{h+\alpha_h\eta}{h+\eta(\alpha_h+\alpha_s)} - \frac{h}{2\eta} \frac{\partial p}{\partial x} \frac{\alpha_s\eta(h+2\alpha_h\eta)}{h+\eta(\alpha_h+\alpha_s)}
\end{aligned}$$

$$\begin{aligned}
-hu = & -\frac{1}{2\eta} \frac{\partial p}{\partial x} h^3 + \left(\frac{h}{2\eta} \frac{\partial p}{\partial x} \frac{h+2\alpha_h\eta}{h+\eta(\alpha_h+\alpha_s)} + \frac{U}{h+\eta(\alpha_h+\alpha_s)} \right) h^2 \\
& -Uh \frac{h+\alpha_h\eta}{h+\eta(\alpha_h+\alpha_s)} + \frac{h^2}{2\eta} \frac{\partial p}{\partial x} \frac{\alpha_s\eta(h+2\alpha_h\eta)}{h+\eta(\alpha_h+\alpha_s)}
\end{aligned}$$

$-hu = c + d$, where

$$c = -\frac{h^3}{2\eta} \frac{\partial p}{\partial x} + \frac{h^3}{2\eta} \frac{\partial p}{\partial x} \frac{h+2\alpha_h\eta}{h+\eta(\alpha_h+\alpha_s)} + \frac{h^2}{2\eta} \frac{\partial p}{\partial x} \frac{\alpha_s\eta(h+2\alpha_h\eta)}{h+\eta(\alpha_h+\alpha_s)}$$

$$d = \frac{Uh^2}{h+\eta(\alpha_h+\alpha_s)} - Uh \frac{h+\alpha_h\eta}{h+\eta(\alpha_h+\alpha_s)}$$

➤ Determination of (c) value

$$\begin{aligned}
c &= \left(\frac{-h^3(h+\eta(\alpha_h+\alpha_s)) + (h^4+2h^3\alpha_h\eta) + (h^3\alpha_s\eta+2h^2\alpha_h\alpha_s\eta^2)}{2\eta(h+\eta(\alpha_h+\alpha_s))} \right) \frac{\partial p}{\partial x} \\
&= \left(\frac{-h^4 - h^3\alpha_h\eta - h^3\alpha_s\eta + h^4 + 2h^3\alpha_h\eta + h^3\alpha_s\eta + 2h^2\alpha_h\alpha_s\eta^2}{2\eta(h+\eta(\alpha_h+\alpha_s))} \right) \frac{\partial p}{\partial x} \\
&= \left(\frac{h^3\alpha_h\eta + 2h^2\alpha_h\alpha_s\eta^2}{2\eta(h+\eta(\alpha_h+\alpha_s))} \right) \frac{\partial p}{\partial x} \\
&= \frac{h^2}{2\eta} \frac{\partial p}{\partial x} \frac{h\alpha_h\eta + 2\alpha_h\alpha_s\eta^2}{h+\eta(\alpha_h+\alpha_s)}
\end{aligned}$$

➤ Determination of (d) value

$$\begin{aligned}
d &= \frac{Uh^2}{h+\eta(\alpha_h+\alpha_s)} - \frac{Uh^2}{h+\eta(\alpha_h+\alpha_s)} - \frac{Uh\alpha_h\eta}{h+\eta(\alpha_h+\alpha_s)} \\
&= -U \frac{h\alpha_h\eta}{h+\eta(\alpha_h+\alpha_s)}
\end{aligned}$$

$$-hu \frac{\partial h}{\partial x} = (c+d) \frac{\partial h}{\partial x}$$

$$-hu \frac{\partial h}{\partial x} = \frac{h^2}{2\eta} \frac{\partial p}{\partial x} \frac{\partial h}{\partial x} \frac{h\alpha_h\eta + 2\alpha_h\alpha_s\eta^2}{h + \eta(\alpha_h + \alpha_s)} - U \frac{h\alpha_h\eta}{h + \eta(\alpha_h + \alpha_s)} \frac{\partial h}{\partial x} \quad (3.8)$$

q_x becomes:

$$\begin{aligned} q_x = & -\frac{h^3}{12\eta} \frac{\partial p}{\partial x} \frac{h^2 + 4h\eta(\alpha_h + \alpha_s) + 12\eta^2\alpha_h\alpha_s}{h(h + \eta(\alpha_h + \alpha_s))} \\ & + \frac{U}{2} \frac{h^2 + 2h\alpha_h\eta}{2(h + \eta(\alpha_h + \alpha_s))} + \frac{h^2}{2\eta} \frac{\partial p}{\partial x} \frac{\partial h}{\partial x} \frac{h\alpha_h\eta + 2\alpha_h\alpha_s\eta^2}{h + \eta(\alpha_h + \alpha_s)} \\ & - U \frac{h\alpha_h\eta}{h + \eta(\alpha_h + \alpha_s)} \frac{\partial h}{\partial x} \end{aligned} \quad (3.9)$$

3.1.1. Theoretical modelling

3.1.1.1. Load support, no cavitation in recess (single pocket)

To find the resulting load support W , the flow rate equation should be obtained. The analysis below is based on continuity of flow through the bearing and uses the terminology as shown in Fig. 3.1. The condition in which the slip velocity, at the moving and stationary solid-liquid interface respectively, follows the Navier slip model.

FIGURE 3.1: Schematic of a parallel sliding textured bearing.

The corresponding flow rate equation as discussed by Tauviquirrahman et al. [32], if no-wedge effect $\left(\frac{\partial h}{\partial x} = 0\right)$ the Eq. (3.9) will reduce to:

$$q_x = -\frac{h^3}{12\eta} \frac{\partial p}{\partial x} \frac{h^2 + 4h\eta(\alpha_h + \alpha_s) + 12\eta^2\alpha_h\alpha_s}{h(h + \eta(\alpha_h + \alpha_s))} + \frac{U}{2} \frac{h^2 + 2h\alpha_h\eta}{2(h + \eta(\alpha_h + \alpha_s))} \quad (3.10)$$

If boundary slip is present only on the stationary surface ($\alpha_h \neq 0$), the flow rate per unit length will reduce to:

$$q_x = -\frac{h^3}{12\eta} \frac{\partial p}{\partial x} \frac{h^2 + 4h\eta\alpha_h}{h(h + \eta\alpha_h)} + \frac{U}{2} \frac{h^2 + 2h\eta\alpha_h}{2(h + \eta\alpha_h)} \quad (3.11)$$

For the areas of the bearing containing parallel surfaces, the film thickness h is constant and the pressure gradient must be linear (see Eq. (3.11)). Equation (3.11) can be modified for each region of the bearing to

$$q_{AB} = \frac{U}{2} \frac{h_o^2 + 2\eta h_o\alpha_h}{h_o + \eta\alpha_h} - \frac{h_o^3}{12\eta} \frac{h_o + 4\eta\alpha_h}{h_o + \eta\alpha_h} \left(\frac{p_2 - p_{atm}}{a} \right) \quad (3.12)$$

$$q_{BC} = \frac{U}{2} \frac{h_p^2 + 2\eta h_p\alpha_h}{h_p + \eta\alpha_h} - \frac{h_p^3}{12\eta} \frac{h_p + 4\eta\alpha_h}{h_p + \eta\alpha_h} \left(\frac{p_3 - p_2}{b} \right) \quad (3.13)$$

$$q_{CD} = \frac{U}{2} \frac{h_o^2 + 2\eta h_o\alpha_h}{h_o + \eta\alpha_h} - \frac{h_o^3}{12\eta} \frac{h_o + 4\eta\alpha_h}{h_o + \eta\alpha_h} \left(\frac{p_{atm} - p_3}{c} \right) \quad (3.14)$$

with one condition $q_{AB} = q_{BC} = q_{CD}$ based on volume conservation. When q_{AB} is set to be equal to q_{CD}

$$\begin{aligned} \frac{U}{2} \frac{h_o^2 + 2\eta h_o\alpha_h}{h_o + \eta\alpha_h} - \frac{h_o^3}{12\eta} \frac{h_o + 4\eta\alpha_h}{h_o + \eta\alpha_h} \left(\frac{p_2 - p_{atm}}{a} \right) = \\ \frac{U}{2} \frac{h_o^2 + 2\eta h_o\alpha_h}{h_o + \eta\alpha_h} - \frac{h_o^3}{12\eta} \frac{h_o + 4\eta\alpha_h}{h_o + \eta\alpha_h} \left(\frac{p_{atm} - p_3}{c} \right) \end{aligned}$$

$$\left(\frac{P_2 - P_{atm}}{a} \right) = \left(\frac{P_{atm} - P_3}{c} \right) \quad (3.15)$$

For the non-cavitating case, p_2 is unknown. Equating flow rate in inlet land and recess,

$$\begin{aligned} q_{AB} &= q_{BC} \\ \frac{U}{2} \frac{h_o^2 + 2\eta h_o \alpha_h}{h_o + \eta \alpha_h} - \frac{h_o^3}{12\eta} \frac{h_o + 4\eta \alpha_h}{h_o + \eta \alpha_h} \left(\frac{p_2 - p_{atm}}{a} \right) &= \\ \frac{U}{2} \frac{h_p^2 + 2\eta h_p \alpha_h}{h_p + \eta \alpha_h} - \frac{h_p^3}{12\eta} \frac{h_p + 4\eta \alpha_h}{h_p + \eta \alpha_h} \left(\frac{p_3 - p_2}{b} \right) & \end{aligned} \quad (3.16)$$

If, $K_o = \frac{\eta \alpha_h}{h_o + \eta \alpha_h}$ and $K_p = \frac{\eta \alpha_h}{h_p + \eta \alpha_h}$, then

$$\begin{aligned} \frac{U h_o}{2} (1 + K_o) - \frac{h_o^3}{12\eta} (1 + 3K_o) \left(\frac{p_2 - p_{atm}}{a} \right) &= \\ \frac{U h_p}{2} (1 + K_p) - \frac{h_p^3}{12\eta} (1 + 3K_p) \left(\frac{p_3 - p_2}{b} \right) & \end{aligned} \quad (3.17)$$

From Eq. (3.15), it is known that $p_3 = p_{atm} - \frac{c}{a} (p_2 - p_{atm})$ therefore

$$\left(\frac{p_3 - p_2}{b} \right) = \frac{ab + bc}{ab^2} (p_{atm} - p_2) \quad (3.18)$$

Substitution of Eq. (3.18) in Eq. (3.17) gives

$$\begin{aligned} p_2 \left[\left(h_p^3 + 3h_p^3 K_p \right) \frac{ab + bc}{ab^2} + \left(\frac{h_o^3 + 3h_o^3 K_o}{a} \right) \right] &= \\ p_{atm} \left[\left(h_p^3 + 3h_p^3 K_p \right) \frac{ab + bc}{ab^2} + \left(\frac{h_o^3 + 3h_o^3 K_o}{a} \right) \right] & \\ - 6\eta U \left[(h_p + h_p K_p) - (h_o + K_o) \right] & \end{aligned} \quad (3.19)$$

where,

$$C_p^* = h_p^3 + 3h_p^3 K_p; C_o^* = h_o^3 + 3h_o^3 K_o$$

$$C_p = h_p + h_p K_p; C_o = h_o + h_o K_o$$

Combining Eqs. (3.18) and (3.19) to eliminate p_3 gives

$$p_2 \left(\frac{ab+bc}{ab^2} C_p^* + \frac{C_o^*}{a} \right) = p_{atm} \left(\frac{ab+bc}{ab^2} C_p^* + \frac{C_o^*}{a} \right) - 6\eta U (C_p - C_o) \quad (3.20)$$

and thus,

$$p_3 = p_{atm} + \frac{6\eta U (C_p - C_o) bc}{aC_p^* + cC_p^* + bC_o^*} \quad (3.21)$$

Due to linearity between the pressure gradient in the inlet and outlet lands and in the recess, each of the three areas of the bearing has a triangular pressure distribution that can be integrated to determine the load support. Therefore, the total normal force reads:

$$W = W_{AB} + W_{BC} + W_{CD} - p_{atm} (a+b+c) \quad (3.22)$$

$$W = -\frac{6\eta U (C_p - C_o) ab}{aC_p^* + cC_p^* + bC_o^*} \left(\frac{a+b}{2} \right) + \frac{6\eta U (C_p - C_o) bc}{aC_p^* + cC_p^* + bC_o^*} \left(\frac{b+c}{2} \right) \quad (3.23)$$

Finally, the load support reads

$$W = \frac{6\eta U (C_p - C_o)}{C_p^*} b \left(\frac{c^2 - ab - a^2 + bc}{a+c+b \frac{C_o^*}{C_p^*}} \right) \quad (3.24)$$

3.1.1.2. Friction force, no cavitation in recess (single pocket)

The shear stress at the lower surface can be derived as follows:

$$\tau = -\frac{\eta U}{h + \eta \alpha_h} - \frac{h}{2} \frac{\partial p}{\partial x} \frac{h + 2\eta \alpha_h}{h + \eta \alpha_h} \quad (3.25)$$

For the no cavitation case, the friction force per unit width reads:

$$F = F_{AB} + F_{BC} + F_{CD}$$

$$F = -\eta U \left[\frac{a}{h_o + \eta \alpha_h} + \frac{c}{h_o + \eta \alpha_h} + \frac{b}{h_p + \eta \alpha_h} \right] - \eta U \left[\frac{3(C_p - C_o)^2 b(a+c)}{aC_p^* + cC_p^* + bC_o^*} \right] \quad (3.26)$$

3.1.1.3. Load support, cavitation in recess (single pocket)

For the case of cavitation in recess, p_2 is assumed to be equal to p_{cav} , so $q_{AB} = q_{BC}$ gives

$$\frac{Uh_o}{2}(1+K_o) - \frac{h_o^3}{12\eta}(1+3K_o) \left(\frac{p_{cav} - p_{atm}}{a} \right) = \frac{Uh_p}{2}(1+K_p) - \frac{h_p^3}{12\eta}(1+3K_p) \left(\frac{p_3 - p_{cav}}{X_b} \right) \quad (3.27)$$

where

$$X_b = \frac{C_p^*(p_3 - p_{cav})}{6\eta U(C_p - C_o) - C_o^* \left(\frac{p_{atm} - p_{cav}}{a} \right)} \quad (3.28)$$

It should be noted that for the cavitation zone, the pressure must be integrated separately, i.e. the non-cavitated area and cavitated area of the recess, so the total load support reads,

$$W = W_{AB} + W_{BB} + W_{BC} + W_{CD} - p_{atm}(a+b+c)$$

$$W = \left[p_{atm}a + \frac{a^2}{2a}(p_{cav} - p_{atm}) \right] + \left[p_{cav}b(1 - X_b) \right]$$

$$+ \left[p_{cav}X_b + \frac{6\eta U(X_b)^2}{2C_p^*}(C_p - C_o) + \frac{C_o^*(X_b)^2}{2aC_p^*}(p_{cav} - p_{atm}) \right]$$

$$+ \left[p_{atm}c - \frac{c^2}{2a}(p_{cav} - p_{atm}) \right] - p_{atm}(a+b+c) \quad (3.29)$$

3.1.1.4. Friction force, cavitation in recess (single pocket)

The friction force F over the surface for the case in which cavitation is considered in recess of single pocket, can be expressed as follows:

$$F = F_{AB} + F_{BB'} + F_{B'C} + F_{CD}$$

$$F = -\eta U \left[\frac{a}{h_o + \eta \alpha_h} + \frac{c}{h_o + \eta \alpha_h} + \frac{X_b}{h_p + \eta \alpha_h} \right] - \eta U \left[\left(\frac{C_p}{2} - \frac{C_o}{2} \right) \left(1 + \frac{c}{a} \right) (p_{atm} - p_{cav}) \right] \quad (3.30)$$

3.1.1.5. Slip in the first pocket, cavitation in recess (double pocket)

To obtain the load support of the parallel sliding double pocketed bearing considering cavitation for either slip or no slip conditions as reflected in Fig. 3.2, the flow rate equation is needed. The analysis below is based on continuity of flow through the bearing and uses the terminology as shown in Fig. 3.2. For the condition in which the slip velocity, at the stationary solid-liquid interface, follows the Navier slip model as discussed in Section 2.4 and by Alicia et al. [24], the corresponding flow rate equation can be expressed as Fowell et al., [9]:

$$q_x = -\frac{h^3}{12\eta} \frac{h^2 + 4\eta h(\alpha_h + \alpha_s) + 12\eta^2 \alpha_h \alpha_s}{h(h + \eta(\alpha_h + \alpha_s))} \frac{\partial p}{\partial x} + \frac{U}{2} \frac{h^2 + 2\eta h \alpha_h}{h + \eta(\alpha_h + \alpha_s)} + \frac{h^2}{2\eta} \frac{\partial p}{\partial x} \frac{\partial h}{\partial x} \frac{h\eta \alpha_h + 2\eta^2 \alpha_h \alpha_s}{h + \eta(\alpha_h + \alpha_s)} - U \frac{\partial h}{\partial x} \frac{h\eta \alpha_h}{h + \eta(\alpha_h + \alpha_s)} \quad (3.31)$$

If slip is present only on the stationary surface ($\alpha_s = 0$ but $\alpha_h \neq 0$) and no-wedge effect $\left(\frac{\partial h}{\partial x} = 0 \right)$, the flow rate will reduce to:

$$q_x = -\frac{h^3}{12\eta} \frac{\partial p}{\partial x} \frac{h^2 + 4\eta h \alpha_h}{h(h + \mu \alpha_h)} + \frac{U}{2} \frac{h^2 + 2\eta h \alpha_h}{(h + \eta \alpha_h)} \quad (3.32)$$

FIGURE 3.2: Schematic of a parallel sliding double pocketed bearing considering cavitation.

For the areas of the bearing containing parallel surfaces the film thickness h is constant, the pressure gradient must be linear (see Eq. (3.32)). Equation (3.32) can be modified for each region of the bearing:

$$q_{AB} = \frac{U}{2} h_o - \frac{h_o^3}{12\eta} \left(\frac{p_{cav} - p_{atm}}{a} \right) \quad (3.33)$$

$$q_{B'C} = \frac{U}{2} \frac{h_p^2 + 2\eta h_p \alpha_h}{h_p + \eta \alpha_h} - \frac{h_p^3}{12\eta} \frac{h_p + 4\eta \alpha_h}{h_p + \eta \alpha_h} \left(\frac{p_3 - p_{cav}}{X_b} \right) \quad (3.34)$$

$$q_{CD} = \frac{U}{2} h_o - \frac{h_o^3}{12\eta} \left(\frac{p_{atm} - p_3}{c} \right) \quad (3.35)$$

with the condition $q_{AB} = q_{B'C} = q_{CD}$ based on volume conservation.

When q_{AB} is set to be equal to q_{CD}

$$\frac{U}{2}h_o - \frac{h_o^3}{12\eta} \left(\frac{p_{cav} - p_{atm}}{a} \right) = \frac{U}{2}h_o - \frac{h_o^3}{12\eta} \left(\frac{p_{atm} - p_3}{c} \right) \quad (3.36)$$

$$p_3 = p_{atm} + \frac{c}{a}(p_{atm} - p_{cav}) \quad (3.37)$$

For the case of cavitation in the recess, p_2 is assumed to be equal to p_{cav} , so $q_{AB} = q_{B'C}$ gives

$$\begin{aligned} \frac{U}{2}h_o - \frac{h_o^3}{12\eta} \left(\frac{p_{cav} - p_{atm}}{a} \right) = \\ \frac{U}{2} \frac{h_p^2 + 2\eta h_p \alpha_h}{h_p + \eta \alpha_h} - \frac{h_p^3}{12\eta} \frac{h_p + 4\eta \alpha_h}{h_p + \eta \alpha_h} \left(\frac{p_3 - p_{cav}}{X_b} \right) \end{aligned} \quad (3.38)$$

If

$$\begin{aligned} K_p &= \frac{\eta \alpha_h}{h_p + \eta \alpha_h}; K_{p1} = \frac{\eta \alpha_h}{h_{p1} + \eta \alpha_h}; K_{p2} = \frac{\eta \alpha_h}{h_{p2} + \eta \alpha_h} \\ C_p &= h_p + h_p K_p; C_{p1} = h_{p1} + h_{p1} K_{p1}; C_{p2} = h_{p2} + h_{p2} K_{p2} \\ C_p^* &= h_p^3 + 3h_p^3 K_p; C_{p1}^* = h_{p1}^3 + 3h_{p1}^3 K_{p1}; C_{p2}^* = h_{p2}^3 + 3h_{p2}^3 K_{p2} \end{aligned}$$

Equation (3.38) becomes:

$$\frac{U}{2}h_o - \frac{h_o^3}{12\eta} \left(\frac{p_{cav} - p_{atm}}{a} \right) = \frac{U}{2}C_p - \frac{C_p^*}{12\eta} \left(\frac{p_3 - p_{cav}}{X_b} \right) \quad (3.39)$$

where

$$X_b = \frac{C_p^*(p_3 - p_{cav})}{6\eta U(C_p - h_o) - h_o^3 \left(\frac{p_{atm} - p_{cav}}{a} \right)} \quad (3.40)$$

3.1.1.6. Slip in the first pocket and second pocket, cavitation in recess (double pocket)

The flow rate equation q for the case of double pocket in which slip occurs in the first pocket and second pocket while cavitation exists in recess.

$$q_{AB} = \frac{U}{2} h_o - \frac{h_o^3}{12\eta} \left(\frac{p_{cav} - p_{atm}}{a} \right) \quad (3.41)$$

$$q_{B'C} = \frac{U}{2} C_{p1} - \frac{C_{p1}^*}{12\eta} \left(\frac{p_3 - p_{cav}}{X_b} \right) \quad (3.42)$$

$$q_{CD} = \frac{U}{2} h_o - \frac{h_o^3}{12\eta} \left(\frac{p_{cav} - p_3}{c} \right) \quad (3.43)$$

$$q_{D'E} = \frac{U}{2} C_{p2} - \frac{C_{p2}^*}{12\eta} \left(\frac{p_5 - p_{cav}}{X_d} \right) \quad (3.44)$$

$$q_{EF} = \frac{U}{2} h_o - \frac{h_o^3}{12\eta} \left(\frac{p_{atm} - p_5}{e} \right) \quad (3.45)$$

with the condition $q_{AB} = q_{B'C} = q_{CD} = q_{D'E} = q_{EF}$ based on volume conservation.

When q_{AB} is set to be equal to $q_{B'C}$

$$\left(\frac{p_3 - p_{cav}}{X_b} \right) = \frac{6\mu U}{C_{p1}^*} (C_{p1} - h_o) - \frac{h_o^3}{C_{p1}^*} \left(\frac{p_{cav} - p_{atm}}{a} \right) \quad (3.46)$$

When $q_{B'C}$ is set to be equal to q_{CD}

$$\left(\frac{p_3 - p_{cav}}{X_b} \right) = \frac{6\eta U}{C_{p1}^*} (C_{p1} - h_o) - \frac{h_o^3}{C_{p1}^*} \left(\frac{p_{cav} - p_3}{c} \right) \quad (3.47)$$

Substitution of Eq. (3.46) in Eq. (3.47) gives

$$p_3 = p_{cav} + \frac{c}{a} (p_{atm} - p_{cav}) \quad (3.48)$$

$$X_b = \frac{c C_{p1}^* (p_{atm} - p_{cav})}{6\eta U a (C_{p1} - h_o) - h_o^3 (p_{atm} - p_{cav})} \quad (3.49)$$

When q_{CD} is set to be equal to $q_{D'E}$

$$\left(\frac{p_5 - p_{cav}}{X_d} \right) = \frac{6\eta U}{C_{p2}^*} (C_{p2} - h_o) - \frac{h_o^3}{C_{p2}^*} \left(\frac{p_{cav} - p_3}{c} \right) \quad (3.50)$$

When $q_{D'E}$ is set to be equal to q_{EF}

$$\left(\frac{p_5 - p_{cav}}{X_d} \right) = \frac{6\eta U}{C_{p2}^*} (C_{p2} - h_o) - \frac{h_o^3}{C_{p2}^*} \left(\frac{p_{atm} - p_5}{e} \right) \quad (3.51)$$

Substitution of Eq. (3.50) in Eq. (3.51) gives

$$p_5 = p_{atm} + \frac{e}{a} (p_{atm} - p_{cav}) \quad (3.52)$$

$$X_d = \frac{(a + e) C_{p2}^* (p_{atm} - p_{cav})}{6\eta U a (C_{p2} - h_o) - h_o^3 (p_{atm} - p_{cav})} \quad (3.53)$$

3.1.2. Numerical approach

In addition to the theoretical approach, a numerical approach with respect to the Reynolds equation is also proposed. In this way, a two-dimensional lubrication model can be obtained so that the physical phenomena in the x - and y -direction can be made. The modified Reynolds equation (Eq. (3.54)) is derived for an incompressible lubricant with constant viscosity from a simple form of the Navier Stokes equation, ignoring the inertia in the film.

$$\begin{aligned} \frac{\partial p}{\partial x} &= \frac{\partial}{\partial z} \left(\eta \frac{\partial u}{\partial z} \right) \\ \frac{\partial p}{\partial y} &= \frac{\partial}{\partial z} \left(\eta \frac{\partial v}{\partial z} \right) \\ \frac{\partial p}{\partial z} &= 0 \end{aligned} \quad (3.54)$$

In order to obtain the velocity distribution by integration of Eq. (3.54), it is necessary to define the surface boundary conditions. The surface boundary conditions are proposed as:

$$\begin{aligned}
&\text{at } z=0, u=U \quad ; v=0 \\
&\text{at } z=h, u=-\alpha_h \eta \frac{\partial u}{\partial z} \bigg|_{z=h} \quad ; v=-\alpha_h \eta \frac{\partial v}{\partial z} \bigg|_{z=h}
\end{aligned} \tag{3.55}$$

The expression of the fluid velocity in the x-direction, subject to the above boundary conditions is:

$$u = \frac{1}{2\eta} \frac{\partial p}{\partial x} z^2 - \left(\frac{U}{h + \eta\alpha_h} + \frac{h}{2\eta} \frac{\partial p}{\partial x} \frac{h + 2\eta\alpha_h}{h + \eta\alpha_h} \right) z + U \frac{h + \eta\alpha_h}{h + \eta\alpha_h} \tag{3.56}$$

A similar expression can be written for the fluid velocity in the y-direction. Assuming the fluid density is constant, the conservation of mass requires

$$\int_0^h \frac{\partial u}{\partial x} dz + \int_0^h \frac{\partial v}{\partial y} dz + \int_0^h \frac{\partial w}{\partial z} dz = 0 \tag{3.57}$$

Deriving each term of the above equation (Eq. (3.57)), the modified Reynolds equation obtained is:

$$\begin{aligned}
&\frac{\partial}{\partial x} \left(\frac{h^3}{12\eta} \frac{\partial p}{\partial x} \frac{h^2 + 4h\eta\alpha_h}{h(h + \eta\alpha_h)} \right) + \frac{\partial}{\partial y} \left(\frac{h^3}{12\eta} \frac{\partial p}{\partial y} \frac{h^2 + 4h\eta\alpha_h}{h(h + \eta\alpha_h)} \right) = \frac{U}{2} \frac{\partial}{\partial x} \left(\frac{h^2 + 2h\eta\alpha_h}{h + \eta\alpha_h} \right) \\
&- U \frac{\alpha_h \eta}{h + \eta\alpha_h} \frac{\partial h}{\partial x} + \frac{h}{2\eta} \frac{h\eta\alpha_h}{h + \eta\alpha_h} \left(\frac{\partial p}{\partial x} \frac{\partial h}{\partial x} + \frac{\partial p}{\partial y} \frac{\partial h}{\partial y} \right)
\end{aligned} \tag{3.58}$$

Equation (3.58) can be applied to the pressure generation and cavitation zones. Thus, Eq. (3.58) is modified as follows:

$$\begin{aligned}
&\left\{ \frac{\partial}{\partial x} \left(\frac{h^3}{12\eta} \frac{\partial D}{\partial x} \frac{h^2 + 4h\eta\alpha_h}{h(h + \eta\alpha_h)} \right) + \frac{\partial}{\partial y} \left(\frac{h^3}{12\eta} \frac{\partial D}{\partial y} \frac{h^2 + 4h\eta\alpha_h}{h(h + \eta\alpha_h)} \right) \right\} F' = \\
&\frac{U}{2} \frac{\partial}{\partial x} \left(\frac{h^2 + 2h\eta\alpha_h}{h + \eta\alpha_h} \right) - U \frac{\alpha_h \eta}{h + \eta\alpha_h} \frac{\partial h}{\partial x} + \frac{h}{2\eta} \frac{h\eta\alpha_h}{h + \eta\alpha_h} \left(\frac{\partial D}{\partial x} \frac{\partial h}{\partial x} + \frac{\partial D}{\partial y} \frac{\partial h}{\partial y} \right) F' \\
&+ \left\{ \frac{U}{2} \frac{\partial}{\partial x} \left(\frac{D}{h} \frac{h^2 + 2h\eta\alpha_h}{h + \eta\alpha_h} \right) - \frac{D}{h} U \frac{\eta\alpha_h}{h + \eta\alpha_h} \frac{\partial h}{\partial x} \right\} (1 - F')
\end{aligned} \tag{3.59}$$

In the active zones:

$$\begin{cases} D = p, \\ F' = 1 \end{cases} \quad D \geq 0$$

$$\begin{aligned} \frac{\partial}{\partial x} \left(\frac{h^3}{12\eta} \frac{\partial p}{\partial x} \frac{h^2 + 4h\eta\alpha_h}{h(h+\eta\alpha_h)} \right) + \frac{\partial}{\partial y} \left(\frac{h^3}{12\eta} \frac{\partial p}{\partial y} \frac{h^2 + 4h\eta\alpha_h}{h(h+\eta\alpha_h)} \right) &= \frac{U}{2} \frac{\partial}{\partial x} \left(\frac{h^2 + 2h\eta\alpha_h}{h + \eta\alpha_h} \right) \\ -U \frac{\eta\alpha_h}{h + \eta\alpha_h} \frac{\partial h}{\partial x} + \frac{h}{2\eta} \frac{h\eta\alpha_h}{h + \eta\alpha_h} \left(\frac{\partial p}{\partial x} \frac{\partial h}{\partial x} + \frac{\partial p}{\partial y} \frac{\partial h}{\partial y} \right) & \end{aligned} \quad (3.60)$$

In the cavitation zones:

$$\begin{aligned} \frac{U}{2} \frac{\partial}{\partial x} \left(\frac{h^2 + 2h\eta\alpha_h}{h + \eta(\alpha_h + \alpha_s)} \right) - U \frac{\eta\alpha_h}{h + \eta\alpha_h} \frac{\partial h}{\partial x} + \frac{U}{2} \frac{\partial}{\partial x} \left(\frac{r-h}{h} \frac{h^2 + 2h\eta\alpha_h}{h + \eta\alpha_h} \right) \\ - \frac{r-h}{h} U \frac{\eta\alpha_h}{h + \eta(\alpha_h + \alpha_s)} \frac{\partial h}{\partial x} = 0 \end{aligned} \quad (3.61)$$

where: $D = r - h = h(\bar{\rho} - 1)$; $r = \bar{\rho}h$; $\bar{\rho} = \frac{\rho_o}{\rho}$, ρ_o is the density of the lubricant-gas mixture and ρ is the density of the lubricant.

$$\frac{U}{2} \frac{\partial}{\partial x} \left(\bar{\rho} \frac{h^2 + 2h\eta\alpha_h}{h + \eta(\alpha_h + \alpha_s)} \right) - U \frac{\bar{\rho}\eta\alpha_h}{h + \eta(\alpha_h + \alpha_s)} \frac{\partial h}{\partial x} = 0 \quad (3.62)$$

In order to simultaneously solve Eqs. (3.59) - (3.62) over the whole domain of the bearing, a universal variable D is defined and a unified modified Reynolds equation is obtained as stated in Eq. (3.59). As a note, the universal function D in Eq. (3.59) for the no cavitation case (i.e. active zone) refers to pressure or p . However, for the cavitation zone, the D in Eq. (3.59) refers to the function of the density (lubricant and gas/vapour). So, in the cavitation zone, the vapour density can be calculated from Eq. (3.62) above. Then, based on the continuity equation, the pressure in the cavitation region can be obtained. It should be highlighted that in Eq. (3.61), the value D (in which $D = r - h$) in the cavitation region is always smaller than 1.

3.2. Lubrication performance

Factors such as traction, lubrication performance (i.e. load support and friction), and service life are determined when designing any type of lubricated sliding contacts. The performance of a hydrodynamically lubricated contact is certainly affected by the operating conditions, the thermal properties of the surface material, the choice of the lubricant, and the surface topography. However, as previously mentioned, the artificial chemical surface treatment (i.e. boundary slip) and physical roughness (i.e. texturing) are of particular interest here.

Load support per unit length, W , is obtained by integration of pressure and is the total load the contact can support for a given film thickness distribution. It reads:

$$W = \int_0^l p dx \quad (3.63)$$

The friction force generated by the lubrication system is due to the fluid viscous forces. It is calculated by integrating interface shear stress over the interface surface area. Friction force per unit length, F , is given by:

$$F = \int_0^l \tau dx \quad (3.64)$$

with the shear stress at the bottom surface being:

$$\tau = \eta \left(\frac{\partial u}{\partial z} \right)_{z=0} \quad (3.65)$$

3.3. Slip texture consideration

In general, surface texturing as a predefined roughness was introduced as a surface engineering technique to reduce friction. As previously mentioned, with respect to the reduced friction, the potentially useful implications of boundary slip were also suggested by recent researchers. Knowledge about slip parameters, as well as surface texturing parameters, is essential for achieving this in, for instance, lubricated bearings.

3.3.1. Slip parameters

One of the developed treatments to eliminate stiction is the development of new materials or design of surfaces and interfaces with hydrophobic behaviour [12-13]. Non-wetting (hydrophobicity) is an important surface behaviour for materials or devices in micro-applications. The hydrophobicity of a surface is generally presented in terms of slippage length, which quantifies the extent to which the fluid elements near the surface are affected by corrugation of the surface energy [71].

3.3.1.1. Slip length

With the application of sliding surfaces in very narrow-gap conditions and the emergence of hydrophobic materials, the classical no-slip boundary condition can break down. When a lubricant slips along a solid-liquid interface, the slip length, b , is generally used to address the relation between the slip velocity and the surface shear rate, which is

$$u_s = b \left. \frac{\partial u}{\partial z} \right|_{\text{surface}} \quad (3.66)$$

where u_s indicates the streamwise slip velocity at the hydrophobic surface, b denotes the slip length, and $\partial u / \partial z|_{\text{surface}}$ is the surface shear rate. It is usually postulated that a large value of b implies larger slippage. Furthermore, it is also conventionally implied that the large slip is also associated with a large friction force reduction. Numerous works have demonstrated that the chemical treatment of the surface generates a slip length in the order of $1 \mu\text{m}$ [96], whereas a longer slip length of up to $100 \mu\text{m}$ can be obtained through a combination of a textured pattern with a hydrophobic surface [18, 97-98]. In the present study, the slip length of a hydrophobic surface is assumed as uniform in space due to the application on microscopic scale.

3.3.1.2. Critical shear stress

As mentioned earlier, the better the surface hydrophobicity, the lower the critical shear stress. In this thesis, the effect of artificial surface roughness (i.e. texturing) on the critical shear stress is not investigated. This is because the contradiction between the roughness effects in inhibiting slip as observed by Granick, et al. [78] and the roughness effects in promoting slip as observed by Bonaccorso, et al. [79] has been discussed in [33].

3.3.2. Surface texture

The use of artificial surface texturing is becoming popular in oil-lubricated devices because of its potential benefits in terms of load support and friction force, both experimentally and theoretically. It was shown experimentally that such texturing enhances the load support and reduces the hydrodynamic friction force in, for instance, systems with two parallel sliding surfaces [1, 99-100] and reciprocating (cylinder-liner) contacts [2]. The shapes of the surface feature were usually dimples or grooves; for grooves see Figs. 3.1. and 3.2 and for dimples see Fig. 3.3. In the hydrodynamically lubricated sliding contact in particular, artificial surface texturing supports the formation of a lubricating oil film.

FIGURE 3.3: *Partially textured parallel thrust pad bearing with rectangular dimples [101].*

Full and partial texturing are two extreme cases of artificial arrangement of textured areas on the contact surface. It is worth mentioning the early work of Tonder [102] who analysed the partial texturing mode by theoretical studies on the positive effect of a series of dimples or roughness at the inlet of a sliding surface. A comparison of partially and fully textured surfaces comprising micro-roughness in parallel thrust bearings was given by Brizmer, et al. [103] through employing a numerical approach. They demonstrated that full width texturing is not useful for developing the large load support as expected from a hydrodynamically lubricated contact. Subsequently, several studies were published in various literature [57, 104-106] confirming the work of Brizmer, et al. [103]. One of the conclusions that emerges from these studies is that partial texturing has an advantage over full texturing.

In addition, more attention has been paid to the optimization of texturing parameters. Various surface texture models (elliptical model, sinusoidal model, rectangular model, cylindrical model, etc.) were employed to conclude that surface texturing improves the tribological performance. Efforts were made in several works to establish the optimal texturing parameters such as texture depth, width, number of textures, and location of textures that would maximize the load support. The previously-mentioned models and simulation results also provide an excellent set of guidelines for the optimum design of a surface texture in some fields.

Recently, in addition to surface texturing, the use of an artificial slip surface is also of great interest with respect to lubrication. Such a surface was introduced deliberately on the lubricated sliding contact with the help of microfabrication techniques. Thus, the artificial slip surface and the surface texturing are closely related to the manufacturing process so as to gain the expected performance. In general, the choice of the boundary condition, i.e. slip or no-slip situation as well as the cavitation, will play a very important role in determining the fluid flow behaviour, and thus must be considered.

3.4. Summary

A more detailed modified Reynolds equation for lubrication analysis has been developed, considering cavitation and slip. The developed model is based on a first-order Reynolds approach, which is an extension of Fowell's research [9].

Chapter 4

Results

4.1. Significance of the cavitation phenomena

Recently, tribologists have faced a great challenge to improve lubricated bearings with the aim of minimizing energy use. The use of artificial surface texturing is becoming popular on oil lubricated contacts such as parallel sliding contacts, because of its potential benefits in terms of tribological performance. The case of parallel sliding contacts with single texture has been studied by Fowell et al. [9] and it was found that cavitation occurs due to the presence of surface texturing. Thus, it is important to analyse the phenomenon of cavitation in the case of surface texturing.

In the present work, an analytical equation based on first-order Reynolds theory is derived. The Reynolds equation is extended by considering slip and cavitation. Here, slip is engineered to occur on the whole stationary surface, thus also on the pocket surface (i.e. ‘full slip’ configuration) as shown in Figure 3.1. The methodology used in the derivation of the extended Reynolds equation at the bearing surface is quite similar to that employed by Fowell [9] and the effect of the boundary slip on the hydrodynamic performance (load support and friction) has been analysed.

Figure 4.1 reflects the correlation between the inlet land width a/B_o and pocket depth h_d . It can be observed that the pocket depth has a strong influence on the inlet length as the initiator for cavitation in both the slip and no-slip cases. The minimum values are noted as the critical points of pocket depth h_d for both cases. This is to say that below these pocket depth values, the lubricant in the inlet starts to reduce the amount of flow in the pocket. It is worth noting that the critical value of h_d for the slip case is higher than for the no-slip case (see inset for details). Obviously, as can also be seen from Fig. 4.1, the predicted values of a/B_o for the slip case are higher than for the no-slip case for all values of h_d considered. From a physical point of view, it indicates that adding slip to the pocket bearing enables a reduction in cavitation. For example, for the no-slip textured bearing of interest with $h_d = 4.0 \times 10^{-6}$ m and an inlet length a/B_o 0.01, lubrication will generate the cavitation phenomena. However, if slip is applied on such a bearing, cavitation can be prevented.

Regarding the slip and texture condition in parallel sliding with cavitation, analytical simulations have been conducted to investigate the influence of slip on lubrication performance. It was concluded that cavitation can be reduced significantly with slip (see Fig. 4.3, as well as appended Papers A, B, D and E).

FIGURE 4.1: Cavitation as a function of inlet land width a/B_o and pocket depth h_d . The profiles are calculated for $h_o = 1.0 \times 10^{-6} \text{ m}$, $a = 0.004 \text{ m}$, $b = 0.006 \text{ m}$, $B_o = 0.02 \text{ m}$, $\alpha = 0.02 \text{ m}^2\text{s/kg}$, and $p_{cav} = 0 \text{ kPa}$. The inset shows the predicted a/B_o for low values of h_d (i.e. up to $2.0 \times 10^{-6} \text{ m}$).

4.2. Combined effects of slip and cavitation

As is known, the artificial full slip surface gives a very beneficial effect with respect to friction reduction. However, if the performance is related to load support, full slip is not recommended. In relation to lubricated contacts, the reduction in pressure generation and resulting lower load support is generally unwanted because it can lead to the failure of the system [36]. For this reason, an analysis of load support as a function of pocket depth is made.

Figure 4.2 shows that in relation to the effect of the pocket depth on load support, for both the full slip and no-slip surfaces, the deeper the pocket, the more difficult it is for cavitation to take place. As explained by Fowell et al. [9], for a traditional (no-slip) textured bearing this is due to the restriction of the flow, which is greater for deeper pockets than for shallow ones. It seems that this condition also prevails with respect to the slip pocket. Figure 4.2 correlates with Figure 4.1, for small values of a/B_o (smaller than 0.01) and

pocket depths greater than $2.5 \mu\text{m}$, there is no cavitation condition (see Fig. 4.1). If the phenomenon of cavitation does not occur, full slip delivery is not recommended because it will reduce load support. It is, however, recommended for the cavitation region. From a physical point of view, this new finding is interesting. As can be observed in Fig. 4.3 (for example, in the case of a pocketed bearing with $h_d / h_o = 0.71$), adding boundary slip will generate a positive effect, i.e. more hydrodynamic pressure than without boundary slip. The explanation for this effect could be that introducing boundary slip over the whole surface could retard the level of cavitation. On the other hand, slip gives rise to the inlet suction mechanism.

FIGURE 4.2: The effect of inlet length on the load support. The profiles are calculated for $h_o = 1.0 \times 10^{-6} \text{ m}$, $b = 0.006 \text{ m}$, $B_o = 0.02 \text{ m}$, $\alpha = 0.02 \text{ m}^2/\text{s/kg}$.

FIGURE 4.3: The pressure distribution as a function of coordinate in sliding direction for a pocketed bearing of interest with $h_d = 5.0 \times 10^{-6}$ m, $h_o = 7.0 \times 10^{-6}$ m, $a = 0.004$ m, $b = 0.006$ m, $B_o = 0.02$ m, and $\alpha = 0.02$ m²/s/kg. Cavitation is present.

Figure 4.4 illustrates the load support which is plotted against the ratio b/B_o . As expected, for low h_d ($h_d = 2.5 \times 10^{-6}$ m in this case), the load support prediction between slip and no-slip textured bearings is very close. As b/B_o increases, the load support decreases. For high h_d , the difference in load support between the slip and no-slip situation is not as small as is the case for low h_d , especially for b/B_o which is lower than, say, 0.1. Fig. 4.4 also reflects the fact that the load support for a slip textured bearing is, for all h_d values considered here, greater than for a no-slip textured bearing. With respect to the utilization of slip (i.e. $\alpha = 0.02$ m²/s/kg) on a textured bearing, the load support can be improved from 1.2 to 11.8 % (at $h_d = 2.5 \times 10^{-6}$ m) and 5.4 to 38.4 % (at $h_d = 7.5 \times 10^{-6}$ m) depending on b/B_o . This means that the slip effect has a significant contribution on the pressure distribution, and thus on load support, especially when the pocket depth is great. In relation to the texture parameter (i.e. pocket depth), as shown in Fig. 4.4, for h_d considered here, it can be observed that the greater the pocket depth, the greater the predicted load support, which matches well with the previous results. This is the case for both the slip and no-slip textured bearing.

FIGURE 4.4: The effect of pocket length b/B_o on the load support W . The profiles are calculated for $h_o = 1.0 \times 10^{-6}$ m, $a = 0.004$ m, $B_o = 0.02$ m, and $\alpha = 0.02$ m²/s/kg.

4.2.1. Slip in the pocket area

Full slip on the textured slider bearings can reduce the cavitation area (see Fig 4.3). However, slip also reduces the area of cavitation if present only in the pocket area. Paper D explores slip in the pocket area on slider bearings with single and double pockets. It is found that slippage in the pocket areas also reduces the cavitation area, although it is not significant, and load support increases as a result.

Figure 4.5 depicts the comparison of the pressure distribution of a single-pocketed bearing between the slip case and the no-slip case varying the pocket depth. The slip case is limited to the pocket of the bearing in which slip is allowed at the top edge of the pocket (Fig. 4.12 (b)) by setting the slip coefficient $\alpha = 0.02$ m²/s/kg. It should be noted that the chosen slip coefficient is an optimal value with respect to the maximum load support based on the published works, both experimentally [17-18] and numerically [28].

Several specific features can be observed based on Figure 4.5. Firstly, for single-pocketed bearings with both slip and no-slip, the pressure distribution shows a similar trend for all pocket depth values studied. At the leading edge of the contact, the pressure decreases to the cavitation pressure for all configurations. This indicates that if the pocket is placed not at the inlet length, the suction mechanism, as discussed by Bayada et al. [28], will dominate the bearing even if boundary slip is employed. Secondly, it is easy to observe that cavitation occurs in the bearing for both cases (slip and no-slip). This is characterized by the presence of a cavitation area as marked in Figs 4.5 (a) and 4.5 (b). For the inset of

Fig. 4.5 (a) regarding the slip case, it can be observed that a great pocket depth HD leads to a decrease in the cavitation area. However, different with the slip case, for the no-slip case, a low HD behaves otherwise.

FIGURE 4.5: The pressure distribution as a function of coordinate in sliding direction for the pocketed bearing for $h_o = 1 \times 10^{-6}$ m, $a = 0.002$ m, $b = 0.0015$ m, $e = 0.0165$ m, $B_o = 0.02$ m, and $\alpha = 0.02$ m²/s/kg.

Figure 4.6 shows the pressure distribution of the double pocket bearing for the slip and no-slip conditions. The figure shows some findings: firstly, the pressure distribution for the slip or no-slip condition has a similar trend. The maximum pressure P_5 occurs in the second pocket, while the value of the maximum pressure is the same for the slip or no-slip condition ($p_5^* = 0.0034$). The maximum pressure in the first pocket is equal to the atmospheric pressure. Secondly, the pressure distribution in the first or second pocket is not influenced by the first pocket depth HD_1 . The variation in the first pocket depth HD_1 will influence the length of the cavitation area in the first pocket (see insets of Figs. 4.6 (a) and 4.6 (b)). However, it does not affect the cavitation area in the second pocket. Consequently, the length of the cavitation area is not a function of the first pocket depth HD_1 . Thirdly, the slip condition will affect the cavitation area in the first or second pocket, in other words it reduces the cavitation area.

(a) slip condition.

(b) no-slip condition.

FIGURE 4.6: The pressure distribution as a function of coordinate in sliding direction x^* for the double pocketed bearing for $h_{d2} = 1 \times 10^{-6}$ m, $h_o = 1 \times 10^{-6}$ m, $a = 0.002$ m, $b = 0.0015$ m, $c = 0.002$ m, $d = 0.003$ m, $e = 0.0115$ m, $B_o = 0.02$ m, and $\alpha = 0.02$ m²/s/kg.

4.2.2. Slip in the inlet area of bearing

As discussed in the previous section, full slip in the pocket can reduce the cavitation zone. In this section, the effect of slip used at the inlet and at inlet (dam) between the pockets for a parallel slider bearing with double pockets are investigated.

Figure 4.7 shows the effect of dimensionless inlet length a^* on the dimensionless pressure distribution on parallel sliding double pocket bearings. For the no-slip condition, the variation of dimensionless inlet lengths does not have an effect on the dimensionless maximum pressure distribution on the slider pocket bearing. The maximum dimensionless pressure distribution in the first pocket p_3^* is 0.0005 and for the second pocket p_5^* is 0.0038, the value closest of all to inlet lengths a^* . Increasing the dimensionless inlet length a^* leads to a negative effect on load support, a decrease in W is found. This is due to decrease of out length land e (see Fig. 4.7 (a)). Cavitation occurs in the pockets and the cavitation increase in the pockets (see inset Fig. 4.7 (a)). However, opposite to the previous case, for the condition in which slip is present in the inlet length, no cavitation take place. The effect of slip in the inlet increases the pressure distribution. A pressure drop still occurs when the fluid enters the second pocket. The pressure drop does not occur when slip is present between the pockets as well (see Fig. 4.7 (c)). So, boundary slip in the inlet and the area before the second pocket can eliminate cavitation. In the case of boundary slip, the extension of the inlet length has a positive effect of increasing the maximum pressure and therefore on load support. The maximum load support is 0.0417 for an inlet length a^* of 0.2 for the condition in which boundary slip is present in the inlet and between pockets.

From Fig. 4.7 (b), slip in the inlet area increases the pressure in the pocket area (so that cavitation does not occur). Although in the next pocket there is a decrease in pressure, however no cavitation occurs (see Fig. 4.7 (b)). It can be concluded that one wants to increase load support or eliminate cavitation, slip is should be present in the inlet and space between the pockets (see Fig. 4.7 (c)).

FIGURE 4.7: The effect of non-dimensional inlet length a^* on the dimensionless pressure p^* . (Note: $h_o = 1.0 \times 10^{-6} \text{ m}$, $h_d = 1.0 \times 10^{-6} \text{ m}$, $b = 0.0015 \text{ m}$, $c = 0.002 \text{ m}$, $d = 0.0015 \text{ m}$, $B_o = 0.02 \text{ m}$).

4.2.3. Slip in the full area of bearing

The effect of boundary slip at the whole bearing surface on the tribological behaviour is discussed in this section. Variations in suction inlet, pocket length and pocket depth are studied. The results of the study show that slip on all bearing surfaces will reduce the cavitation area.

Figure 4.8 shows the relationship between the non-dimensional inlet length and dimensionless pressure. It can be observed that a smaller non-dimensional inlet length, results in a larger pressure. It means that an increased load support is noted with smaller inlet length a^* . When a^* is 0.025, cavitation is reduced significantly for both the slip and no-slip situation. It is also seen that by giving full slip, there is a slight increase in the p_2^* value. However, the peak value of p_3^* is smaller for the no-slip situation. It is also noted that if $a^* = 0.025$ for the no-slip condition, cavitation still occurs, however the length of the cavitation area is very small. It indicates that for small a^* , full slip does not have much of an effect on pressure. This is also the case for high a^* ($a^* = 0.1$ here). Full slip tends to reduce the cavitation area slightly. In general, by shortening the inlet length, it could significantly increase the load support. As discussed by Fowell et al. [9], the cavitation pressure has a strong effect on producing inlet suction as a mechanism for pressure generation. In this work, such a mechanism is also found for both the slip and no-slip conditions. The positive effect is increased when full slip is introduced.

FIGURE 4.8: The effect of non-dimensional inlet length a^* on the dimensionless pressure p^* .

Figure 4.9 compares the no-slip condition with the full-slip condition for variations of pocket lengths b^* . From this figure, it can be seen that if b^* is 0.15, the area below the pressure distribution for the full slip condition, and thus the load support, is slightly larger than for the no-slip situation. However, when the value b^* is set to 0.05, the area below the pressure distribution for full slip is much smaller than that for the no-slip contact. This is because in the full-slip condition, in addition to the absence of a cavitation area at the beginning of the pocket, the peak value of p_3^* is smaller than in the no-slip situation. On the other hand, for the full slip-slip condition, the increase in p_2^* is not very different from the decrease in p_3^* . In general, shortening the pocket length can reduce the cavitation area and a greater decrease in the cavitation effect is highlighted when the applied full slip decreases.

FIGURE 4.9: The effect of non-dimensional pocket length b^* on the dimensionless pressure p^* .

Figure 4.10 shows the effect of the non-dimensional texture depth HD on the dimensionless pressure. It can be observed that for $HD = 5$, cavitation will occur in the pocket whether slip is applied or not. However, as depicted in Figure 4.10, for the full slip case, the length of X_b is greater than for the no-slip case. From a physical point of view, it means that the application of slip tends to prevent the occurrence of cavitation. For a greater pocket depth, for example $HD = 10$, cavitation will occur for the no-slip pocketed bearing, whereas for the full slip bearing, cavitation does not exist as expected. Before cavitation occurs, the value of p_3^* is the same for both the slip and no-slip condition. When cavitation occurs, p_3^* tends to decrease, whereas p_2^* increases.

FIGURE 4.10: *The effect of non-dimensional texture depth HD on the dimensionless pressure p^* .*

Figure 4.11 depicts the effect of the non-dimensional slip length A on the dimensionless pressure p^* . From this figure, it can be seen that introducing slip on the pocketed parallel slider bearing leads to shortening of the cavitation area. The result strengthens the previous result that states that slip has a positive effect on reducing the cavitation effect. Another interesting result is that the value of p_3^* (the peak of the pressure distribution) is the same for both the slip and no-slip cases. However, the bearing with slip condition has a greater X_b value, which indicates that the cavitation area can be reduced by slip. Regarding the variations of the dimension slip lengths (50 and 200 in this case), the predicted dimensionless pressure profiles have a similar trend. It indicates that the slip length considered here refers to the optimal value and consequently the optimal performance with respect to the pressure, i.e. the load support.

FIGURE 4.11: *The effect of the non-dimensional slip length value A on the dimensionless pressure p^* .*

4.3. Single texturing (theoretical approach)

The textured bearing configuration is shown in Fig. 4.12.

FIGURE 4.12: Schematic diagram of parallel sliding single pocketed bearings considering cavitation.

Here, boundary slip is engineered to occur at the stationary surface, as shown in Figs. 4.12 (b) and 4.12 (c). The characteristics of the single pocket bearing analysed can be seen in Table 4.1.

TABLE 4.1: *Characteristics of the single pocket bearing analysed.*

Parameters	Symbols	Value
Bearing length	B_o	0.02 m
Inlet length	a	0.004 m
Pocket length	b	0.006 m
Exit land length	c	0.01 m
Atmospheric pressure	p_{atm}	100 kPa
Cavitation pressure	p_{cav}	0-100 kPa
Sliding velocity	u	1 m/s
Slip coefficient	α	0.02 m ² s/kg

In detail, with respect to the effect of the pocket depth, it can be observed that in the case of a pocketed bearing with slip within the pocket, the pocket depth will affect the cavitation area C_{Lb} . This trend also prevails for the no-slip case. Fig. 4.13 reflects the effect of the pocket depth on the cavitation area for both slip at the pocket and the no-slip pocketed case. For the slip at the pocket case, increasing the pocket depth will decrease the C_{Lb} considerably. It is very interesting to note that cavitation exists when $HD = 0$, which is associated with the smooth parallel sliding surface. This result matches well with published work [28]. As indicated in [28], as the slip zone is displaced from the inlet to the middle of the bearing, which is similar to the case of “slip at pocket” (Fig. 4.12 (b)), the pressure field calculated collapses. It means that the pressure collapse leads to cavitation.

From a design and manufacturing point of view, having no-slip in the inlet is not recommended because it leads to cavitation. The cavitation may reduce the lubrication performance of a sliding contact for the no-slip case of Fig. 4.12 (a). The increase in HD will produce an increase-then-decrease profile of the cavitation area C_{Lb} . This indicates that there will be an optimum value for HD after which an increase in HD will reduce the C_{Lb} significantly. Below this optimum for HD , a decrease in HD will also greatly reduce the C_{Lb} . This is a contrary result than in the slip case b of Fig. 4.12. Based on Fig 4.13, it is seen that the critical HD is around 0.5. Analysing Fig. 4.13, boundary slip in the pocket reduces C_{Lb} more than when there is no-slip (up to 86.57 %). However, this condition prevails when the HD is greater than the critical HD . For example, for $HD = 4$, compared with the no-slip bearing, slip in the pocket gives a 36.40 % lower value in C_{Lb} and if this result is associated with the load support W , the improvement in W by slip is around 2.47 % greater.

FIGURE 4.13: The cavitation area C_{Lb} as a function of pocket depth HD for a sliding single pocketed bearing with $h_o = 1.0 \times 10^{-6} \text{ m}$, $a = 0.002 \text{ m}$, $b = 0.0015 \text{ m}$, $B_o = 0.02 \text{ m}$ and $\alpha = 0.02 \text{ m}^2\text{s/kg}$.

Figure 4.14 shows the influence of the minimum film thickness h_o on the load support, in which the comparison of the performance is made between the pocketed bearing with full slip and no-slip as well as the presence of cavitation. In this figure, for each case there is a peak that separates two areas, the cavitation area and the no cavitation area. Open markers represent cavitated results, while for solid markers no cavitation is present. As can be seen in this figure, introducing slip into the pocketed slider bearing provides greater load support than the traditional pocketed bearing. However, this is not the case for the cavitation region. From a physical point of view, this new finding is interesting. It is evident that by introducing boundary slip at the contact, the tribological performance, in particular the load support, can be enhanced by up to 50 % based on Fig. 4.14.

FIGURE 4.14: The load support as a function of minimum film thickness for $h_d = 5.0 \times 10^{-6} \text{ m}$, $a = 0.004 \text{ m}$, $b = 0.006 \text{ m}$, $B_o = 0.02 \text{ m}$ and $\alpha = 0.02 \text{ m}^2\text{s/kg}$.

4.4. Multi texturing (theoretical and numerical approach)

In this section, multi texturing in a lubricated contact is examined based on the hypothesis that the texture has multiple effects on the lubrication performance.

4.4.1. Theoretical approach

The textured bearing configurations are shown in Figs. 4.15 (a) and 4.15 (b) representing a double texturing with slip and no-slip conditions. Here, slip is engineered to occur on the top edge of the pockets.

The derivation of the flow rate and pressure, which are the basis of this analysis, are presented in Chapter 3. Solving the set of governing equations is discussed in Paper D.

(a) no-slip condition.

(b) slip at pockets.

FIGURE 4.15: Schematic diagram of parallel sliding double pocketed bearings considering cavitation.

In the present work, the occurrence of the cavitation is characterized by zero pressure. Based on Figs. 4.15 (a) and 4.15 (b), in this subsection the cavitation area in the first pocket is denoted by C_{Lb} in which $C_{Lb} = b - X_b$ and the cavitation area in the second pocket is defined by C_{Ld} in which $C_{Ld} = d - X_d$. This means that when cavitation occurs, C_L is greater than zero.

In the present study, solving the lubrication problem for a pocketed slider bearing from a numerical point of view, the following solution method is proposed. For all computations the cavitation phenomenon is assumed to occur in the first and second pocket of the bearing. The cavitation equations (in modelling) are used to obtain X_b and X_d (non-cavitated region in the first and second pocket as seen in Fig. 4.15). Then, X_b and X_d will be used as an indicator of whether or not cavitation occurs. It should be noted that cavitation in the first pocket occurs if $0 < X_b < b$ where b is the first pocket length and cavitation in the second pocket occurs if $0 < X_d < d$ where d is the second pocket length.

The validity of the Reynolds equation in lubricated contacts with surface texturing has been discussed in published literature. The limits of this equation were well established based on [10]. In this study, care has been taken to ensure that the configurations considered fall within the validity domain of the Reynolds equation for the case of both single and double pocket texturing.

In this section, there is interest regarding the effect of texturing and cavitation. It should be noted that for the double texturing case, only the first pocket becomes the main interest. This means that the variation of texturing parameters is applied to the first pocket. The effect of cavitation on the lubrication in the second pocket is also explored, so as to analyse the correlation of the lubrication characteristic between the first and second pocket.

TABLE 4.2: *Characteristics of the double pocket bearing analysed.*

Parameter	Data setting	Unit
Pocket depth, h_{d1}	0.1; 0.5; 1.0 and 2.0	μm
Land film thickness, h_o	1×10^{-6}	m
Total length, B_o	0.02	m
Inlet length, a	0.002	m
Pocket length, b	0.0015; 0.006	m
Length between pockets, c	0.002	m
Second pocket length, d	0.003	m
Exit land length, e	0.0115; 0.007	m
Atmospheric pressure, P_{atm}	100	kPa
Cavitation pressure, P_{cav}	0	kPa
Sliding velocity, U	1	m/s
Slip coefficient, α	0.02	$\text{m}^2/\text{s/kg}$
Lubricant viscosity, η	0.01	Pa.s

Figure 4.16 depicts the length of the cavitation area in the first pocket as a function of pocket depth. The length of the cavitation area is similar to that of the single pocketed bearing for the slip (at pocket) or no-slip situation. The maximum pocket length (C_{Lb}) occurs when pocket depth $HD_I = 0.5$, that is 0.0746. The cavitation area decreases as a function of pocket depth HD_I . For the slip condition, the length of cavitation area decreases with increasing pocket depth HD_I . If the pocket depth HD_I is equal to zero, the length of the cavitation area still remains large. However, for the no-slip condition a contrary result is observed; the length of the cavitation area decreases significantly, which means that the cavitation will disappear if the pocket depth is reduced to zero (the smooth surface condition).

FIGURE 4.16: The cavitation area as a function of pocket depth sliding direction of the pocketed bearing for $h_o = 1 \times 10^{-6} \text{ m}$, $a = 0.002 \text{ m}$, $b = 0.0015 \text{ m}$, $e = 0.0165 \text{ m}$, $B_o = 0.02 \text{ m}$ and $\alpha = 0.02 \text{ m}^2/\text{s/kg}$.

Figure 4.17 reflects the no cavitation area of the first pocket X_b as a function of the first pocket depth for the slip (at pocket) and no-slip situation. The figure also depicts that the no cavitation area of the first pocket is not affected by the first pocket length. Slip at pocket and pocket depth improve X_b . For the slip situation, X_b increases with increasing pocket depth. For very small HD_I , the no cavitation area is close to zero. However, the no-slip situation has a minimum X_b for $HD_I = 0.5$, and it increases with increasing HD_I . For $HD_I = 0.0$ (i.e. smooth surface), X_d does not exist.

FIGURE 4.17: The no cavitation area in the first pocket as a function of pocket depth sliding direction of the multi pocketed bearing for $h_{d1} = 1 \times 10^{-6} \text{ m}$, $h_{d1} = 1 \times 10^{-6} \text{ m}$, $h_o = 1 \times 10^{-6} \text{ m}$, $a = 0.002 \text{ m}$, $c = 0.002 \text{ m}$, $d = 0.003 \text{ m}$, $B_o = 0.02 \text{ m}$ and $\alpha = 0.02 \text{ m}^2/\text{s/kg}$.

Figure 4.18 shows the comparison of the no cavitation area X_b in the cases of single and double pocket bearing for slip within the pocket and the no-slip situation respectively. It can be observed that the X_b in the case of a double pocket is not affected by the pocket length for both situations. In other words, such a pocket pattern gives the same X_b profile. Another interesting result is that the X_b in the case of double pocket bearing is smaller than that for a single pocket for both slip and no-slip conditions. The no cavitation area increases with greater first pocket length HD_1 and is close to zero for smaller HD_1 .

(a) slip condition.

(b) no-slip condition.

FIGURE 4.18: The no cavitation area as a function of pocket depth sliding direction of single and the double pocketed bearing for $h_{d1} = 1 \times 10^{-6} \text{ m}$, $h_{d1} = 1 \times 10^{-6} \text{ m}$, $h_o = 1 \times 10^{-6} \text{ m}$, $a = 0.002 \text{ m}$, $c = 0.002 \text{ m}$, $d = 0.003 \text{ m}$, $B_o = 0.02 \text{ m}$ and $\alpha = 0.02 \text{ m}^2/\text{s/kg}$.

4.4.2. Numerical approach

The hydrodynamic performance of a textured contact with slip is investigated through numerical simulations. In order to analyse the boundary slip, the slip is modelled using the critical shear stress criterion [107]. The flow is assumed to be isothermal, isoviscous, and laminar. Fig. 4.19 shows the schematic of a lubricated textured sliding contact in MEMS. The global contact geometry in this study is that of a rectangular textured sliding surface. It is assumed that the slip boundaries are engineered to all faces of the textured cells. In Fig. 4.19, an example of a textured cell with the grooves placed at the leading edge of the contact is presented. The lower smooth body slides at a constant velocity U , while the upper body is stationary. Such a lubrication system can be described by solving the three-dimensional modified Reynolds equation:

$$\left\{ \frac{\partial}{\partial x} \left(\frac{h^3}{12\eta} \frac{\partial D}{\partial x} \frac{h^2 + 4h\eta\alpha_h}{h(h + \eta\alpha_h)} \right) + \frac{\partial}{\partial y} \left(\frac{h^3}{12\eta} \frac{\partial D}{\partial y} \frac{h^2 + 4h\eta\alpha_h}{h(h + \eta\alpha_h)} \right) \right\} F =$$

$$\frac{U}{2} \frac{\partial}{\partial x} \left(\frac{h^2 + 2h\alpha_h\eta}{h + \eta\alpha_h} \right) - U \frac{\alpha_h\eta}{h + \eta\alpha_h} \frac{\partial h}{\partial x} + \frac{h}{2\eta} \frac{h\alpha_h\eta}{h + \eta\alpha_h} \left(\frac{\partial D}{\partial x} \frac{\partial h}{\partial x} + \frac{\partial D}{\partial y} \frac{\partial h}{\partial y} \right) F + \quad (4.1)$$

$$\left\{ \frac{U}{2} \frac{\partial}{\partial x} \left(\frac{D}{h} \frac{h^2 + 2h\alpha_h\eta}{h + \eta\alpha_h} \right) - \frac{D}{h} U \frac{\alpha_h\eta}{h + \eta\alpha_h} \frac{\partial h}{\partial x} \right\} (1 - F)$$

It should be noted that Eq. (4.1) is the modified Reynolds equation derived for over the whole bearing. The universal variable D and the index F are defined to anticipate the non-active (cavitation) zone, as well as the active zone.

In active zones:

$$\begin{cases} D = p, \\ F = 1 \end{cases} \quad D \geq 0$$

In the cavitation zones:

$$\begin{cases} D = r - h, \\ F = 0 \end{cases} \quad D < 0$$

In this study, dealing with the derivation of a modified Reynolds equation, the conditions for cavitation are based on the JFO (Jakobsson-Floberg-Olsson) theory to model the rupture zone for the textured surface. The methodology used with respect to the cavitation phenomena is similar to the work of Gherca et al. [62] and is explained in detail in the Appendix of Paper C.

(a) isometric view.

(b) longitudinal view (in x direction).

FIGURE 4.19: Schematic representation of a lubricated parallel sliding contact with a textured surface combined with boundary slip in the textured region.

Figure 4.20 shows the dimensionless load support W as a function of the groove number N for several values of dimensionless groove depth HD . It can be seen that for both cases (i.e. solely textured pattern and combined textured slip configuration), the trend of W is slightly similar. Increasing the groove number decreases W initially and then increases for all values of the groove depth. This results in the existence of a minimum value for W for a certain HD . The inflection of this point differs with the groove number. For example, for pure texturing, using a high groove number the inflection point occurs when N is approximately 6, but for HD which is less than 0.4, there is a shift of the inflection point towards the right-hand side of the graph. This indicates that for a surface which is "rough", it is necessary to increase the groove number in order to improve the load support. This is, however, not the case for a smooth surface (i.e. small HD). From Fig. 4.20, it is also observed that the inflection of W in the case of the combined textured slippage pattern is relatively small. The predicted results tend to be linear, especially for a low groove number. For high HD , the increase in W after N reaches the inflection point is clearly visible. It is interesting to note that this result is similar to the case of pure texturing. The difference lies in the inflection point, which refers to a smaller groove number. As a concluding remark, for the textured pattern both with and without wall slip, increasing the groove number will be beneficial to improving the load-carrying capacity. It is worth mentioning that when $N = 1$, the bearing configuration is similar to the Rayleigh step bearing.

(a) textured surface.

(b) combined textured slip surface.

FIGURE 4.20: Effect of the groove number N on the dimensionless load support W for several values of dimensionless groove depth HD . (Note: inset of Fig. 4.20 (b) shows the effect of the groove number N for the range of 0 to 2 on the dimensionless load support W).

4.5. Validation

As indicated in Fig. 4.21, the upper sample of the quartz glass ring was fixed into the upper shaft, which was kept stationary during tests. The arm installed on the upper shaft pressed against a load cell. The textured rings were fixed into a holder, which rested on a steel ball and was driven by a motor via a pin. In the experiments, a high speed CCD camera was used to capture the cavitation in the texture cells. A thermocouple was used to measure the average lubricant temperature in the holder. For details and the methodology of the experiments conducted by Zhang et al. [108], the reader is referred to the work of Zhang et al. [108].

FIGURE 4.21: Schematic description of the test rigs of a test for visualizing cavitation in a texture cell [108].

FIGURE 4.22: Geometry of the analytical model: a textured ring; b periodic sector; c section A–A [108].

FIGURE 4.23: Schematic diagram of parallel textured bearing model.

In this section, the three-dimensional thrust bearing as shown in Fig. 4.22 (b) is approached with infinite lubricated contact as shown in Fig. 4.23. The slider bearing specifications are shown in Table 4.3. The bearing length is the average circumference in Fig. 4.22 (b) as well as the length of the pocket. The rotational speed ω is converted with the radius of the ring into the linear velocity U .

TABLE 4.3: Main parameters of bearing analysis.

Parameter	Data setting	Unit
Bearing length, B_o	0.005757	m
Inlet length, a	0.000700	m
Pocket length, b	0.003224	m
Exit land length, c	0.001833	m
Atmospheric pressure, P_{atm}	100	kPa
Cavitation pressure, P_{cav}	0	Pa
External load, W	8	N

Figure 4.24 compares the results expressed by the cavitation ratio, i.e. $(b-X_b)/b$, of the experiment of Zhang et.al [108] with the analytical present study at the same bearing number ($\eta\omega/p$). In which η is the viscosity of the fluid, ω the rotational speed and p the average contact pressure ($p = 0.013$ MPa in this case). In the experiments, cavitation occurs at a rotational speed of 1.7 rpm and the cavitation area starts to increase with the increase in rotational speed up to 296 rpm, whereas the analytical solution predicts that cavitation will occur at 2.8 rpm and increase to 12 rpm; after that the cavitation ratio tends to be constant in the range of 0.68.

FIGURE 4.24: Experimental and analytic result of the cavitation area ratio versus the bearing number.

Overall, rather good agreement is found. The deviation found can be explained as follows:

1. The present study assumes an infinite length of the bearing (i.e. the effect of side leakage is ignored).
2. The curved pad in the experiment has been approached as straight.

4.6. Summary

The significance of the cavitation on the lubrication is shown based on the developed model. Analytically, the cavitation and no cavitation condition with slip and no-slip on parallel sliders bearing with a single pocket have been discussed in Paper A, while a double pocket bearing has been investigated in Paper D. It was found that having boundary slip can reduce the cavitation area and affect the pressure and load support on the textured slider bearing. Consequently, eliminating the cavitation phenomena from lubricated contacts is not recommended. The pressure distribution for the cavitation and no cavitation condition by varying inlet length, pocket length and pocket depth is discussed in detail in Paper B. The slip pattern from optimal inlet suction and pressure distribution are explained in detail in Papers E, F and G. In Paper H, the comparison of the lubrication analysis between Navier-Stokes and the Reynolds equation is explored. Lastly, the condition of multi texturing with slip and cavitation on the performance of a parallel slider bearing can be found in Paper C.

Chapter 5

Conclusion and future work

5.1. Conclusion

The main objective of the work presented is to investigate the effect of boundary slip and/or texturing in lubricated sliding devices considering cavitation. A theoretical lubrication equation based on the first-order Reynolds approach is developed by taking cavitation into account. The load support, the friction and the cavitation region are discussed as indicators of the lubrication performance.

The artificial slip surface is introduced in improving the load support and reducing the cavitation of a lubricated textured sliding contact. Within the current research, simulations/calculations were made to obtain an artificial slip surface placed in the inlet (suction) textured area and full slip. Two approaches are used, namely an analytical analysis and a numerical approach. Based on the analytical approach, single and double textured sliding contacts with slip are of particular interest. Numerically, multi-textured sliding contact with slip has been simulated.

In conclusion, it is shown that the cavitation pressure has a huge impact on the performance characteristics both for the slip pocketed bearing and no-slip pocketed one. Firstly, setting the cavitation pressure at zero helps to improve the load support and reduce friction. Secondly, the best performance can be achieved when the configuration of a pocket bearing with slip has a small ratio of inlet to outlet length. Thirdly, the most significant finding is that introducing slip in pocketed bearings enhances the effect of “suction” lubricant into the bearing, and therefore increases the lubricant flow. Consequently, this improves the pressure generation within the bearing in comparison with the no-slip pocketed bearing.

5.2. Future work

The long term goal is to develop engineering tools to prolong the life time of lubricated sliding devices. Therefore, five main research directions are suggested:

- (1) Include thermal effects in the developed lubrication equation.
- (2) For texturing, it is advised to put some effort in investigating cavitation in multi texture shapes.
- (3) The analysis should be extended for slip-textured sliding contacts with a certain slope incline ratio.
- (4) To perform cavitation research on rough surfaces combined with boundary slip.
- (5) Experiments are needed to validate the developed model for textured sliding contacts with slip areas.

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Part II

Part II

Paper A:

M. Muchammad, M. Tauviquirrahman, J. Jamari, D.J. Schipper, 2016, "An Analytical Approach on the Tribological Behaviour of Pocketed Slider Bearings with Boundary Slip Including Cavitation", *Lubrication Science*, Volume 29, Issue 3, pp. 133-152.
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Paper B:

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Paper C:

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Paper E:

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Paper G:

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Paper H:

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Paper A

M. Muchammad, M. Tauviquirrahman, J. Jamari, D.J. Schipper, 2016
An Analytical Approach on the Tribological
Behaviour of Pocketed Slider Bearings with
Boundary Slip Including Cavitation
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An Analytical Approach on the Tribological Behaviour of Pocketed Slider Bearings with Boundary Slip Including Cavitation

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Abstract Slider bearing performance depends on the boundary conditions at the interface between the solid surfaces and the fluid. This paper presents the combined effect of pockets and boundary slip on the load support and friction of parallel sliding systems using analytical solutions for a simple pocketed bearing. The effect of cavitation was of particular interest with respect to the inlet suction mechanism. It was demonstrated that applying boundary slip in a pocketed slider bearing gives less load support than the textured bearing without wall slip. Adding slip over the whole surface could retard the presence of cavitation. The influence of boundary slip is explored and was found to significantly affect the frictional behaviour.

Keywords: analytical method, slider bearing, slip, load support, friction.

1. Introduction

Surface texturing is an effective approach to increasing the tribological performance of lubricated mechanical components. It is known that introducing textured surfaces can affect friction and load support of sliding bearings significantly. In recent years, a great deal of effort has been deployed to investigate the influence of the texture's geometric parameters. It has been shown experimentally that such texturing enhances load support and reduces hydrodynamic friction; for instance, systems with two parallel sliding surfaces [1-2], mechanical seals [3], and reciprocating (cylinder-liner) contacts [4].

The effect of surface texturing on the tribological performance have been analysed numerically by many researchers, such as Arghir et al. [5], Sahlin et al. [6], Brajdic-Mitidieri et al. [7], Kligerman et al. [8], Fowell et al. [9], Dobrica and Fillon [10] and Han et al. [11]. There are two approaches to investigating textured surfaces: using CFD models based on Navier-Stokes [5-7,10-11] and using Reynolds [6,8-10]. Sahlin et al. [6] confirmed that by using CFD the introduction of a micro-groove on the stationary surface of a parallel slider leads to a net pressure build-up and hydrodynamic load support. In their research it was found that the effect of inertia on lubrication was dominant. Similarly, Han et al. [11] reported a three-dimensional CFD analysis of a spherical dimple on the stationary surface of a parallel sliding contact. An optimum micro dimple (depth and width) was defined that gives the greatest load support and lowest coefficient of friction. Brajdic-Mitidieri et al. [7] used the Navier-Stokes equation combined with a cavitation model to analyse the lubricant behaviour in plain pad bearings having a closed pocket. Such texture could produce a reduction in the friction coefficient and increase in load support. This effect depends on the bearing convergence ratio (inlet over outlet film thickness). It should be pointed out that these analyses [5-7,11] focused solely on single texture cells with centred dimples. Such configurations can lead to wrong conclusions in the prediction of hydrodynamic performance concerning the inertia effect [10].

In the case of multiple cells in hydrodynamic parallel sliding textured surfaces, Dobrica and Fillon [10] conducted a parametric study on the texturing parameters, dimension (depth and width), number and density of the cells to improve the hydrodynamic performance. They analysed the validity of the Reynolds equation and inertia effects in parallel textured sliders of infinite width using the Reynolds equation and commercial CFD simulation software (FLUENT). Among the original conclusions of this study was that inertia was noted to have a negative effect when applied to partially textured surfaces, i.e. reduced load support which was also found for the fully textured configurations. This result contradicts the works of Refs. [5-7,11] in the case of two parallel moving surfaces with a single cell texture, and the work of Kligerman et al. [8] in the case of the contact between a piston ring and cylinder liner with applying multiple texture cells. It was demonstrated that partial texturing has an advantage over full texturing of the surface in view of friction using a theoretical model based on a modified Reynolds equation [8]. A load support mechanism between parallel sliding surfaces was proposed by Fowell et al. [9]. They showed that the inlet suction mechanism in textured hydrodynamic bearings that has micropockets on its surface provides enhanced load support and reduced friction.

In most of the theoretical studies cited above, it has been assumed that the lubricated contact was represented by no-slip planes. Obviously, this assumption simplified the theoretical analysis and was, to some extent, applied successfully to engineering problems. However, with the continuous progress of the nano measurement techniques during recent years, a nano scale measurement is possible with respect to wall slip [12-14].

Recently, in addition to surface texturing, the use of an artificial slip surface is also of great interest with respect to lubrication. Such surface was introduced and applied to bearings with the help of microfabrication techniques. On micro scale such as in MEMS (micro-electro-mechanical system), the boundary condition, i.e. slip or no-slip situation, will play a very important role in determining the fluid flow behavior. Control of the boundary condition will

allow a degree of control over the hydrodynamic pressure in confined systems and will be important in a lubricated sliding contact. How to control the wall slip was one of the challenging research questions in recent investigations. However, in tribological lubricated system such as in bearing, the presence of the slip can be engineered in certain area by modifying the contacting surfaces. Based on literature survey, at least there are three mechanisms which lead to the presence of the slip. Firstly, the slip exists due to the effect of wetting and surface energy [15-18]. Secondly, the slip occurs due to the surface topography, in other words a low roughness value, as suggested by some researchers [17,19]. Thirdly, one can add components (additives) to a base oil to generate wall slip as investigated by Spikes and his group [20-21]. As a remark, it is clear that at macro scale, the roughness and wetting properties have a significant role in promoting the slip at the wall in lubricated systems. Also, in grease lubricated systems in which the film formation is governed by the base oil viscosity, the slip phenomenon can be controlled by modifying the roughness [22-23]. Several researchers such as Salant and Fortier [24], Wu et al. [25], Ma et al. [26-27] have explored the behaviour of the sliding contact using an optimized slip zone with respect to load support. The results of all these investigations show the existence of a lift force (load support) even there is no wedge effect (two parallel sliding surfaces) using such artificial slip zone. However, none of these studies have included surface texturing.

Very few researchers appear to have considered the interplay of surface texture and boundary slip effect on lubrication. Bayada and Meurisse [28] compared the slip/no-slip heterogeneity with roughness. They concluded that film rupture in diverging zones can be provoked by a certain artificial slip boundary configuration which is similar to that caused by geometrical roughness. However, in their study the combination of the effect of artificial slip and surface texture on the hydrodynamic behaviour was not conducted. Rao [29] employed artificial slip on a stationary surface having a single-groove at a slider and a journal bearing and calculated pressure and shear stress distributions. The modified Reynolds equation with the Navier-slip boundary condition was

solved numerically with regard to the effect of wall slip and single groove texture simultaneously. In a recent publication, Aurelian et al. [30] investigated the influence of boundary slip on load support and power loss in hydrodynamic fluid bearings. A simple textured/boundary slip combination was investigated. The main conclusion of their study was that the texture/slip zone geometry should be chosen carefully because an inappropriate choice can lead to a drastic deterioration of the bearing performance, especially in relation to the load support. However, the authors focused only on the slip property with zero critical shear stress. Tauviquirrahman et al. [31-34] explored the possibility of slip and texture in order to improve the performance characteristics. Their results indicate that the combined texture/slip pattern increase load-carrying capacity and decrease friction.

In most of the previous works the load support generation mechanisms were demonstrated using a simple simulation approach for a one-dimensional parallel bearing having a single cell texture without cavitation. Regarding the cavitation effect in a textured contact with slip on the lubrication performance, it is noted that in recent literature, only Wang and Lu [35] have taken into account the cavitation effect for a two-dimensional sleeve bearing. In the present study, an improvement in the performance of pocketed slider bearings is examined based on the wall slip phenomenon. The flow pattern in the bearing can be altered if the wall slip is located in a deterministic way in certain regions of the pocketed bearing surface. A modified form of the Reynolds equation containing the cavitation effect is derived in which wall slip is allowed to occur in a textured bearing.

It is interesting to note that when boundary slip is of particular interest in bearing analysis, for most previously published works such as the work of [24-27, 29-34], cavitation was ignored, and most of them used non-mass conservaton equations. As can be seen in the numerical results below, cavitation has a significant effect and the exclusion of cavitation in a mathematical model may lead to wrong conclusions. Therefore, to complement previous findings, an analytical lubrication model with boundary slip taking into account the cavitation effect is

proposed. As a note, in this paper the mechanisms of slip based on physical point of view are not investigated.

2. Mathematical Model and Its Solution

The textured bearing configuration is shown in Fig. 1. The textured surface is composed of a single recess with boundary slip. In hydrodynamic lubrication, the governing equations are described by the well-known Reynolds equation, which is a simplification of the Navier-Stokes equation. In the Reynolds approach, the assumptions used are as follows:

1. Small film thickness relative to the contact length.
2. Non-varying pressure across the film thickness.
3. The compressibility of the fluid is negligible.
4. The flow is laminar.
5. The gravity and inertia forces acting on the fluid can be ignored.

In the present work, an analytical equation based on first-order Reynolds theory is derived. In this way, the Reynolds equation is extended by considering slip and cavitation. Here, slip is engineered to occur on the whole stationary surface, thus also on the pocket surface (i.e. “full slip” configuration) as shown in Fig 1. The methodology used in the derivation of the extended Reynolds equation at the bearing surface is quite similar to that employed by Fowell [9] and is given in detail in Appendix A.

FIGURE 1: Schematic diagram of parallel textured bearing.

For a pocketed bearing with slip as illustrated in Fig. 1, expressions of the pressure and flow rate can be obtained. Applying pressure continuity at boundary points, a set of governing equations and constants of integration in the pressure expressions can be obtained for the flow rate,

see Appendix A for details. In Appendix A, the derivation of the flow rate and pressure, which are the basis of this analysis, are presented in Eqs. A.1-A.14. Solving the set of governing equations and employing the quantities in the nomenclature, the analytical solution for the slip case is:

$$P_2 \left[\left(h_p^3 + 3h_p^3 K_p \right) \frac{ab + bc}{ab^2} + \left(\frac{h_o^3 + 3h_o^3 K_o}{a} \right) \right]$$

$$= P_{atm} \left[\left(h_p^3 + 3h_p^3 K_p \right) \frac{ab + bc}{ab^2} + \left(\frac{h_o^3 + 3h_o^3 K_o}{a} \right) \right]$$

$$- 6\mu U \left[\left(h_p + h_p K_p \right) - \left(h_o + h_o K_o \right) \right] \quad (1)$$

For the analysis of a pocketed bearing with slip condition, the boundary slip is applied to all sides of the pocket cell (see Fig. 1). The load support W and friction F_f are of particular interest. The derivation of load support and friction either with cavitation or no-cavitation condition is given in Appendix A, Eqs. A.15-A.22. For all the following simulations, the standard one-dimensional bearing geometry is investigated, with values for the parameter as presented in Table 1.

In the present study, solving the lubrication problem for a pocketed slider bearing from a numerical point of view, the following solution method is proposed:

1. For all computations, the cavitation phenomenon is assumed to occur in the bearing. Here, the cavitation equations (Eqs. A.21 and A.22 of Appendix A) are used to obtain X_b (non-cavitated region as seen in Figure 1). Then, X_b will be used as an indicator of whether or not cavitation occurs. It should be mentioned that cavitation occurs if $0 < X_b < b$ where b is the pocket length.
2. If cavitation is present in the pocket bearing, the other cavitation equations (section A.1. in Appendix A) are implemented to predict the pressure as well as the friction force.
3. For the condition where cavitation does not exist (in this case, $X_b \leq 0$ or $X_b \geq b$), the non-cavitation equations (section A.1. in Appendix A) are used to calculate the pressure as well as the friction force.

TABLE 1: Characteristics of main bearing analysed.

Parameters	Data Setting	Unit
Bearing length B_o	0.02	m
Inlet length a	0.004	m
Pocket length b	0.006	m
Exit land length c	0.01	m
Atmospheric pressure p_{atm}	100	kPa
Cavitation pressure p_{cav}	0-100	kPa
Sliding Velocity u	1	m/s
Slip coefficient α	0.02	m ² /kg

3. Results and Discussions

As discussed by Fowell et al. [9], the existence of inlet suction with respect to the hydrodynamic lubrication mechanism has a significant effect in a textured bearing. Combining texture and wall slip on one surface will be explored in more detail in this paper.

The analysis of inlet suction is performed in full by deriving the equation for the net load support and the friction as shown in Appendix A. The main basis of derivation of this equation is equating the flows through the three parts of the bearing. In the analysis, the pressure at the inlet of the recess must be evaluated, whether or not it falls to the cavitation pressure.

Figure 2 shows the bearing load support plotted against the slip coefficient for various depth values of the pocket. It can be observed that the predicted load support is greater for a bearing having greater h_d . This prevails for a pocketed bearing either with slip or without slip. It matches well with previous research, for example by Fowell et al. [9] for the case of a no-slip ($\alpha = 0$) pocketed bearing. At first sight it seems that the load support of the pocketed bearing with slip is slightly greater than for that without slip ($\alpha = 0$); it is just 3-15 % depending on the value of h_d . However, for all values of the pocket depth considered here, the increase in the slip coefficient does not influence the predicted load support. This is to say that the performance of the pocketed bearing with respect to the load support improves very little by adding boundary slip.

FIGURE 2: Influence of slip coefficient on load support for several values of pocket depth h_d (Note: $h_o = 1.0 \times 10^{-6}$ m, $a = 0.004$ m, $b = 0.006$ m, $B_o = 0.02$ m).

Figure 3 shows the influence of the minimum film thickness h_o on the load support. One area of interest is the comparison of the performance between the pocketed bearing with slip and without slip. In this figure, for each case, there is a peak that separates two areas, the cavitation area and the no-cavitation area. Open markers represent cavitated results while for solid markers, no cavitation is present. As can be seen in this figure, introducing slip into the pocketed slider bearing increases the load support in comparison with the traditional pocketed bearing. However, this applies for the cavitation region. From the physical point of view, this new finding is interesting. As can be observed in Figure 4 (for example, in the case of a pocketed bearing with $h_d / h_o = 0.71$), adding boundary slip will generate a positive effect, i.e. more hydrodynamic pressure than that without boundary slip. The explanation of this effect could be that introducing boundary slip over the whole surface could retard the level of cavitation. On the other hand, slip gives rise to the inlet suction mechanism. For the non-cavitation region ($h_d / h_o = 0.31$ in this case), a contradictory result is found: slip generates less pressure and thus produces the reduced load support, as can be seen in more detail in Figure 5. This is as expected because the full slip configuration as investigated by other researchers [25-27,31-32] has a negative effect with respect to the load support in the absence of cavitation. In general, based on these figures, a specific observation can be drawn: slip is able to increase the load support when cavitation occurs.

Figure 6 shows the variation of the load support with pocket depth for two conditions- slip and no-slip. As can be seen from this figure, the lubrication performance of a textured slider bearing with slip is greater than that without slip for all pocket depths. In addition, Figure 6 shows that increasing the pocket depth results in the decrease-then-increase behaviour of the load support. This trend seems to be very clear for both conditions (slip and no-slip). Where $h_d = 0.5 \times 10^{-6}$ m, a minimum in load support is achieved for both cases, 0.4638 kN/m and 0.4568 kN/m, for the slip and no-slip situation respectively.

FIGURE 3: The load support as a function of minimum film thickness for the case of $h_d = 5.0 \times 10^{-6}$ m, $a = 0.004$ m, $b = 0.006$ m, $B_o = 0.02$ m. The slip coefficient $\alpha = 0.02$ m²/s/kg.

FIGURE 4: The pressure distribution as a function of coordinate in sliding direction for the pocketed bearing of interest with $h_d = 5.0 \times 10^{-6}$ m, $h_o = 7.0 \times 10^{-6}$ m, $a = 0.004$ m, $b = 0.006$ m, $B_o = 0.02$ m, and $\alpha = 0.02$ m²/s/kg. Cavitation is present.

FIGURE 5: The pressure distribution as a function of coordinate in sliding direction for the pocketed bearing of interest with $h_d = 5.0 \times 10^{-6}$ m, $h_o = 1.6 \times 10^{-5}$ m, $a = 0.004$ m, $b = 0.006$ m, $B_o = 0.02$ m, and $\alpha = 0.02$ m²/s/kg. Cavitation is absent.

Where $h_d = 0.5 \times 10^{-6}$ m, a minimum in load support is achieved for both cases, 0.4638 kN/m and 0.4568 kN/m, for the slip and no-slip situation respectively. Where the h_d is low (less than 2×10^{-6} m in this case), the predicted load support in the two cases is very close. It can be observed that the deviation between the case of the slip pocketed bearing and the no-slip pocketed bearing is lower than 8 %, even when the predicted load support is calculated for the lowest h_d , i.e. 0.03×10^{-6} m, which means that the bearing pattern is very close to that of a flat bearing. In other words, adding slip to the texture for low h_d does not influence the load support very much. From a physical point of view, it means that slip is not able to encourage the occurrence of cavitation; it is just the texture which influences the performance.

FIGURE 6: The load support as a function of pocket depth for the case of $h_o = 1.0 \times 10^{-6}$ m, $a = 0.004$ m, $b = 0.006$ m, $B_o = 0.02$ m. The slip coefficient $\alpha = 0.02$ m²/s/kg.

Figure 7 illustrates the load support which is plotted against the ratio b/B_o . As expected, for low h_d ($h_d = 2.5 \times 10^{-6}$ m in this case), the load support prediction between slip and no-slip textured bearing is very close. This is consistent with the previous results (see Fig. 6). As b/B_o increases, the load support decreases. For high h_d , the difference in load support between the slip and no-slip situation is not as small as is the case for low h_d , especially for b/B_o which is lower than, say, 0.1. Figure 7 also reflects the fact that the load support for a slip textured bearing is, for all h_d values considered here, greater than that of no-slip textured bearing. With respect to the utilization of slip (i.e. $\alpha = 0.02$ m²/s/kg) on a textured bearing, the load support can be improved from 1.2 to 11.8 % (at $h_d = 2.5 \times 10^{-6}$ m) and 5.4 to 38.4 % (at $h_d = 7.5 \times 10^{-6}$ m) depend on b/B_o . This means that the slip effect has a significant contribution on the pressure distribution and thus the load support especially when the pocket depth is great. In relation to the texture parameter (i.e. pocket depth), as is shown in Fig. 7, for h_d considered here, it can be observed that the greater the pocket depth, the greater the predicted load support, which matches well with the previous results, see Fig. 6. This prevails for both the slip and no-slip textured bearing.

FIGURE 7: The effect of pocket length b/B_o on the load support W . The profiles are calculated for $h_o = 1.0 \times 10^{-6}$ m, $a = 0.004$ m, $B_o = 0.02$ m, and $\alpha = 0.02$ m²/s/kg.

In Fig. 8, the load support of the textured bearing (with and without slip) versus the variation of the inlet length a/B_o is presented. The results are evaluated for several pocket depths. In this figure, data points where cavitation does not take place are presented by solid markers and those where cavitation occurs

are shown in open markers. It is interesting to observe that the no-cavitation region for the case of slip textured bearing both for h_d of 2.5×10^{-6} m and 7.5×10^{-6} m is wider than the results for the no-slip textured bearing. It seems counterintuitive that no-slip pockets cavitate more easily at low values of a/B_o than pockets with slip. The explanation is that the lubricant can move more easily at the slip surface and fill the pocket more quickly. Thus, adding slip can reduce the presence of cavitation and therefore leads to improved performance of a textured bearing.

FIGURE 8: The effect of inlet length on the load support. The profiles are calculated for $h_o = 1.0 \times 10^{-6}$ m, $b = 0.006$ m, $B_o = 0.02$ m, $\alpha = 0.02$ m²/s/kg.

Figure 8 also shows that in relation to the effect of the pocket depth, both for the slip and no-slip surface, the deeper the pocket, the more difficult it is for cavitation to take place. As explained by Fowell et al, [9] for a traditional (no-slip) textured bearing, this is due to the restriction of the flow which is higher for deeper pockets than for shallow ones. It seems that this condition also prevails for the slip pocket.

Figure 9 reflects the correlation between the inlet land width a/B_o and pocket depth h_d . It can be observed that the pocket depth has a strong influence on the inlet length as the initiator for cavitation both for the slip and no-slip cases. The minimum values are noted as the critical points of pocket depth h_d for both cases. This is to say that below these pocket depth values, the lubricant in the inlet starts to reduce the amount of flow in the pocket. It is worth noting that the critical value of h_d for the slip case is higher than for the no-slip case (see inset for details).

Obviously, as can also be seen from Fig. 9, the predicted values for a/B_o for the slip case is higher than for the no-slip case for all values of h_d considered here. From a physical point of view, it indicates that adding slip to the pocket bearing enables the presence of cavitation to be reduced. For example, for the no-slip textured bearing of interest with $h_d = 4.0 \times 10^{-6}$ m and the inlet length a/B_o 0.01, lubrication will generate the cavitation phenomena. However, if the slip is applied on such a bearing, cavitation can be prevented.

FIGURE 9: Cavitation as a function of inlet land width a/B_o and pocket depth h_d . The profiles are calculated for $h_o = 1.0 \times 10^{-6}$ m, $a = 0.004$ m, $b = 0.006$ m, $B_o = 0.02$ m, $\alpha = 0.02$ m²/s/kg, and $p_{cav} = 0$ kPa. The inset shows the predicted a/B_o for low values of h_d (i.e up to 2.0×10^{-6} m).

Figure 10 shows the effect of the cavitation pressure on the load support. It can be observed that the trend of the load support prediction both for the case of slip and no slip is identical, in other words the load support decreases linearly with cavitation pressure. When the cavitation pressure equals atmospheric pressure, the predicted load support falls to zero, indicating that the slip has an effect when the cavitation pressure is quite low as well as in the no-slip case to cause inlet suction.

From Fig. 11, it can be seen that for all values of pocket depth h_d , increasing the slip coefficient shows a reduction in friction. The increase in the α more than a specified number hardly changes that performance. Also, there is a decrease in friction in the range for the slip coefficient α from 0.02 to 0.10 m²/s/kg.

FIGURE 10: Effect of cavitation pressure on the load support. The profiles are calculated for $h_o = 1.0 \times 10^{-6}$ m, $h_d = 5.0 \times 10^{-6}$ m, $a = 0.004$ m, $b = 0.006$ m, $B_o = 0.02$ m, $\alpha = 0.02$ m²/s/kg.

FIGURE 11: Effect of slip coefficient on friction force. The profiles are calculated for $h_o = 1.0 \times 10^{-6}$ m, $a = 0.004$ m, $b = 0.006$ m, $B_o = 0.02$ m.

Figure 12 shows the influence of the cavitation pressure on friction. For a no-slip bearing, it can be observed that the cavitation pressure has no effect on the friction and hardly any for the slip bearing. It can be noted from this figure that the friction predicted for a bearing with slip is 50 to 200 times lower than the no-slip bearing depending on the cavitation pressure. In relation to the friction reduction by full slip, some of the earliest works [26-28,31] revealed the same result.

Figure 13 plots friction F_f versus the minimum film thickness h_o . It can be seen that the introduction of slip to the textured bearing results in low friction both for low h_o and high h_o . Increasing the h_o to the value of 1.0×10^{-6} m does not affect the friction very much. However, after h_o of 1.0×10^{-6} m, the friction decreases significantly. This phenomenon is opposite with

the no-slip textured bearing. For low h_o , the friction is very great. Only for high h_o (i.e. 1.0×10^{-4} m and above), the prediction of such bearing achieves similar values to that of a textured bearing with slip. It also indicates that adding slip to a textured bearing with high h_o does not have a significant effect in reducing friction.

FIGURE 12: Effect of cavitation pressure on friction for a bearing with $h_o = 1.0 \times 10^{-6}$ m, $h_d = 5.0 \times 10^{-6}$ m, $a = 0.004$ m, $b = 0.006$ m, $B_o = 0.02$ m, $\alpha = 0.02$ m²/s/kg.

FIGURE 13: Influence of the minimal film thickness h_o on friction F_f for a bearing with $h_d = 5.0 \times 10^{-6}$ m, $a = 0.004$ m, $b = 0.006$ m, $B_o = 0.02$ m, $\alpha = 0.02$ m²/s/kg.

Figure 14 presents the influence of h_d on the friction F_f . It can be observed that friction is not affected by the pocket depth significantly for the case of both a slip and no-slip textured bearing. However, as shown in previous results, slip generates very low friction.

FIGURE 14: Influence of the pocket depth h_d on friction F_f for bearing with $h_o = 1.0 \times 10^{-6}$ m, $a = 0.004$ m, $b = 0.006$ m, $B_o = 0.02$ m, $\alpha = 0.02$ m²/s/kg.

Figure 15 shows friction force F_f versus b/B_o . It can be seen that slip leads to the reduction in friction for two values of h_d and b/B_o considered here. The predicted reduction in friction is spectacular, both for high b ($b = 0.35 B_o$ in this case) and low b ($b = 0.01 B_o$). For example, for high b the friction reduction resulting from applying the slip is up to 8856 % and 4190 % respectively for $h_d = 2.5 \times 10^{-6}$ m and 7.5×10^{-6} m compared with without slip. The same order of friction reduction is also found for low b . This indicates that introducing slip in a sliding bearing is advisable with respect to the friction force reduction.

FIGURE 15: Plot of friction force F_f versus b/B_o for a bearing with $h_o = 1.0 \times 10^{-6}$ m, $a = 0.004$ m, $B_o = 0.02$ m, $\alpha = 0.02$ m²/s/kg.

Figure 16 depicts how friction varies with inlet length a/B_o for a pocketed bearing of interest with $h_o = 1.0 \times 10^{-6}$ m, $b = 0.006$ m, and $B_o = 0.02$ m. For the slip bearing, α is set to 0.02 m²/s/kg. It can be observed that increasing the inlet length a/B_o to a certain value leads to the

presence of cavitation, both for the no-slip and slip cases for two values of h_d considered. For the slip bearing, the value of “transition” from no-cavitation to cavitation is greater than that for the no-slip bearing for the two h_d values; extending the inlet length combined with slip is still able to prevent cavitation.

FIGURE 16: Plot of F_f versus inlet land width a/B_o for bearing with $h_o = 1.0 \times 10^{-6}$ m, $b = 0.006$ m, $B_o = 0.02$ m, $\alpha = 0.02$ m²/s/kg.

4. Conclusions

This paper focused on the possibility of inlet suction by the use of pockets in a bearing with and without boundary slip. A 1-D analytical model for a pocketed bearing with boundary slip has been developed and the effect of the boundary slip on the hydrodynamic performance (load support and friction) has been analysed. The following conclusions summarize the results of the present study:

1. The cavitation pressure has a huge impact on performance characteristics both for the slip pocketed bearing and no-slip pocketed bearing. Setting the cavitation pressure at very low helps to improve the load support and reduce friction.
2. The best performance can be achieved when the configuration of a pocket bearing with slip has a small ratio of inlet to outlet pad length.
3. The most significant finding is that introducing slip in pocketed bearings enhances the effect of “suction” the lubricant into the bearing, and therefore increases the lubricant flow. Consequently, this improves the pressure generation within the bearing in comparison with the no-slip pocketed bearing.

Nomenclatures

a	= inlet length
b	= pocket width
B_o	= slider length
c	= outlet land length
F_f	= friction
h_1	= film thickness at A
h_2	= inlet land film thickness at B
h_{2p}	= film thickness in pocket at B
h_3	= film thickness in outlet land at C
h_{3p}	= film thickness in pocket at C
h_d	= height of microtexture pocket (excluding film thickness)
h_o	= minimum film thickness in inlet land at B
p_2	= pressure at B
p_3	= pressure at C
p_{atm}	= atmospheric pressure
p_{cav}	= cavitation pressure
U	= sliding velocity
W	= load support
q	= flow rate
x	= coordinate in sliding direction
X_b	= non-cavitated fraction of pocket breadth
z	= coordinate through film thickness
μ	= lubricant dynamic viscosity
α_s, α_h	= slip coefficient at surface s (moving) and h (stationary)

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Appendix A: Detail of Mathematical Model

The analysis below is based on continuity of flow through the bearing and uses the terminology shown in Fig. 1. For the condition in which the slip velocity at both the moving and stationary solid-liquid interface follows the Navier slip model, the corresponding flow rate equation can be expressed respectively as [31]:

$$q_x = -\frac{\partial}{\partial x} \left(\frac{h^3}{12\mu} \frac{h^2 + 4\mu h(\alpha_h + \alpha_s) + 12\mu^2 \alpha_h \alpha_s}{h(h + \mu(\alpha_h + \alpha_s))} \frac{\partial p}{\partial x} \right) + \frac{U}{2} \frac{\partial}{\partial x} \left(\frac{h^2 + 2\mu h \alpha_h}{h + \mu(\alpha_h + \alpha_s)} \right) + \frac{h}{2\mu} \frac{\partial p}{\partial x} \frac{\partial h}{\partial x} \frac{\mu h \alpha_h + 2\mu^2 \alpha_h \alpha_s}{h + \mu(\alpha_h + \alpha_s)} - U \frac{\partial h}{\partial x} \frac{\mu \alpha_h}{h + \mu(\alpha_h + \alpha_s)} \quad (\text{A.1})$$

If slip is present only on the stationary surface ($\alpha_s = 0$ but $\alpha_h \neq 0$), equation (A.1) will reduce to:

$$q_x = -\frac{\partial}{\partial x} \left(\frac{h^3}{12\mu} \frac{\partial p}{\partial x} \frac{h^2 + 4\mu h \alpha_h}{h(h + \mu \alpha_h)} \right) + \frac{U}{2} \frac{\partial}{\partial x} \left(\frac{h^2 + 2\mu h \alpha_h}{2(h + \mu \alpha_h)} \right) \quad (\text{A.2})$$

For the first-order Reynolds equation, the flow rate is given:

$$q_x = \frac{U}{2} \frac{h^2 + 2\mu h \alpha_h}{h + \mu \alpha_h} - \frac{h^3}{12\mu} \frac{h + 4\mu \alpha_h}{h + \mu \alpha_h} \frac{dp}{dx} \quad (\text{A.3})$$

A.1. No cavitation in recess

A.1.1. Load support

For the areas of the bearing containing parallel surfaces the film thickness h is constant and the pressure gradient must be linear (see Eq. A.3). Equation (A.3) can be modified for each region of the bearing to

$$q_{AB} = \frac{U}{2} \frac{h_o^2 + 2\mu h_o \alpha_h}{h_o + \mu \alpha_h} - \frac{h_o^3}{12\mu} \frac{h_o + 4\mu \alpha_h}{h_o + \mu \alpha_h} \left(\frac{p_2 - p_{atm}}{a} \right) \quad (\text{A.4})$$

$$q_{BC} = \frac{U}{2} \frac{h_p^2 + 2\mu h_p \alpha_h}{h_p + \mu \alpha_h} - \frac{h_p^3}{12\mu} \frac{h_p + 4\mu \alpha_h}{h_p + \mu \alpha_h} \left(\frac{p_3 - p_2}{b} \right) \quad (\text{A.5})$$

$$q_{CD} = \frac{U}{2} \frac{h_o^2 + 2\mu h_o \alpha_h}{h_o + \mu \alpha_h} - \frac{h_o^3}{12\mu} \frac{h_o + 4\mu \alpha_h}{h_o + \mu \alpha_h} \left(\frac{p_{atm} - p_3}{c} \right) \quad (\text{A.6})$$

with one condition $q_{AB} = q_{BC} = q_{CD}$ based on volume conservation.

When q_{AB} is set to be equal to q_{CD}

$$\frac{U}{2} \frac{h_o^2 + 2\mu h_o \alpha_h}{h_o + \mu \alpha_h} - \frac{h_o^3}{12\mu} \frac{h_o + 4\mu \alpha_h}{h_o + \mu \alpha_h} \left(\frac{p_2 - p_{atm}}{a} \right) = \frac{U}{2} \frac{h_o^2 + 2\mu h_o \alpha_h}{h_o + \mu \alpha_h} - \frac{h_o^3}{12\mu} \frac{h_o + 4\mu \alpha_h}{h_o + \mu \alpha_h} \left(\frac{p_{atm} - p_3}{c} \right) \quad (A.7)$$

or:

$$\left(\frac{p_2 - p_{atm}}{a} \right) = \left(\frac{p_{atm} - p_3}{c} \right) \quad (A.8)$$

For the non-cavitation case, P_2 is unknown. Equating the flow rate in inlet land and recess, $q_{AB} = q_{BC}$

$$\frac{U}{2} \frac{h_o^2 + 2\mu h_o \alpha_h}{h_o + \mu \alpha_h} - \frac{h_o^3}{12\mu} \frac{h_o + 4\mu \alpha_h}{h_o + \mu \alpha_h} \left(\frac{p_2 - p_{atm}}{a} \right) = \frac{U}{2} \frac{h_p^2 + 2\mu h_p \alpha_h}{h_p + \mu \alpha_h} - \frac{h_p^3}{12\mu} \frac{h_p + 4\mu \alpha_h}{h_p + \mu \alpha_h} \left(\frac{p_3 - p_2}{b} \right) \quad (A.9)$$

If, $K_o = \frac{\mu \alpha_h}{h_o + \mu \alpha_h}$ and, $K_p = \frac{\mu \alpha_h}{h_p + \mu \alpha_h}$ then

$$\frac{U h_o}{2} (1 + K_o) - \frac{h_o^3}{12\mu} (1 + 3K_o) \left(\frac{p_2 - p_{atm}}{a} \right) = \frac{U h_p}{2} (1 + K_p) - \frac{h_p^3}{12\mu} (1 + 3K_p) \left(\frac{p_3 - p_2}{b} \right) \quad (A.10)$$

From Eq. (A.8), it is known that $p_3 = p_{atm} - \frac{c}{a}(p_2 - p_{atm})$ therefore

$$\left(\frac{p_3 - p_2}{b} \right) = \frac{ab + bc}{ab^2} (p_{atm} - p_2) \quad (A.11)$$

Substitution of Eq. (A.8) in Eq. (A.9) gives

$$p_2 \left[\left(h_p^3 + 3h_p^3 K_p \right) \frac{ab + bc}{ab^2} + \left(\frac{h_o^3 + 3h_o^3 K_o}{a} \right) \right] = p_{atm} \left[\left(h_p^3 + 3h_p^3 K_p \right) \frac{ab + bc}{ab^2} + \left(\frac{h_o^3 + 3h_o^3 K_o}{a} \right) \right] - 6\mu U \left[(h_p + h_p K_p) - (h_o + K_o) \right] \quad (A.12)$$

where

$$C_p^* = h_p^3 + 3h_p^3 K_p$$

$$C_o^* = h_o^3 + 3h_o^3 K_o$$

$$C_p = h_p + h_p K_p$$

$$C_o = h_o + h_o K_o$$

Combining Eqs. (A.11) and (A.12) to eliminate P_3 gives

$$\left(\frac{ab+bc}{ab^2} C_p^* + \frac{C_o^*}{a} \right) = p_{am} \left(\frac{ab+bc}{ab^2} C_p^* + \frac{C_o^*}{a} \right) - 6\mu U (C_p - C_o) \quad (\text{A.13})$$

and thus,

$$p_3 = p_{am} + \frac{6\mu U (C_p - C_o) bc}{aC_p^* + cC_p^* + bC_o^*} \quad (\text{A.14})$$

Due to linearity between the pressure gradient in the inlet and outlet lands and in the recess, each of three areas of the bearing has a triangular pressure distribution that can be simply integrated to determine the load support. Therefore, the total normal force reads:

$$w = w_{AB} + w_{BC} + w_{CD} - p_{am} (a + b + c) \quad (\text{A.15})$$

$$W = -\frac{6\mu U (C_p - C_o) ab}{aC_p^* + cC_p^* + bC_o^*} \left(\frac{a+b}{2} \right) + \frac{6\mu U (C_p - C_o) bc}{aC_p^* + cC_p^* + bC_o^*} \left(\frac{b+c}{2} \right) \quad (\text{A.16})$$

Finally, the load support reads

$$w = \frac{6\mu U (C_p - C_o) b}{C_p^*} \frac{1}{2} \left(\frac{c^2 - ab - a^2 + bc}{a + c + b \frac{C_o^*}{C_p^*}} \right) \quad (\text{A.17})$$

A.1.2. Shear Stress

The shear stress at the lower surface can be derived as follows:

$$\tau = -\frac{\mu U}{h + \mu \alpha_h} - \frac{h}{2} \frac{\partial p}{\partial x} \frac{h + 2\mu \alpha_h}{h + \mu \alpha_h} \quad (\text{A.18})$$

For the case of no cavitation, the friction force reads:

$$F_f = F_{fAB} + F_{fBC} + F_{fCD} \quad (\text{A.19})$$

$$F_f = -\mu U \left[\frac{a}{h_o + \mu \alpha_h} + \frac{c}{h_o + \mu \alpha_h} + \frac{b}{h_p + \mu \alpha_h} + \left(\frac{3(C_p - C_o)^2 b(a+c)}{aC_p^* + cC_p^* + bC_o^*} \right) \right] \quad (\text{A.20})$$

A.2. Cavitation in recess

A.2.1. Load support

For the case of cavitation in recess, p_2 is assumed to be equal to p_{cav} , so $q_{AB} = q_{BC}$ gives

$$\frac{Uh_o}{2}(1+K_o) - \frac{h_o^3}{12\mu}(1+3K_o)\left(\frac{p_{cav}-p_{atm}}{a}\right) = \frac{Uh_p}{2}(1+K_p) - \frac{h_p^3}{12\mu}(1+3K_p)\left(\frac{p_3-p_{cav}}{X_b}\right) \quad (A.21)$$

where

$$X_b = \frac{C_p^*(p_3-p_{cav})}{6\mu U(C_p-C_o) - C_o^*\left(\frac{p_{atm}-p_{cav}}{a}\right)} \quad (A.22)$$

It should be noted that for the cavitation zone, the pressure must be integrated separately, i.e. the non-cavitated area and cavitated area of the recess, so the total load support reads

$$w = w_{AB} + w_{BB} + w_{BC} + w_{CD} - p_{atm}(a+b+c) \quad (A.23)$$

$$\begin{aligned} w = & \left[p_{atm}a + \frac{a^2}{2a}(p_{cav}-p_{atm}) \right] + [p_{cav}b(1-X_b)] \\ & + \left[p_{cav}X_b + \frac{6\mu U(X_b)^2}{2C_p^*}(C_p-C_o) + \frac{C_o^*(X_b)^2}{2aC_p^*}(p_{cav}-p_{atm}) \right] \\ & + \left[p_{atm}c - \frac{c^2}{2a}(p_{cav}-p_{atm}) \right] - p_{atm}(a+b+c) \end{aligned} \quad (A.24)$$

A.2.2. Shear Stress

$$F_f = F_{fAB} + F_{fBB} + F_{fBC} + F_{fCD} \quad (A.25)$$

$$F_f = - \left\{ \mu U \left[\frac{a}{h_o + \mu\alpha_h} + \frac{c}{h_o + \mu\alpha_h} + \frac{X_b}{h_p + \mu\alpha_h} \right] + \left[\left(\frac{C_p}{2} - \frac{C_o}{2} \right) \left(1 + \frac{c}{a} \right) (p_{atm} - p_{cav}) \right] \right\} \quad (A.26)$$

Paper B

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Theoretical Investigation of Boundary Slip on the Hydrodynamic Lubrication Performance in Pocketed Bearings Including Cavitation

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Abstract Boundary slip in bearings is becoming more and more popular for improving the hydrodynamic performance (pressure and shear stress). However, most of previously published works regarding textured bearings ignored the cavitation effect, thereby making their results questionable. The main aim of this paper is to investigate the hydrodynamic performance of slip pocketed bearings considering cavitation by a theoretical approach. Cavitation is of particular interest with respect to pressure generation and shear stress. One main result presented here is that slip over the whole surface could retard the presence of cavitation and therefore the generated pressure and shear stress could be optimal. The work presented here leads to a design reference guideline that could be used by the designer/engineer to design slip pocketed bearings for improving the hydrodynamic performance.

Keywords: Cavitation, hydrodynamic performance, boundary slip, pocketed bearing.

1. Introduction

Surface texturing is an effective approach to increasing the tribological performance of mechanical components. The performance in hydrodynamic lubrication by surface texturing has been analysed by both analytical and experimental research over the last decades. Experimentally, surface texturing is analysed in

parallel sliding contacts [1-2], piston rings, cylinder-liner contacts [3] and mechanical seals [4] and it is shown that introducing textured surfaces affects friction and load support of sliding bearings [1-6]. It was found that the textured surfaces increase load support and reduce the friction force. The effect of surface texturing on the tribological performance has been analysed numerically by many researchers [5-9]. Two models were used for investigating textured surfaces: computational Fluid Dynamic (CFD) models based on the Navier-Stokes equation [5,8-10] or using the Reynolds equation [7,11-14]. The models were applied to macro textured model surfaces (single texture cell) [5,8-9,11-12,15 and micro-texture (multi texture configuration) [1,6,12-14,16-17].

Analytically, based on the first-order Reynolds equation, flow and cavitation in the pocket was proposed by Fowell et al. [7]. The results showed that the inlet suction mechanism in textured pocket bearings provides enhanced load support and reduces friction. Shi and Ni [10] investigated the effect of a macro-groove texture in a sliding contact including cavitation using CFD. They concluded that with the increase in cavitation pressure, the hydrodynamic pressure becomes more pronounced, while the friction does not change very much. In the case of multiple cells in a hydrodynamic parallel sliding contact, Drobica and Fillon [6] conducted a parameter study on the texturing parameters, dimension of depth and width, number and

density of the cells to improve the hydrodynamic performance.

In most of the theoretical studies cited above, it has been assumed that the lubricated contact was represented by a no-slip condition. Obviously, the use of an artificial slip surface has become popular with respect to lubrication, as it gives a better tribological performance. Two deterministic slip surface modes are used currently; homogeneous slip surface (i.e. slip applied over the whole surface) and partial slip (i.e. surface consisting of slip and no-slip areas). The great challenge for hydrodynamic lubrication is to apply surface texturing and the boundary slip effect [12-14,18-21]. Tauviquirrahman et al. [12-13] studied surface texturing and boundary slip in lubricated sliding contacts. They concluded that, using a combination of texturing and boundary slip, a correct partially textured surface with slip leads to better load support than when using a partially textured surface without boundary slip.

In most of the previous works, the load support generation mechanism was demonstrated using a simple simulation approach for texturing with cavitation [7,10,22-23] or slip including cavitation [21,23]. Giocopini et al. [22] carried out comparisons between analytical and numerical models for a range of scenarios in which cavitation may occur. They concluded that excellent agreement was found, which means that solution of complex cavitation problems involving both rupture and reformation of the liquid film is found. Wang et al. [23] investigated the hydrodynamic pressure generation mechanism using a simple simulation approach for a one-dimensional parallel slider bearing having textured grooves including cavitation. The results presented show that the inlet roughness, inlet suction and quasi-antisymmetric integration are essential features that generate hydrodynamic pressure in parallel bearings.

The present paper proposes a novel analytical equation based on the Reynolds equation considering cavitation combined with boundary slip model in a pocket bearing. In particular, the investigation is focused on the generation of the pressure distribution and the shear stress in pocket bearings.

2. Mathematical Model

Figure 1 gives a schematic illustration of a parallel textured (pocketed) lubricated sliding bearing with boundary slip on the stationary surface. In the present study, the slip is engineered to occur on the whole stationary surface of the bearing. The main characteristics of the bearing studied are presented in Table 1. In a classical hydrodynamic lubrication, the governing equation for the full fluid region is described by the well-known Reynolds equation. The isoviscous Newtonian one-dimensional Reynolds equation is derived from the Navier-Stokes equation that assumes an incompressible flow, ignoring the convective effects in the film and cavitation. The mathematical model used is a modified Reynolds equation with slip at the bearing surface and is quite similar to that employed by Fowell et al. [7]. Details of the equations used are given in Appendix A. In the previous works [1-6,8-9,11-14,15-17], cavitation is ignored. In the present study the cavitation effect is taken into account, and an algorithm is proposed.

The pocket bearing with full slip and no slip on the stationary surface is illustrated in Figure 1. For such a pattern, the expressions for the pressure and flow rate can be obtained. The derivation of the pressure and flow rate, that are the basic of this analysis, are presented in Appendix. The derivation of the analytical equation for the full slip condition is given as follows:

$$p_2 \left[\left(h_p^3 + 3h_p^3 K_p \right) \frac{ab + bc}{ab^2} + \left(\frac{h_o^3 + 3h_o^3 K_o}{a} \right) \right] = p_{atm} \left[\left(h_p^3 + 3h_p^3 K_p \right) \frac{ab + bc}{ab^2} + \left(\frac{h_o^3 + 3h_o^3 K_o}{a} \right) \right] - 6\mu U \left[\left(h_p + h_p K_p \right) - \left(h_o + K_o \right) \right] \quad (1)$$

For the analysis of the pocket bearing with boundary slip, the boundary slip is employed on all sides of the textured cell on the stationary surface (see Figure 1). The pressure distribution, shear stress and cavitation phenomena are of particular interest. The derivation of the pressure distribution and shear stress for the no-

cavitation or cavitation condition is given in Appendix A, Equations A.4 – A.27.

TABLE 1: Characteristics of the bearing analysed.

Parameters	Data Setting	Unit
Bearing length B_o	0.02	m
Inlet length a	0.002	m
Pocket length b	0.003	m
Exit land length c	0.015	m
Atmospheric pressure p_{atm}	100	kPa
Cavitation pressure p_{cav}	0-100	kPa
Min. film thickness h_o	1	μm
Sliding Velocity u	1	m/s
Slip coefficient α	0.02	$\text{m}^2/\text{s/kg}$

FIGURE 1: Schematic diagram of parallel sliding pocketed bearing considering cavitation.

In the present study, solving the lubrication problem for a pocketed slider bearing from numerical point of view, the solution method is proposed as follows:

1. For all computations, the cavitation phenomenon is assumed to occur in the bearing. Here, the cavitation equations (Eqs. A.19 and A.27 of Appendix A) to obtain X_b (no-cavitation region as seen in Figure 1) are used. Then, X_b will be used as an indicator whether cavitation occurs or not. It should be mentioned that cavitation occurs if $0 < X_b < b$ where b is the pocket length.

2. If cavitation is present in the pocketed bearing, the other cavitation equations (section A.2. in Appendix A) are implemented to predict the pressure as well as the friction force.
3. For the condition that cavitation does not exist (in this case, $X_b \leq 0$ or $X_b \geq b$), the non-cavitation equations (section A.1. in Appendix A) are used to calculate the pressure as well as the friction force.

3. Results and Discussions

In the present study, the region of cavitation (see Figure 1) is defined by the length X_b , similar to the work of Fowell et al. [7].

Figure 2 shows the effect of the non-dimensional texture depth HD ($HD = h_d/h_o$) on the dimensionless pressure. It can be observed that for $HD = 5$ cavitation will occur in the pocket whether slip is applied or not. However, as depicted in Figure 2, for the full slip case, the length of X_b is greater than for the case of the no-slip condition. It means that the application of slip tends to prevent the occurrence of cavitation. For a greater pocket depth, for example $HD = 10$, the cavitation will occur for the no-slip pocketed bearing, whereas for the full slip bearing the cavitation does not exist. Before cavitation occurs, the value of P_3 is the same for both the slip and no-slip conditions. When no-cavitation occurs, P_3 tends to decrease, whereas P_2 will increase (see Figure 2).

For simplicity in the present study, the zero shear stress in the region of cavitation (see Figure 1), which is defined by the length of X_b , is assumed, similar to the work of Fowell et al., [7].

Figure 2 shows the effect of non-dimensional texture depth HD on the dimensionless pressure. It can be observed that for $HD = 5$ ($HD = h_d/h_o$), the cavitation will occur in the pocket whether or not slip is applied. However, as depicted in Figure 2, for the full slip case, the length of X_b is greater than for the case of the no-slip condition. From a physical point of view, it means that the application of slip tends to prevent the occurrence of cavitation. For a greater pocket depth, for example $HD = 10$, the cavitation will occur for the no-slip pocketed bearing, whereas

for the full slip bearing the cavitation does not exist as expected. Before cavitation occurs, the value of p_3 is the same for both the slip and no-slip conditions. When cavitation occurs, p_3 tends to decrease, whereas p_2 will increase (see Figure 1).

FIGURE 2: The effect of non-dimensional texture depth HD on the dimensionless pressure p^* .

Figure 3 depicts the effect of the non-dimensional slip length A ($A = \mu a / h_o$) on the dimensionless pressure p^* . From this figure, it can be seen that introducing boundary slip on the pocketed parallel slider bearing leads to shorten the cavitation area. The result strengthens the previous result, which states that slip has a positive effect on reducing the cavitation effect. Another interesting result is that the value of p_3 (the peak of the pressure distribution) is the same for both the slip and no-slip case. Regarding the variations of the dimensionless slip length (50 and 200 in this case), the predicted dimensionless pressure profiles have a similar trend. This indicates that the slip length considered here refers to the optimal value and consequently the optimal performance with respect to the pressure and thus the load-carrying capacity.

FIGURE 3: The effect of non-dimensional slip length value A on the dimensionless pressure p^* .

FIGURE 4: The effect of non-dimensional pocket length b^* on the dimensionless pressure p^* .

Figure 4 compares the no-slip with the full-slip condition as a function the pocket length b^* ($b^* = b / B_o$). From this figure, it can be seen that if b^* is 0.15, the area below the pressure distribution for the full-slip condition and thus the load support is slightly higher than for the no-slip situation. However, when the value b^* is set to 0.05, the area below the pressure distribution for the full slip situation is much smaller than for the no-slip contact. This is because for the full-slip condition, due to the absence of the cavitation area the peak value p_3 is smaller than the no-slip situation. On the other hand, for the full slip-slip condition the increase in p_2 is not significant in comparison with the decrease in p_3 . In general, shortening the pocket length can reduce the cavitation area, and the cavitation effect when full slip is applied.

Figure 5 shows the relationship between non-dimensional inlet length a^* ($a^* = a / B_o$) and dimensionless pressure. It can be observed that the smaller the non-dimensional inlet length, the

greater the load support. When a^* is 0.025, the cavitation effect reduces significantly both for slip and no-slip. It is also seen that for full-slip, there is a slight increase in the p_2 value. However, the peak value of p_3 is lower than for the no-slip situation. It is also noted that for $a^* = 0.025$ and the no-slip condition, cavitation still occurs, however the length of the cavitation area is very small. This indicates that for small a^* , full slip does not have much effect on pressure. This is also the case for high a^* ($a^* = 0.1$ here). Full slip tends to reduce the cavitation area slightly. In general, shortening the inlet length significantly increases the load support. However, this positive effect (i.e. the high load support) decreases when full slip is introduced.

FIGURE 5: The effect of non-dimensional inlet length a^* on the dimensionless pressure p^* .

As discussed by Fowell et al. [7], the cavitation pressure has a strong effect on producing inlet suction, as a mechanism influencing the pressure distribution. In this paper, such a mechanism is also found both for the slip and no-slip condition. Figure 6 shows the effect of the cavitation pressure on the dimensionless pressure profile. From the figure, it can be seen that if the cavitation pressure is set equal to the atmosphere pressure ($p^* = 1.0$ in this case), both p_2 and p_3 will have the same value as the atmosphere pressure. Therefore, the area below the pressure profile becomes very small. This means that some inlet suction results in decreased load support. It is also observed that increasing the cavitation pressure will increase the area below the pressure profile due to the increased value p_2 and p_3 , which leads to improved load support. For the values of cavitation pressure considered here, p^* is 0,

giving the greatest p_3 pressure. As stated before, full slip will shorten the cavitation area.

Figure 7 shows the effect of the dimple depth HD ($HD = h_d/h_o$) on the shear stress τ^+ . It can be seen that for the no-slip case with $HD = 5$ and $HD = 10$, the shear stress at the inlet and outlet has the same trend, but in the pocket area there is a decrease in the shear stress. It is obvious that the shear stress is relatively high at the inlet and outlet, but in the pocket area the shear stress is quite low. When boundary slip is introduced, the shear stress becomes very low (close to zero) at the inlet and outlet; this is a positive effect as wanted. In general, observing the shear stress distribution, the boundary slip condition generates a lower friction force than for the no-slip case.

FIGURE 6: The effect of non-dimensional cavitation pressure p^* on the dimensionless pressure p^+ profile.

FIGURE 7: The effect of non-dimensional texture depth HD on the dimensionless shear stress τ^+ .

Figure 8 depicts the relationship between the non-dimensional slip length Λ and the dimensionless shear stress. It can be seen that for the no-slip case, in the pocket area there is a

decrease in the shear stress, while in the inlet and outlet a high shear stress value is noted. This condition contradicts the full-slip condition; the inlet and outlet area have very low shear stress values approach in a zero, however there is higher shear stress in the pocket. The results also show that the lubricated contact with slip length A produces the same trend in the shear stress profile. Again, as shown before, the shear stress in the pocket for the no-slip condition is greater than that for the full-slip situation. From the physical point of view, to ensure the volume conservation, the shear stress must consist of two terms, i.e. the couette shear (due to the velocity difference) and poiseuille term (due to the pressure gradient). In the non-pocket area the term of couette shear is much more dominant than the poiseuille term. In the pocket area, the poiseuille term is greater than the couette shear. However, introducing the slip for all surfaces of the bearing will reduce the couette shear term considerably. Consequently, the value of shear stress in the pocket area is still high.

FIGURE 8: The effect of non-dimensional slip length value A on the dimensionless shear stress τ^+ .

Figure 9 reveals the relationship between the dimensional shear stress and the pocket length variation on the parallel slider bearing. It can be seen for the no-slip condition, a higher shear stress value in the pocket is noted than for the slip situation. However, for the inlet and outlet area, both situations have similar values. For the full-slip condition, the shear stress in the pocket bearing has the same value for all values of the pocket length considered, whereas the shear stress in the inlet and outlet are not really affected by the variation of the pocket bearing length.

Figure 10 shows the effect of the non-dimensional inlet length a^* on the dimensionless shear stress τ^+ profile. It can be observed that shortening the length will lead to a reduction in the length of the cavitation area. The other interesting result is that if a short a^* is combined with the full slip in the bearing, cavitation does not occur. For the full slip condition, the shear stress profile for $a^* = 0.025$ and 0.1 are given, showing that the area below the profile for $a^* = 0.025$ is larger because cavitation does not happen, whereas for the no-slip condition the length of the cavitation area decreases with decreasing a^* . It can be said that the shorter the inlet length, the lower the shear stress in the pocket and finally the lower the friction force.

FIGURE 9: The effect of non-dimensional pocket length b^* on the dimensionless shear stress τ^+ .

FIGURE 10: The effect of non-dimensional inlet length a^* on the dimensionless shear stress τ^+ .

Figure 11 depicts the relationship between the non-dimensional cavitation pressure and the dimensionless shear stress. Varying the cavitation pressure for the no-slip condition gives the same shear stress for the inlet and outlet, whereas in the pocket it gives results in different cavitation lengths. For $p^* = 0.0$ and 0.5 ,

the shear stress in the pocket has the same value. For the pocketed bearing with full slip, the difference in cavitation pressure values will give different lengths of the cavitation area. The shear stress in the inlet and outlet is close to zero. However, for the pocket area, the shear stress values have the same value for $p^* = 0$ and $p^* = 0.5$.

5

FIGURE 11: The effect of non-dimensional cavitation pressure p^* on the dimensionless shear stress τ^+ .

4. Conclusions

A hydrodynamic lubrication model for a pocketed bearing combined with boundary slip was proposed, and it was shown that the pocketed slip bearing can give many advanced properties compared with solely textured slider bearing (without slip). The following conclusions summarize the results of the present study:

1. The influence of cavitation and consequently an increase in load support can be achieved by introducing full slip.
2. For the no-slip case of a pocket bearing, increasing the pocket depth will decrease the value of shear stress. This result is the opposite when boundary slip is applied.
3. Another interesting result is that shortening the inlet length will lead to a reduced length of the cavitation area, lower shear stress distribution and thus a reduced friction force.

Nomenclatures

a	= inlet length
b	= pocket width
B_o	= slider length
c	= outlet land length

h_d	= height of microtexture pocket (excluding film thickness)
h_o	= minimum film thickness
p_2	= pressure at B
p_3	= pressure at C
p_{atm}	= atmospheric pressure
p_{cav}	= cavitation pressure
U	= sliding velocity
q	= flow rate
x	= coordinate in sliding direction
X_b	= non-cavitated length of pocket width
z	= coordinate through film thickness
μ	= lubricant dynamic viscosity
α_s, α_h	= slip coefficient at surface s (moving) and h (stationary)

Non dimensional parameters

$$a^* = a / B_o$$

$$A = \mu \alpha / h_o$$

$$b^* = b / B_o$$

$$HD = h_d / h_o$$

$$p^+ = p h_o^2 / U \mu B_o$$

$$p^* = p / p_{atm}$$

$$x^+ = x / B_o$$

$$\tau^+ = \tau h_o / U \mu$$

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Appendix : Detail of Mathematical Model

The analysis below is based on continuity of flow through the bearing and uses the terminology as shown in Figure 1. For the condition in which the slip velocity at the stationary solid-liquid interface follows the Navier slip model (Navier, 1823. “Memoire sur les lois du mouvement des fluides”, Memoires de l’Academie Royale des Sciences de l’Institut de France 6, pp. 389-440), the corresponding flow rate equation can be expressed as by Fowell et al. [7]:

$$q_x = -\frac{\partial}{\partial x} \left(\frac{h^3}{12\mu} \frac{h^2 + 4\mu h(\alpha_h + \alpha_s) + 12\mu^2 \alpha_h \alpha_s}{h(h + \mu(\alpha_h + \alpha_s))} \frac{\partial p}{\partial x} \right) + \frac{U}{2} \frac{\partial}{\partial x} \left(\frac{h^2 + 2\mu h \alpha_h}{h + \mu(\alpha_h + \alpha_s)} \right) + \frac{h}{2\mu} \frac{\partial p}{\partial x} \frac{\partial h}{\partial x} \frac{\mu h \alpha_h + 2\mu^2 \alpha_h \alpha_s}{h + \mu(\alpha_h + \alpha_s)} - U \frac{\partial h}{\partial x} \frac{\mu \alpha_h}{h + \mu(\alpha_h + \alpha_s)} \quad (\text{A.1})$$

If slip is present only on the stationary surface ($\alpha_s = 0$ but $\alpha_h \neq 0$), Equation (A.1) will reduce to:

$$q_x = -\frac{\partial}{\partial x} \left(\frac{h^3}{12\mu} \frac{\partial p}{\partial x} \frac{h^2 + 4\mu h \alpha_h}{h(h + \mu \alpha_h)} \right) + \frac{U}{2} \frac{\partial}{\partial x} \left(\frac{h^2 + 2\mu h \alpha_h}{(h + \mu \alpha_h)} \right) \quad (\text{A.2})$$

For the first order Reynolds equation, the flow rate is given:

$$q_x = \frac{U}{2} \frac{h^2 + 2\mu h \alpha_h}{h + \mu \alpha_h} - \frac{h^3}{12\mu} \frac{h + 4\mu \alpha_h}{h + \mu \alpha_h} \frac{dp}{dx} \quad (\text{A.3})$$

A.1. No cavitation in recess

A.1.1. Pressure distribution

For the areas of the bearing containing parallel surfaces the film thickness h is constant, so the pressure gradient must be linear (see Eq. A.3). Equation (A.3) can be modified for each region of the bearing:

$$q_{AB} = \frac{U}{2} \frac{h_o^2 + 2\mu h_o \alpha_h}{h_o + \mu \alpha_h} - \frac{h_o^3}{12\mu} \frac{h_o + 4\mu \alpha_h}{h_o + \mu \alpha_h} \left(\frac{p_2 - p_{am}}{a} \right) \quad (\text{A.4})$$

$$q_{BC} = \frac{U}{2} \frac{h_p^2 + 2\mu h_p \alpha_h}{h_p + \mu \alpha_h} - \frac{h_p^3}{12\mu} \frac{h_p + 4\mu \alpha_h}{h_p + \mu \alpha_h} \left(\frac{p_3 - p_2}{b} \right) \quad (\text{A.5})$$

$$q_{CD} = \frac{U}{2} \frac{h_o^2 + 2\mu h_o \alpha_h}{h_o + \mu \alpha_h} - \frac{h_o^3}{12\mu} \frac{h_o + 4\mu \alpha_h}{h_o + \mu \alpha_h} \left(\frac{p_{am} - p_3}{c} \right) \quad (\text{A.6})$$

with one condition $q_{AB} = q_{BC} = q_{CD}$ based on volume conservation.

When q_{AB} is set as equal to q_{CD}

$$\frac{U}{2} \frac{h_o^2 + 2\mu h_o \alpha_h}{h_o + \mu \alpha_h} - \frac{h_o^3}{12\mu} \frac{h_o + 4\mu \alpha_h}{h_o + \mu \alpha_h} \left(\frac{p_2 - p_{atm}}{a} \right) = \frac{U}{2} \frac{h_o^2 + 2\mu h_o \alpha_h}{h_o + \mu \alpha_h} - \frac{h_o^3}{12\mu} \frac{h_o + 4\mu \alpha_h}{h_o + \mu \alpha_h} \left(\frac{p_{atm} - p_3}{c} \right) \quad (A.7)$$

$$\left(\frac{p_2 - p_{atm}}{a} \right) = \left(\frac{p_{atm} - p_3}{c} \right) \quad (A.8)$$

For the no-cavitation case, p_2 is unknown. Equating the flow rate in the inlet land and recess,

$$q_{AB} = q_{BC}$$

$$\frac{U}{2} \frac{h_o^2 + 2\mu h_o \alpha_h}{h_o + \mu \alpha_h} - \frac{h_o^3}{12\mu} \frac{h_o + 4\mu \alpha_h}{h_o + \mu \alpha_h} \left(\frac{p_2 - p_{atm}}{a} \right) = \frac{U}{2} \frac{h_p^2 + 2\mu h_p \alpha_h}{h_p + \mu \alpha_h} - \frac{h_p^3}{12\mu} \frac{h_p + 4\mu \alpha_h}{h_p + \mu \alpha_h} \left(\frac{p_3 - p_2}{b} \right) \quad (A.9)$$

If, $K_o = \frac{\mu \alpha_h}{h_o + \mu \alpha_h}$ and, $K_p = \frac{\mu \alpha_h}{h_p + \mu \alpha_h}$ then

$$\frac{U h_o}{2} (1 + K_o) - \frac{h_o^3}{12\mu} (1 + 3K_o) \left(\frac{p_2 - p_{atm}}{a} \right) = \frac{U h_p}{2} (1 + K_p) - \frac{h_p^3}{12\mu} (1 + 3K_p) \left(\frac{p_3 - p_2}{b} \right) \quad (A.10)$$

From Eq. (A.8), it is known that $p_3 = p_{atm} - \frac{c}{a}(p_2 - p_{atm})$ therefore

$$\left(\frac{p_3 - p_2}{b} \right) = \frac{ab + bc}{ab^2} (p_{atm} - p_2) \quad (A.11)$$

Substitution of Eq. (A.8) in Eq. (A.9) gives

$$p_2 \left[\left(h_p^3 + 3h_p^3 K_p \right) \frac{ab + bc}{ab^2} + \left(\frac{h_o^3 + 3h_o^3 K_o}{a} \right) \right] = p_{atm} \left[\left(h_p^3 + 3h_p^3 K_p \right) \frac{ab + bc}{ab^2} + \left(\frac{h_o^3 + 3h_o^3 K_o}{a} \right) \right] - 6\mu U \left[(h_p + h_p K_p) - (h_o + h_o K_o) \right] \quad (A.12)$$

if

$$C_p^* = h_p^3 + 3h_p^3 K_p$$

$$C_o^* = h_o^3 + 3h_o^3 K_o$$

$$C_p = h_p + h_p K_p$$

$$C_o = h_o + h_o K_o$$

Combining Eqs. (A.11) and (A.12) to eliminate p_3 gives

$$\left(\frac{ab+bc}{ab^2} C_p^* + \frac{C_o^*}{a} \right) = P_{atm} \left(\frac{ab+bc}{ab^2} C_p^* + \frac{C_o^*}{a} \right) - 6\mu U (C_p - C_o) \quad (\text{A.13})$$

and thus,

$$p_3 = P_{atm} + \frac{6\mu U (C_p - C_o) bc}{aC_p^* + cC_p^* + bC_o^*} \quad (\text{A.14})$$

A.1.2. Shear Stress

The shear stress at the lower surface can be derived as follows:

$$\tau = -\frac{\mu U}{h + \mu\alpha_h} - \frac{h}{2} \frac{\partial P}{\partial x} \frac{h + 2\mu\alpha_h}{h + \mu\alpha_h} \quad (\text{A.15})$$

For the areas of the bearing containing parallel surfaces the film thickness h is constant, so the shear stress must be linear (see Equation A.15). Equation (A.15) can be modified for each region of the bearing to

$$\tau_{AB} = -\frac{\mu U}{h_o + \mu\alpha_h} - \frac{h_o}{2} \left(\frac{p_2 - p_{atm}}{a} \right) \frac{h_o + 2\mu\alpha_h}{h_o + \mu\alpha_h} \quad (\text{A.16})$$

$$\tau_{BC} = -\frac{\mu U}{h_p + \mu\alpha_h} - \frac{h_p}{2} \left(\frac{p_3 - p_2}{b} \right) \frac{h_p + 2\mu\alpha_h}{h_p + \mu\alpha_h} \quad (\text{A.17})$$

$$\tau_{CD} = -\frac{\mu U}{h_o + \mu\alpha_h} - \frac{h_o}{2} \left(\frac{p_{atm} - p_3}{c} \right) \frac{h_o + 2\mu\alpha_h}{h_o + \mu\alpha_h} \quad (\text{A.18})$$

A.2. Cavitation in recess

A.2.1. Pressure distribution

For the case of cavitation in recess, p_2 is assumed to be equal to p_{cav} , so $q_{AB} = q_{BC}$ gives

$$\frac{Uh_o}{2} (1 + K_o) - \frac{h_o^3}{12\mu} (1 + 3K_o) \left(\frac{p_{cav} - p_{atm}}{a} \right) = \frac{Uh_p}{2} (1 + K_p) - \frac{h_p^3}{12\mu} (1 + 3K_p) \left(\frac{p_3 - p_{cav}}{X_b} \right) \quad (\text{A.19})$$

where

$$X_b = \frac{C_p^* (p_3 - p_{cav})}{6\mu U (C_p - C_o) - C_o^* \left(\frac{p_{atm} - p_{cav}}{a} \right)} \quad (\text{A.20})$$

When q_{AB} is set as equal to q_{CD}

$$\frac{U}{2} \frac{h_o^2 + 2\mu h_o \alpha_h}{h_o + \mu \alpha_h} - \frac{h_o^3}{12\mu} \frac{h_o + 4\mu \alpha_h}{h_o + \mu \alpha_h} \left(\frac{p_{cav} - p_{atm}}{a} \right) = \frac{U}{2} \frac{h_o^2 + 2\mu h_o \alpha_h}{h_o + \mu \alpha_h} - \frac{h_o^3}{12\mu} \frac{h_o + 4\mu \alpha_h}{h_o + \mu \alpha_h} \left(\frac{p_{atm} - p_3}{c} \right) \quad (A.21)$$

and thus,

$$p_3 = p_{atm} + \frac{c}{a} (p_{atm} - p_{cav}) \quad (A.22)$$

A.2.2. Shear Stress

The shear stress at the lower surface can be derived as follows:

$$\tau = -\frac{\mu U}{h + \mu \alpha_h} - \frac{h}{2} \frac{\partial p}{\partial x} \frac{h + 2\mu \alpha_h}{h + \mu \alpha_h} \quad (A.23)$$

Equation (A.23) can be modified for each region of the bearing to

$$\tau_{AB} = -\frac{\mu U}{h_o + \mu \alpha_h} - \frac{h_o}{2} \left(\frac{p_{cav} - p_{atm}}{a} \right) \frac{h_o + 2\mu \alpha_h}{h_o + \mu \alpha_h} \quad (A.24)$$

$$\tau_{BB} = 0 \quad (A.25)$$

$$\tau_{BC} = -\frac{\mu U}{h_p + \mu \alpha_h} - \frac{h_p}{2} \left(\frac{p_3 - p_{cav}}{X_b} \right) \frac{h_p + 2\mu \alpha_h}{h_p + \mu \alpha_h} \quad (A.26)$$

$$\tau_{CD} = -\frac{\mu U}{h_o + \mu \alpha_h} - \frac{h_o}{2} \left(\frac{p_{atm} - p_3}{c} \right) \frac{h_o + 2\mu \alpha_h}{h_o + \mu \alpha_h} \quad (A.27)$$

Paper C

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Numerical Investigation of Texturing and Wall Slip
in Lubricated Sliding Contact
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Numerical Investigation of Texturing and Wall Slip in Lubricated Sliding Contact Considering Cavitation

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Abstract Cavitation is of importance in lubricated systems with textured surfaces, therefore in the present paper a modified Reynolds equation including wall slip is combined with a cavitation model. A numerical investigation was carried out to study the interaction of wall slip and surface texturing on the lubrication of lubricated MEMS devices. Textured surfaces with different patterns were studied and compared with the results of similar untextured surfaces. The results showed that the presence of wall slip could result in a significant reduction in the coefficient of friction. It was found that slip can improve the load support because the cavitation zone is reduced. This effect was particularly large for untextured surfaces with an optimized slip surface.

Keywords: Cavitation, MEMS (micro-electro-mechanical-system), texture, wall slip.

1. Introduction

Today, the performance requirements of micro-electro-mechanical-systems (MEMS) containing moving surfaces are increasingly demanding. These devices are lubricated to reduce friction and improve their life span. This means that the design of MEMS devices must undergo an optimization process for improved performance from a tribological point of view. It is generally accepted that a low friction in combination with a high load-carrying capacity in lubricated MEMS is desired [1].

A great challenge in the development of MEMS lubrication is the problem of obtaining optimal tribological performance of their moving parts. This is because the lubricant behaviour is different on a micro or nano-scale from that on a macro-scale. On a micro or nano-scale, the boundary condition will play a significant role in determining the lubricant flow behaviour between the interacting components. Control of the boundary condition will allow a degree of verification over the hydrodynamic pressure in confined systems and is important in lubricated MEMS.

Lately, studies have proven that surface texturing can be beneficial, experimentally and/or theoretically [2-3]. Its main influence is to enhance hydrodynamic pressure locally, and it results in a higher load-carrying capacity of the lubricated contact.

Dobrica and Fillon [2] demonstrated that both the texture aspect ratio and Reynolds number had an equally significant effect on the validity of the Reynolds equation in analysing a textured contact. The authors also stated that a clear loss in the overall hydrodynamic lift is affected partially by the lubricant inertia for the textured sliding contact. More investigations on the interaction of fluid inertia with surface texture in a plain sliding contact was shown by Cupillard et al. [3]. Their results indicated that inertia effects can produce either negative or positive influence on the load-carrying capacity of a textured surface. A critical texture depth exists, below which positive inertia effects are encountered, and above which negative effects

are observed. In relation to the friction reduction, Yuan et al. [4] recently showed experimentally that the orientation of a micro-texture has a strong effect on the friction performance of a sliding contact. Experimentally, Yagi et al. [5] found that the formation of micro-texture on the surface of paper-friction materials can improve the dynamic friction coefficient by changing the groove volume ratio and without changing the pore characteristics of the paper. Numerically, Yagi et al. [6] investigated the magnitude of the load-carrying capacity enabled by textured feature in hydrodynamic lubrication based on the Reynolds equation. Two boundary conditions at the inlet and outlet were investigated and compared: an atmospheric boundary condition and a periodic boundary condition. The authors revealed that the load-carrying capacity for the atmospheric boundary condition is negative while that for the periodic boundary condition is positive when a dimple is located at the centre of the lubricated area. They also found that an atmospheric boundary condition, when a dimple is located on the inlet side and is opened to the outside, exhibits greater load-carrying capacity than that in the enclosed dimple case.

In several of the above-mentioned studies, an assumption of an ideal boundary condition was constructed, i.e. no boundary slip occurs at the interface. With the aim of explaining the wall slip of lubricated contact, several lubrication theories with slip were developed [7-10]. Spikes [7] extended the Reynolds equation including boundary slip. Salant and Fortier [8] studied the slip surface with a recess without considering cavitation. Wu et al. [9], using the finite element method, proposed a modified Reynolds equation to determine the optimal slip zone. Tauviqueirrahman et al. [10] used an optimization approach to explore the optimal slip zone, but for untextured slip zone. Only a few papers [11-13] refer to the generation of engineered slip which was combined with the advantages of

surface texturing with respect to the tribological performance. In Aurelian et al. [11], the combination of slip and surface texturing is investigated for journal bearings. Rao et al. [12] studied slider and journal bearings with slip and surface texturing using a cavitation model. Tauviqueirrahman et al. [13] investigated the combination effect of texturing and the slip on the load support. Unfortunately, in these references, the cavitation which may present is ignored.

This paper presents results of the modified Reynolds equation for parallel textured lubricated sliding contacts combined with boundary slip. The aim of the analysis is to study the impact of surface texturing and slip parameters on the hydrodynamic performance with respect to friction force reduction and load-carrying capacity considering cavitation.

2. Analysis

2.1. Modified Reynolds equation

The hydrodynamic performance of a textured contact with slip is investigated through numerical simulations. In order to analyse the boundary slip, the slip is modelled using the critical shear stress criterion [7]. The flow is assumed to be isothermal, isoviscous, and laminar. Figure 1 shows the schematic of a lubricated textured sliding contact in MEMS. The global contact geometry in this study is that of a rectangular textured sliding surface. It is assumed that the slip boundaries are engineered to all faces of the texture cells. In Fig. 1, an example of a texture cell with the grooves placed at the leading edge of the contact is presented. The lower body slides at a constant velocity U , while the upper body is stationary. Such a lubrication system can be described by solving the three-dimensional modified Reynolds equation:

$$\begin{aligned}
& \left\{ \frac{\partial}{\partial x} \left(\frac{h^3}{12\mu} \frac{\partial D}{\partial x} \frac{h^2 + 4h\mu\alpha_h}{h(h + \mu\alpha_h)} \right) + \frac{\partial}{\partial y} \left(\frac{h^3}{12\mu} \frac{\partial D}{\partial y} \frac{h^2 + 4h\mu\alpha_h}{h(h + \mu\alpha_h)} \right) \right\} F \\
&= \frac{U}{2} \frac{\partial}{\partial x} \left(\frac{h^2 + 2h\alpha_h\mu}{h + \mu\alpha_h} \right) - U \frac{\alpha_h\mu}{h + \mu\alpha_h} \frac{\partial h}{\partial x} + \frac{h}{2\mu} \frac{h\alpha_h\mu}{h + \mu\alpha_h} \left(\frac{\partial D}{\partial x} \frac{\partial h}{\partial x} + \frac{\partial D}{\partial y} \frac{\partial h}{\partial y} \right) F \\
&+ \left\{ \frac{U}{2} \frac{\partial}{\partial x} \left(\frac{D}{h} \frac{h^2 + 2h\alpha_h\mu}{h + \mu\alpha_h} \right) - \frac{D}{h} U \frac{\alpha_h\mu}{h + \mu\alpha_h} \frac{\partial h}{\partial x} \right\} (1 - F)
\end{aligned} \tag{1}$$

It should be noted that Eq. 1 is the modified Reynolds equation derived for over the whole bearing. Universal variable D and the index F are defined to anticipate the non-active (cavitation) zone, as well as the active zone.

In the active zones:

$$\begin{cases} D = p, \\ F = 1 \end{cases} \quad D \geq 0$$

In the cavitation zones:

$$\begin{cases} D = r - h, \\ F = 0 \end{cases} \quad D < 0$$

In this study, when dealing with the derivation of modified Reynolds equation the conditions for cavitation are based on the JFO (Jakobsson-Floberg-Olsson) theory to model the rupture zone for the textured surface. The methodology used with respect to the cavitation phenomena is similar to the work of Gherca et al. [14] and is explained in detail in the Appendix.

It can be seen that if the slip length α is set to zero (no-slip condition), Eq. 1 reduces to the classical Reynolds equation. In the present study, the slip length is normalized by the land film thickness h_F , see Fig. 1, resulting in the dimensionless slip length A . In the following simulations, slip coefficient α is $0.02 \text{ m}^2/\text{s}/\text{kg}$ (the corresponding slip length is $1 \times 10^{-4} \text{ m}$) based on the results published in the literature [15-16].

2.2. Geometry and boundary conditions

The main texture parameters are the relative texture zone (T_R/L_x) and the texture cell aspect ratio ($\lambda = L_D/h_D$), while the slip parameter is the

slip length β , which is defined as an extrapolated distance relative to the wall where the tangential velocity component vanishes. Other parameters are the relative texture depth HD (where $HD = h_D/h_F$) and texture density ρ (where $\rho = L_D/L_C$). However, for all following computations, the parameters ρ and T_R/L_x are fixed at 0.5 and 0.7 respectively. The characteristics of the lubricated sliding contact are given in Table 1.

2.3. Numerical method and solution

Equation 1 was solved numerically using the finite volume method. The system of equations was solved using the tridiagonal matrix algorithm (TDMA) [17]. A mesh number of about 8,000 – 15,000 nodes for the computational domain was utilized depending on the texture number. It should be noted that the meshing process for textured surfaces has been checked to ensure grid independent results. In the present work, the Navier slip model is adopted [18].

TABLE 1: Characteristics of main bearing analysed.

Parameters	Data Setting	Unit
Film thickness h_F	10^{-6}	m
Contact length in x-direction L_x	10^{-3}	m
Contact length in y-direction L_y	10^{-3}	m
Velocity U	1	m/s
Viscosity μ	10^{-3}	Pa.s

(a) Isometric view.

(b) Longitudinal view.

FIGURE 1: Schematic representation of a lubricated parallel sliding contact with textured surface combined with boundary slip in the textured region (in x -direction).

3. Results and discussions

3.1. Significance of the cavitation model

In this section, the implementation of the cavitation model in the lubrication bearing is discussed, i.e. the results of slider bearing with and without cavitation are compared. Figure 2 depicts a lubricated textured contact in which cavitation may occur. In this section, the positive effect of the slip will be shown as well.

Figures 3 and 4 show the calculated hydrodynamic pressures for two situations: (1) without slip but with the JFO cavitation model, (2) with boundary slip and the JFO cavitation model. Based on Figure 3, when the JFO cavitation model is considered, the hydrodynamic pressure becomes more realistic based on the mass conservation. It is interesting to note that when boundary slip is introduced as

shown in Figure 4, the cavitation zone disappears and this area produces more hydrodynamic pressure. This means that the slip has a significant effect on pressure generation and as a result on load-carrying capacity. For the numerical results in Figures 3 and 4, the main parameters considered in the calculations are as follows: slip-texturing zone $T_R = 7 \times 10^{-4}$ m, the slip coefficient $\alpha = 0.02$ m²/s/kg, pocket length $L_D = 1.4 \times 10^{-4}$ m, the pocket depth $h_D = 5 \times 10^{-7}$ m, the land film thickness $h_F = 1 \times 10^{-6}$ m, as shown in Figure 2. The detail algorithm for solving the modified Reynolds equation with JFO model can be seen in the Appendix.

(a) Isometric view.

(b) Longitudinal view.

FIGURE 2: Schematic representation of textured contact in which cavitation may occur (in x -direction).

As explained in [2] to minimize the effect of cavitation pressure, for a textured bearing the inlet should start with a dam, whilst for a bearing with boundary slip the slip should be present in the leading edge of the contact as

explained in [10]. In this paper attention is paid to bearings with optimal slip.

FIGURE 3: Three-dimensional hydrodynamic pressure distribution for a textured bearing with an inlet that starts with no-dam. No-slip and with JFO cavitation model. $HD = 0.5$ and $N = 2$.

FIGURE 4: Three-dimensional hydrodynamic pressure distribution for a textured bearing with an inlet that starts with no-dam. Boundary slip is present and the JFO cavitation model is applied and omitted. $HD = 0.5$ and $N = 2$.

3.2. Effect of groove depth

In the present study, it is convenient to present the simulation results with the dimensionless parameters (see Nomenclature). The first investigation deals with the effect of groove depth. As is known, for the classical lubricated situation i.e. no-slip condition parallel sliding surfaces are not able to produce any load-carrying capacity. Therefore, the groove depth is of interest.

(a) Solely textured surface.

(b) Combined textured slip surface.

FIGURE 5: Effect of the dimensionless groove depth HD on the dimensionless load-carrying capacity W varying the number of grooves N , note the difference in scales for W . (Note: $T_R/L_x = 0.7$ and $\rho = 0.5$)

Figure 5 shows the effect of the dimensionless groove depth HD on the dimensionless load-carrying capacity W and varying the number of grooves N . It can be seen in Fig. 5 (a) that for parallel sliding surfaces, when there is no texturing (i.e. $HD = 0$), the load-carrying capacity W is zero. This result is different in the case of combined texturing and slip in which a high load-carrying capacity is generated. This is because, even though there are no grooves in the inlet of the lubricated contact (i.e. smooth surface), the slip at a certain area of the stationary surface with an optimized pattern is able to create hydrodynamic pressure (and thus load-carrying capacity), even in the absence of the wedge (see Fig. 5b).

Based on Fig. 5, one can also say that, in the case of a solely textured surface, increasing the

groove depth will increase the load-carrying capacity. For a groove number N which is lower than 10, it is interesting to note that the increase in W is limited. There is an optimum value for the groove depth after exceeding this value; the W will decrease but not significantly. For a higher groove number (i.e. $N > 14$), such a trend cannot be identified, i.e. an optimum value of HD is not noted.

In relation to the effect of boundary slip, Fig. 5 also shows that for the case of combined slip and texturing, there is an interesting result. Boundary slip has proven that it is able to generate more hydrodynamic pressure (Fig. 6) and thus load-carrying capacity, even with uniform film thickness. For all groove numbers, W is approximately 1.05 when $HD = 0$. By increasing the groove depth, W increases and then decreases after $HD = 0.2$. From Fig. 5, the increase in W is not very significant ($< 1\%$). After exceeding the optimum value of HD , the W decreases. This means that in an untextured surface (smooth condition), as long as the surface is engineered based on an optimized slip surface, the W is high and this is a desired situation.

FIGURE 6: Three-dimensional hydrodynamic pressure for solely textured pattern (without slip) with $HD = 0.5$ and $N = 2$.

In addition to the load-carrying capacity, the coefficient of friction (COF) is investigated. This is because, in MEMS, by lubrication, a minimum coefficient of friction means that the viscous friction force is low and is very beneficial with respect to future high-speed sliding MEMS. In this study, the coefficient of

friction (COF) is determined by normalizing the friction force f to the load-carrying capacity w .

(a) Solely textured surface.

(b) Combined textured slip surface.

FIGURE 7: Effect of the dimensionless groove depth HD on the dimensionless coefficient of friction COF for two groove numbers N (note the difference in scales for the COF).

Figure 7 shows the effect of the groove depth HD on the coefficient of friction for two values of the groove number N . It is noted that the profiles presented in Fig. 7 are selected for a high and a low groove number. It is shown that adding boundary slip will decrease the COF significantly in comparison with the solely textured pattern. Compared with the case of pure texturing, the COF can decrease by a factor 20. Based on Fig. 7, with respect to the effect of the dimensionless groove depth, there is a different trend of the COF for the case of pure texturing and for the combined texturing pattern with slip. For the former, the increase in groove depth has a positive effect on reducing the COF , while for the latter, increasing HD will increase the COF especially for the case of low N . For the case of solely textured configuration, the rapid decrease

in COF occurs when the groove depth is increased to 0.5. For HD which is greater than this value, the COF is virtually insensitive to the groove depth. This trend prevails for both a low and a high groove number. A contradictory result is found when wall slip is combined with the textured surfaces. It can be seen in Fig. 7 that, for all values of HD and N , the predicted COF is higher than that without texturing ($HD = 0$).

FIGURE 8: Effect of the dimensionless groove depth HD and groove number N on the dimensionless friction force F_f for the slip and no-slip situation.

Figure 8 shows the effect of the dimensionless groove depth HD and groove number N on the dimensionless friction force F_f for the slip and no-slip situation. The dimensionless friction force shown in Fig. 8 is nearly constant for both cases (slip and no-slip texturing). An interesting aspect to note, with respect to the friction force, is that the dimensionless friction force is independent of the groove number in the case of both a solely textured pattern and a combined textured slip configuration. This result matches well with the one-dimensional analytical work of Pascovisi et al. [19] from a pure texturing point of view. As mentioned earlier, it indicates that an untextured (smooth surface) with slip is recommended to be applied in real engineering applications with respect to the coefficient of friction.

3.3. Effect of groove number

Figure 9 shows the dimensionless load-carrying capacity W as a function of the groove number N for several values of dimensionless groove depth HD . It can be seen that for both cases (i.e. solely textured pattern and combined textured slip configuration), the trend of W is slightly similar.

Increasing the groove number decreases W initially and then increases for all values of groove depth. This results in the existence of a minimum value for W for a certain HD . The inflection of this point differs with the groove number. For example, for pure texturing, using a high groove number the inflection point occurs when N is approximately 6, but for HD which is less than 0.4 there is a shift of the inflection point towards the right-hand side of the graph. This indicates that for a surface which is "rough" it is necessary to increase the groove number in order to improve the load-carrying capacity. This is, however, not the case for a smooth surface (i.e. small HD). From Fig. 9, it is also observed that the inflection of W in the case of the combined textured slippage pattern is relatively small. The predicted results tend to be linear, especially for a low groove number. For high HD , the increase in W after N reaches the inflection point is clearly visible. It is interesting to note that this result is similar to the case of pure texturing. The difference lies in the inflection point, which refers to a smaller groove number. As a concluding remark, for the textured pattern both with and without wall slip, increasing the groove number will be beneficial to improving the load-carrying capacity. It is worth mentioning that when $N = 1$, the bearing configuration is similar to that of the Rayleigh step bearing.

(a) Textured surface.

(b) Combined textured slip surface.

FIGURE 9: Effect of the groove number N on the dimensionless load carrying capacity W for several values of dimensionless groove depth HD (Note: inset of Fig. 6 (a) the effect of the groove number N for the range of 0 to 2 on the dimensionless load carrying capacity W).

4. Conclusions

In the present study, the effect of both boundary slip and surface texturing in a lubricated parallel sliding contact were analysed. A modified Reynolds equation that takes slip and cavitation into account was presented and solved. The results obtained can be summarized as follows:

1. The presence of wall slip results in a reduction of the cavitation zone.
2. Under parallel-gap conditions, an effective load-carrying capacity of the lubricant film can be obtained by either a solely textured surface (without slip) or a combined textured surface with boundary slip configuration.
3. In terms of the friction coefficient, an untextured slip surface is beneficial with respect to the hydrodynamic performance. In addition, slip has a positive effect to improve the load support.

Nomenclatures

COF	= dimensionless coefficient of friction
F	= friction force
L_x	= length of lubricated surface in the x-direction
L_y	= length of lubricated surface in the y-direction
p	= fluid film pressure
U	= velocity

w	= load carrying capacity,
α	= slip coefficient
α_h	= slip coefficient at surface h (stationary surface)
β	= slip length
μ	= dynamic viscosity
λ	= texture aspect ratio
ρ	= texture density
h	= film thickness
h_i	= inlet film thickness
h_D	= groove depth
h_F	= land film thickness
L_D	= groove length
L_C	= texture cell length
T_R	= texturing zone

Non-dimensional parameters

$$A = \beta / h_F$$

$$H_D = h_D / h_F$$

$$F_f = fh_F / (U \mu L_x L_y)$$

$$L_x^* = x / L_x$$

$$L_y^* = y / L_y$$

$$P = ph_F^2 / (U \mu L_x)$$

$$W = wh_F^2 / (U \mu L_x L_y)$$

$$COF = f / w$$

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Appendix A:

The modified Reynolds equation (Equation 1) is derived for an incompressible lubricant with constant viscosity from a simple form of the Navier Stokes equation, ignoring the inertia in the film.

$$\begin{aligned}\frac{\partial p}{\partial x} &= \frac{\partial}{\partial z} \left(\mu \frac{\partial u}{\partial z} \right) \\ \frac{\partial p}{\partial y} &= \frac{\partial}{\partial z} \left(\mu \frac{\partial v}{\partial z} \right) \\ \frac{\partial p}{\partial z} &= 0\end{aligned}\tag{A.1}$$

In order to obtain the velocity distribution by integration of Equation 1, it is necessary to define the surface boundary conditions. The surface boundary conditions are proposed as:

$$\begin{aligned}\text{at } z=0, u &= U & ; v &= 0 \\ \text{at } z=h, u &= -\alpha_h \mu \frac{\partial u}{\partial z} \Big|_{z=h} & ; v &= -\alpha_h \mu \frac{\partial v}{\partial z} \Big|_{z=h}\end{aligned}\tag{A.2}$$

The expression of the fluid velocity in the x -direction, subject to the above boundary conditions, is:

$$u = \frac{1}{2\mu} \frac{\partial p}{\partial x} z^2 - \left(\frac{U}{h + \mu\alpha_h} + \frac{h}{2\mu} \frac{\partial p}{\partial x} \frac{h + 2\alpha_h \mu}{h + \mu\alpha_h} \right) z + U \frac{h + \alpha_h \mu}{h + \mu\alpha_h}\tag{A.3}$$

A similar expression can be written for the fluid velocity in the y -direction. Assuming the fluid density is constant, the conservation of mass requires

$$\int_0^h \frac{\partial u}{\partial x} dz + \int_0^h \frac{\partial v}{\partial y} dz + \int_0^h \frac{\partial w}{\partial z} dz = 0\tag{A.4}$$

Deriving each term of above equation (Eq. a.4), the modified Reynolds equation obtained is:

$$\begin{aligned}\frac{\partial}{\partial x} \left(\frac{h^3}{12\mu} \frac{\partial p}{\partial x} \frac{h^2 + 4h\mu\alpha_h}{h(h + \mu\alpha_h)} \right) + \frac{\partial}{\partial y} \left(\frac{h^3}{12\mu} \frac{\partial p}{\partial y} \frac{h^2 + 4h\mu\alpha_h}{h(h + \mu\alpha_h)} \right) &= \frac{U}{2} \frac{\partial}{\partial x} \left(\frac{h^2 + 2h\alpha_h \mu}{h + \mu\alpha_h} \right) \\ -U \frac{\alpha_h \mu}{h + \mu\alpha_h} \frac{\partial h}{\partial x} + \frac{h}{2\mu} \frac{h\alpha_h \mu}{h + \mu\alpha_h} \left(\frac{\partial p}{\partial x} \frac{\partial h}{\partial x} + \frac{\partial p}{\partial y} \frac{\partial h}{\partial y} \right)\end{aligned}\tag{A.5}$$

Equation A.5 can be applied to the active and non-active zone. Thus, Equation A.5. is modified as follows:

$$\begin{aligned}
& \left\{ \frac{\partial}{\partial x} \left(\frac{h^3}{12\mu} \frac{\partial D}{\partial x} \frac{h^2 + 4h\mu\alpha_h}{h(h + \mu\alpha_h)} \right) + \frac{\partial}{\partial y} \left(\frac{h^3}{12\mu} \frac{\partial D}{\partial y} \frac{h^2 + 4h\mu\alpha_h}{h(h + \mu\alpha_h)} \right) \right\} F = \\
& \frac{U}{2} \frac{\partial}{\partial x} \left(\frac{h^2 + 2h\alpha_h\mu}{h + \mu\alpha_h} \right) - U \frac{\alpha_h\mu}{h + \mu\alpha_h} \frac{\partial h}{\partial x} + \frac{h}{2\mu} \frac{h\alpha_h\mu}{h + \mu\alpha_h} \left(\frac{\partial D}{\partial x} \frac{\partial h}{\partial x} + \frac{\partial D}{\partial y} \frac{\partial h}{\partial y} \right) F + \\
& \left\{ \frac{U}{2} \frac{\partial}{\partial x} \left(\frac{D}{h} \frac{h^2 + 2h\alpha_h\mu}{h + \mu\alpha_h} \right) - \frac{D}{h} U \frac{\alpha_h\mu}{h + \mu\alpha_h} \frac{\partial h}{\partial x} \right\} (1 - F)
\end{aligned} \tag{A.6}$$

In the active zones:

$$\begin{cases} D = p, \\ F = 1 \end{cases} \quad D \geq 0$$

$$\begin{aligned}
& \frac{\partial}{\partial x} \left(\frac{h^3}{12\mu} \frac{\partial p}{\partial x} \frac{h^2 + 4h\mu\alpha_h}{h(h + \mu\alpha_h)} \right) + \frac{\partial}{\partial y} \left(\frac{h^3}{12\mu} \frac{\partial p}{\partial y} \frac{h^2 + 4h\mu\alpha_h}{h(h + \mu\alpha_h)} \right) = \\
& \frac{U}{2} \frac{\partial}{\partial x} \left(\frac{h^2 + 2h\alpha_h\mu}{h + \mu\alpha_h} \right) - U \frac{\alpha_h\mu}{h + \mu\alpha_h} \frac{\partial h}{\partial x} + \frac{h}{2\mu} \frac{h\alpha_h\mu}{h + \mu\alpha_h} \left(\frac{\partial p}{\partial x} \frac{\partial h}{\partial x} + \frac{\partial p}{\partial y} \frac{\partial h}{\partial y} \right)
\end{aligned} \tag{A.7}$$

In the cavitation zones:

$$\begin{aligned}
& \frac{U}{2} \frac{\partial}{\partial x} \left(\frac{h^2 + 2h\alpha_h\mu}{h + \mu(\alpha_h + \alpha_s)} \right) - U \frac{\alpha_h\mu}{h + \mu\alpha_h} \frac{\partial h}{\partial x} + \frac{U}{2} \frac{\partial}{\partial x} \left(\frac{r - h}{h} \frac{h^2 + 2h\alpha_h\mu}{h + \mu\alpha_h} \right) \\
& - \frac{r - h}{h} U \frac{\alpha_h\mu}{h + \mu(\alpha_h + \alpha_s)} \frac{\partial h}{\partial x} = 0
\end{aligned} \tag{A.8}$$

where: $D = r - h = h(\bar{\rho} - 1)$; $r = \bar{\rho}h$; $\bar{\rho} = \frac{\rho}{\rho_o}$, ρ is the density of lubricant-gas mixture and ρ_o the density of the lubricant.

$$\frac{U}{2} \frac{\partial}{\partial x} \left(\bar{\rho} \frac{h^2 + 2h\mu\alpha_h}{h + \mu(\alpha_h + \alpha_s)} \right) - U \frac{\bar{\rho}\alpha_h\mu}{h + \mu(\alpha_h + \alpha_s)} \frac{\partial h}{\partial x} = 0 \tag{A.9}$$

In order to simultaneously solve Equations (A.5) - (A.9) over the whole domain of the bearing, a universal variable D is defined and a unified modified Reynolds equation is obtained as stated in Equation 1, section 2.

Paper D

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Analysis of the Effect of the Slip Pocket in Single and Double Parallel Bearing Considering Cavitation: A Theoretical Approach

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Abstract In this paper, based on an analytical approach, the effect of pockets and boundary slip on the hydrodynamic performance of parallel sliding surfaces considering cavitation is investigated. A modified Reynolds theory is developed for solving two kinds of bearings: a single and a double pocket bearing. The performance is compared with respect to the variation of the pocket depth, pocket length, slip, and no-slip situation. The results show that the maximum pressure and load support increases with the reduction in pocket length. The main finding is that the pocket depth reduces the cavitation area. However, in the case of a single pocket, the role of pocket depth is more significant in reducing the cavitation effect than that in the case of a double pocket bearing.

Keywords: cavitation, hydrodynamic lubrication, pocket, slip boundary, theoretical model.

1. Introduction

In past decades, boundary slip has gained significant momentum thanks to researchers. Slip can be induced by (super) hydrophobic surfaces, as demonstrated by [1,2]. Recently, boundary slip has been engineered to improve bearing performance. Slip can occur on the surface if the surface shear stress achieves its critical shear stress [3].

Introducing a pocket as well as boundary slip at introducing pocket and boundary slip at a bearing surface is becoming popular in oil-

lubricated systems because it has proven to improve the performance in load support and friction. Experimentally, it has been shown that surface texturing improves the load support for mechanical systems with parallel sliding contacts [4,5], cylinder liner contacts [6] and mechanical seals [7]. With the introduction of boundary slip, high load support was highlighted by some works [8–15]. Recently, in addition to surface texturing, the use of an artificial slip surface is also of interest with respect to lubrication. The earliest work on artificial complex slip/no-slip for lubricated parallel sliding devices was reported by Salant and Fortier [16]. They focused on the ability of artificial slip to improve the load support and to reduce the friction force in the absence of the wedge effect. Subsequently, several studies were published confirming the findings of Salant and Fortier [16]. Guo and Wong [17] confirmed that the introduction of so-called tailored boundary slip on the stationary surface of a slider leads to a net pressure build-up. Similarly, Wu et al. [8] studied the behaviour of a slider bearing with mixed slip surface conditions, and their results indicated that convergent, parallel, and even a divergent wedge could provide hydrodynamic load support. Rao [18] investigated the pressure and the shear stress distribution of the single-groove journal and slider bearing. Extended Reynolds equations with slip situations were proposed. Tauviquirrahman et al. [19–22] explored the potential application of slip as well as texturing in order to enhance the tribological performance behaviour. Their findings indicated that the combined slip/texture bearing improves the load support and friction. However, in their

works, particularly in [19–21], the cavitation treatment was questionable due to the non-mass-conserving cavitation approach.

Theoretical analysis of a textured surface has been carried out using the Reynolds equation [23–26] and CFD (Computational Fluid Dynamics) approaches based on Navier–Stokes equations [27–32]. Two texture modes were used, namely micro and macro roughness [33,34]. The first mode is based on a single-cell texture configuration [9,24,27,29,32,35,36] and the second one is based on a multi-cell texture pattern [23,25,31,35]. For the two approaches and texturing modes mentioned, cavitation was not considered, so their results may be questionable.

Following the insight on the significance of cavitation, interesting results have been explored by some researchers, for example Fowell et al. [24], Gherca et al. [37], Mao et al. [38], and Galda et al. [39]. On the basis of the Reynolds theory, Fowell et al. [24] investigated in more detail the effect of the mechanism of inlet suction on the performance of textured bearings with pockets to reduce friction. Recent interesting work regarding the effect of texturing was performed by Gherca et al. [37]. Using finite element modelling, they presented the results of hydrodynamic thrust bearings operating in both the steady-state and the transient lubrication regime. It was found that the performance of the bearing is not the same when texturing is applied to the rotor and the stator. Mao et al. [38] proposed an analytical model coupled with the wedge effect for predicting cavitation characteristics under sliding motion. Optimization was also performed to improve the performance of lubricated textured components. Galda et al. [39] found, based on an experimental approach, that dimples improved the tribological characteristics under starved lubrication conditions at low sliding speeds. It was also found that the beneficial effect of dimples was more significant at a lower load. Later, Kasem et al. [40] studied experimentally the effect of low depth surface on friction reduction. They found that dimples with smaller diameter lead to a significant decrease in the coefficient of friction in comparison with the dimples with a larger diameter.

Based on the previously published works, the analysis was conducted for a one-dimensional parallel sliding contact, ignoring the cavitation effect. Wall slip, surface texturing, and cavitation have a significant effect on the bearing's characteristics. Bayada and Meurisse [12] showed the importance of the cavitation model in the analysis of slip/no-slip hydrodynamic contacts. Fortier et al. [41] presented the numerical analysis of a journal bearing using a modified slip length model and mass-conserving algorithm. The results illustrated high load support and low friction using a heterogeneous slip/no-slip bearing surface. Based on the modified slip length model and cavitation model, Wang and Lu [42] presented results for a two-dimensional sleeve bearing.

Additionally, the effect of boundary slip and surface texturing was also studied, for example [8,16,19,20,43–48], but ignoring the cavitation effect and mostly using a non-mass-conservation cavitation equation. In the present work, it will be shown that cavitation has a strong influence on performance and unrealistic conclusions may be drawn if the cavitation is ignored. A few publications considered texturing and slip in lubricated contacts considering cavitation [42]. Another interesting study dealing with slip combined with texturing was reported by Biancofiore et al. [49]. In the presence of cavitation, the authors proposed a stable numerical model that can be used to study the slip-lubricated contact. The guidelines to improve and optimize the design of textured superoleophobic surfaces have been highlighted.

The novel contribution of the present work is to develop a new lubrication model including mass-conserving cavitation based on an extended Reynolds approach. In this paper, the texturing configurations of single- and double-grooved pockets are of particular interest. The correlation between the texturing parameters with slip and cavitation on the load-carrying capacity is the main concern.

2. Mathematical model

The textured bearing configurations considered in this paper are shown in Figs. 1 and 2. Figs. 1 and 2 represent a single and double pocket

slider slip bearing. Here, the slip condition is assumed to occur on the top edge of the pocket. The slip condition is indicated by the rollers, as shown in Figs. 1 and 2. It is noted that for the case of a double pocket, the first pocket depth is set to be lower than the second pocket depth. In this work, the pocket depth is varied for the first pocket, while the pocket depth of the second pocket is kept constant. For calculations, the inlet and outlet pressures are set to atmospheric pressure. The slip condition is represented by the slip coefficient, α of 0.02 m²/s/kg, while for the sliding velocity, U , it is set to 1 m/s.

In practice, the boundary slip can be induced by modifying the roughness and/or wettability. From the physical point of view, the slip is observed as the condition in which the velocity of the fluid at the interface is not equal to surface velocity. This is different from the no-

slip condition. In this way, the classical assumption stated that the fluid velocity at the interface has the same value as the surface velocity. To solve the lubrication problem, an analytical equation is proposed by extending the Reynolds theory, including the cavitation and the slip. The methodology used in the derivation of the extended Reynolds equation is quite similar to that employed by Fowell et al. [24] and is given in detail in Appendix A.

Based on Figs. 1 and 2, the mathematical flow rate can be derived. For flow rate, a set of governing equations can be obtained (by using the pressure continuity at the boundaries Eqs. A.1–A.24, see Appendix A). Solving the set of governing equations, the analytical solution for the single pocket with slip is given in Eq. 1 and for the double pocket with slip in Eqs. 2 and 3.

$$h_p^3 \frac{h_p + 4\mu\alpha_h}{h_p + \mu\alpha_h} \left(\frac{p_3 - p_{cav}}{X_b} \right) = 6\mu U \left(h_p \frac{h_p + 2\mu\alpha_h}{h_p + \mu\alpha_h} - h_o \right) - h_o^3 \left(\frac{p_{atm} - p_{cav}}{a} \right) \quad (1)$$

$$h_{p1}^3 \frac{h_{p1} + 4\mu\alpha_h}{h_{p1} + \mu\alpha_h} \left(\frac{p_3 - p_{cav}}{X_b} \right) = 6\mu U \left(h_{p1} \frac{h_{p1} + 2\mu\alpha_h}{h_{p1} + \mu\alpha_h} - h_o \right) - h_o^3 \left(\frac{p_{cav} - p_3}{c} \right) \quad (2)$$

$$h_{p2}^3 \frac{h_{p2} + 4\mu\alpha_h}{h_{p2} + \mu\alpha_h} \left(\frac{p_5 - p_{cav}}{X_d} \right) = 6\mu U \left(h_{p2} \frac{h_{p2} + 2\mu\alpha_h}{h_{p2} + \mu\alpha_h} - h_o \right) - h_o^3 \left(\frac{p_{atm} - p_5}{e} \right) \quad (3)$$

FIGURE 1: Schematic of the parallel sliding single pocket bearing considering cavitation.

FIGURE 2: Schematic of the parallel sliding double pocket bearing considering cavitation.

3. Methodology

In the present work, the occurrence of cavitation is assumed to take place at zero pressure. The cavitation area in the first pocket is denoted by C_{Lb} in which $C_{Lb} = b - X_b$ and the cavitation area in the second pocket is defined by C_{Ld} in which $C_{Ld} = d - X_d$. This means that when the cavitation occurs, C_L is larger than zero.

The solution for the lubrication problem for a pocketed slider bearing (single and double pocket) is as follows:

1. Single pocket. For all computations, the cavitation phenomenon is assumed to occur in the pocket. Here, the cavitation equations (Eqs. A.1–A.11 of Appendix A) are used to obtain X_b (non-cavitated region in the pocket as seen in Fig. 1) X_b is then used as an

indicator of whether cavitation occurs or not. Cavitation in a pocket occurs if $0 < X_b < b$, where b is the pocket length.

2. Double pocket. For all computations, the cavitation phenomenon is assumed to occur in the first and second pockets of the bearing. The cavitation equations (Eqs. A.12–A.24, see Appendix A) to obtain X_b and X_d (a non-cavitated region in the first and second pockets as indicated in Fig. 2) are used. X_b and X_d will then be used as an indicator of whether cavitation occurs or not. It should be noted that cavitation in the first pocket occurs if $0 < X_b < b$, where b is the first pocket length and cavitation in second pocket occurs if $0 < X_d < d$, where d is the second pocket length.

The validity of the Reynolds equation in lubricated contacts with surface texturing has

been discussed in the literature. The limits of this equation are given in [31]. In this study, care has been taken to ensure that the configurations considered here fall within the validity domain of the Reynolds equation for the case of both single and double pocket texturing. As a note, the model developed here (Eqs. 1-3) has been validated, as shown in Appendix B.

4. Results and discussions

In this section, the effect of texturing and cavitation on a single-pocketed bearing and on a double-pocketed bearing is studied. It should be noted that for the last case, only the first pocket is of main interest. It means that the variation of the texturing parameters is applied to the first pocket. The cavitation effect on the lubrication of the second pocket is also explored to analyze the correlation of the lubrication characteristics between the first and second pockets.

4.1. Case of single-pocketed bearing

Table 1 depicts the dimension of the single-pocket bearing considered here. As a note, the pocket depth and length are of particular interest in this research. As seen in Table 1, when the pocket length is varied, based on Fig. 1, the exit land length changes following the constraint of the total length.

TABLE 1: Characteristics of a single pocket bearing.

Parameters	Data Setting	Unit
Pocket depth h_d	0.1; 0.5; 1.0 and 2.0	μm
Land film thickness h_o	10^{-6}	m
Total length B_o	0.02	m
Inlet length a	0.002	m
Pocket length b	0.0015; 0.006	m
Exit land length e	0.0165; 0.012	m
Atmospheric pressure P_{atm}	100	kPa
Cavitation pressure P_{cav}	0	kPa
Sliding velocity U	1	m/s
Slip coefficient α	0.02	$\text{m}^2\text{s/kg}$
Lubrication viscosity μ	0.01	Pa.s

4.1.1 Effect of pocket depth with low pocket length

Figure 3 depicts the comparison of the pressure distribution of the single pocketed bearing between the slip case and the no slip case varying the pocket depth. In this section, the

parameter HD is introduced as the normalized pocket depth. HD is defined as the ratio of pocket depth, h_d , over the land film thickness, h_o . Furthermore, the load support, w , is of particular interest. Here, w is defined as the integration of the pressure over the surface area.

In this discussion, the slip case refers to the pocketed bearing in which slip is allowed on the top edge of the pocket (Fig. 1) by setting the slip coefficient as $\alpha = 0.02 \text{ m}^2\text{s/kg}$. It should be noted that the slip coefficient chosen is the optimal value with respect to the maximum load support based on published works, both experimental [10,11] and numerical [12].

FIGURE 3: The pressure distribution p^* as a function of coordinate x^* in the sliding direction for the pocketed bearing for $h_o = 1 \times 10^{-6} \text{ m}$, $a = 0.002 \text{ m}$, $b = 0.0015 \text{ m}$, $e = 0.0165 \text{ m}$, $Bo = 0.02 \text{ m}$, and $\alpha = 0.02 \text{ m}^2\text{s/kg}$ for the case.

Several specific features can be observed based on Fig. 3. Firstly, for the single pocketed bearing for both the slip case and no slip case, the pressure distribution shows similar trends for all pocket depth values studied. At the inlet, the pressure decreases to cavitation pressure for all configurations. This indicates that if the pocket is not placed at the inlet length, the suction mechanism will, as discussed by Bayada et al. [12] dominate the bearing even if boundary slip is employed. Secondly, it is easy to observe that cavitation occurs in the bearing for both cases (slip and no slip). This is characterized by the presence of the cavitation area as marked in Fig. 3. In the inset of Fig. 3 regarding the slip case, it can be observed that a high pocket depth, HD , leads to a decrease in the cavitation area. However, for the no slip case, a low HD behaves differently.

In detail, dealing with the pocket depth, it can be seen that in the case of a pocketed bearing with slip the pocket depth will affect the cavitation area, C_{Lb} , considerably. This trend also prevails for the no-slip case. Fig. 4 reflects the effect of pocket depth on the cavitation area for both slip and no-slip pocketed cases. For the slip case, increasing the pocket depth will decrease the C_{Lb} considerably. It is very interesting to note that cavitation exists when $HD = 0$, which is associated with the smooth parallel sliding surface. This result matches well with published work [12].

From a design and manufacturing point of view, to have the condition of no slip in the inlet is not recommended because it leads to cavitation, which may reduce the lubrication performance of the sliding contact: for the no-slip case, the increase in HD will produce an increase then decrease profile of the cavitation area, C_{Lb} . This indicates that there will be an optimum value for HD after which an increase in HD will reduce the C_{Lb} significantly. Below this optimal HD , a decrease in HD will also reduce the C_{Lb} considerably. This is a contrary result to that of the slip case. Based on Fig. 4, it is seen that the critical HD is around 0.5. Analyzing Fig. 4, boundary slip reduces C_{Lb} more than that for no slip (up to 86.57%). However, this condition prevails when HD is larger than the critical HD . For example, for $HD = 4$, compared with the no-slip bearing, slip gives a 36.40% lower value in C_{Lb} , and if this result is associated with the load support W^* , the improvement in W^* by slip is around 2.47% larger.

Thirdly, employing boundary slip on the pocket does not enhance the peak dimensionless pressure very much. For both cases (slip and no slip), the peak hydrodynamic pressure refers to the same value, i.e. $p^* 0.0046$. However, if the slip effect is subjected to the lubrication performance (load support in this case), the slip has a positive effect. The predicted load support for all cases considered in Fig. 1 is summarized in Table 2. Based on Table 2, for all pocket depths considered here, the load support predicted by a pocketed bearing with slip has a higher value than that without slip. From a physical point of view, this is because the cavitation area is reduced by employing the slip,

as mentioned earlier, and thus the integration of pressure over the surface area becomes larger.

FIGURE 4: The cavitation area as a function of pocket depth and sliding direction of the pocketed bearing for $h_o = 1 \times 10^{-6}$ m, $a = 0.002$ m, $b = 0.0015$ m, $e = 0.0165$ m, $Bo = 0.02$ m, and $\alpha = 0.02$ m²/s/kg.

TABLE 2: Dimensionless load support in single pocket bearing with $b = 0.0015$ m.

HD	Dimensionless load support, $W^* (10^{-3})$		Change in load support (%)
	Slip	No-slip	
0.1	1.647	1.663	-0.97
0.5	1.651	1.651	0
1.0	1.658	1.653	0.30
2.0	1.677	1.663	0.84

4.1.2 Effect of pocket depth with high pocket length

Figure 5 shows the comparison of the pressure distribution of the single pocketed bearing between the slip case and the no-slip case varying the pocket depth with high pocket length, $b = 0.006$ m. In general, the pressure distribution is similar to the pressure profile for low b , as depicted in Fig. 3. However, some specific features can be highlighted from Fig. 5.

Firstly, it can be stated that a higher b , results in a lower value for the maximum pressure. The most likely explanation is that increasing the pocket length will reduce the exit land length, while the maximum pressure, P_3 , is a function of inlet and exit length (see Eq. A.8 in the Appendix A). In this case of high b (300% higher than the first case, i.e., low b), P_3^* reaches 0.0035 (or 31.43% lower). Secondly, the value of the maximum pressure, P_3 , is not affected by varying the pocket depth, HD . As mentioned earlier, the pocket length has a

significant role in influencing the cavitation area.

FIGURE 5: The pressure distribution p^* as a function of coordinate x^* in the sliding direction for the pocketed bearing for $h_o = 1 \times 10^{-6}$ m, $a = 0.002$ m, $b = 0.006$ m, $e = 0.012$ m, $Bo = 0.02$ m, and $\alpha = 0.02$ m²/s/kg for the case.

Figure 6 depicts the length of the cavitation area for the slip and no-slip condition. In the no-slip condition, the optimum cavitation area, C_{Lb}^* , is 0.2975 for $HD = 0.5$. The cavitation area decreases with increasing HD . For the slip case, increasing the pocket depth will decrease C_{Lb} . It is very interesting to note that cavitation exists when $HD = 0$, which is associated with a smooth (non-textured) parallel sliding surface. Fig. 6 shows that the cavitation area has the same value for pocket depth, i.e., HD equal to 0.5, while for the case of the slip and no-slip condition, if $HD = 4$, the difference in cavitation area is around 5%. Therefore, it can be said that the introduction of the cavitation area is reduced.

Table 3 shows the load support for the slip and no-slip condition with variations of pocket depth, HD . For the slip condition, the load support increases with increasing pocket depth, HD , while the maximum load support W^* is 0.000747 if the pocket depth $HD = 2.0$. For the no-slip condition, the minimum load support W^* is 0.000732 for a pocket depth, HD , of 0.05. The maximum load support W^* is 0.000739 when the pocket depth HD is 1.0 or 2.0. The result is similar to the research by Bayada et al. [12] for the case in which the slip is not placed on the inlet length, i.e., load support increased significantly. From the table, it can be seen that when pocket depth $HD = 0.5$, the slip and no-slip conditions generate the same load support. It

can be said that HD becomes optimal in the case of the no slip when HD is set to 0.5.

FIGURE 6: The cavitation area as a function of pocket depth and sliding direction of the pocketed bearing for $h_o = 1 \times 10^{-6}$ m, $a = 0.002$ m, $b = 0.006$ m, $e = 0.012$ m, $Bo = 0.02$ m, and $\alpha = 0.02$ m²/s/kg.

TABLE 3: Dimensionless load support in single pocket bearing with $b = 0.006$ m.

HD	Dimensionless load support, W^* (10^{-3})		Change in load support (%)
	Slip	No-slip	
0.1	0.730	0.739	-1.23
0.5	0.732	0.732	0
1.0	0.736	0.733	0.41
2.0	0.747	0.739	1.07

4.1.3 Effect of pocket length, b

Figure 7 shows the pressure distribution for the slip and no-slip condition with pocket depth $HD = 1$, pocket length $b = 0.0015$ and 0.006 m. The pressure distribution for the slip and no slip condition shows a similar trend. The pressure distribution decreases to the cavitation pressure, then increases to maximum pressure P_3 , and decreases toward the atmosphere pressure at the exit land length. The maximum pressure, P_3 , is of particular interest, it decreases with increasing pocket length. From the Fig. 7, it can be seen that the maximum pressure distribution, P_3 , for different pocket lengths, b , has the same gradient; thus, minimizing the pocket length will increase the maximum pressure, P_3 , in the cavitation condition. However, it should be noted that there is a threshold in this increase in P_3 as a function of a decreasing pocket length, b . When cavitation does not occur for a specific pocket pattern, in particular, with a very low pocket length, the maximum pressure may reduce. For a more detailed discussion about this

threshold, the reader is referred to reference [49]. The slip condition will decrease the cavitation area. However, the reduction in cavitation area is not significant. The slip condition will not significantly affect the maximum pressure distribution, P_3 , for the same pocket length.

FIGURE 7: The pressure distribution as a function of coordinate in sliding direction for the single pocketed bearing for $h_d = 1 \times 10^{-6}$ m, $h_o = 1 \times 10^{-6}$ m, $a = 0.002$ m, $B_o = 0.02$ m, and $\alpha = 0.02$ m²/s/kg.

Figure 8 gives the cavitation area as a function of pocket depth. The cavitation area increases with increasing pocket length. However, the cavitation area shows a similar trend with the pocket depth for the slip and the no slip condition. Fig. 9 depicts the no cavitation area, X_b , as a function of pocket depth. For the slip condition, the no cavitation area increases with increasing pocket depth, HD . However, it decreases with increasing pocket length, b . In other words, the load support will increase for a lower pocket length, b , and a higher pocket depth, HD . In the no-slip condition, a minimum X_b occurs when the pocket length, HD , is equal to approximately 0.5. For $HD > 0.5$, X_b increases with increasing pocket depth, HD . As for the slip condition, on increasing the pocket length, b , for the no slip condition X_b will decrease. From Fig. 9, it can be seen that the slip condition improves the no-cavitation area. The result is consistent with the results mentioned earlier and the work of Bayada et al. [12], i.e., slip will reduce the cavitation area.

FIGURE 8: The cavitation area as a function of pocket depth and sliding direction of the single pocketed bearing for $h_o = 1 \times 10^{-6}$ m, $a = 0.002$ m, $B_o = 0.02$ m, and $\alpha = 0.02$ m²/s/kg.

FIGURE 9: The no cavitation area as a function of pocket depth sliding direction of the single pocketed bearing for $h_o = 1 \times 10^{-6}$ m, $a = 0.002$ m, $B_o = 0.02$ m, and $\alpha = 0.02$ m²/s/kg.

4.2. Case of double pocketed bearing

Based on Fig. 2, the main interest of this work is the effect of pocket depth as well as the pocket length of the first pocket on the lubrication characteristics of parallel sliding surfaces. It is noted that for the second pocket, the dimensions are assumed to be constant. The slip is applied at the top of the pocket for both the first and second pocket. Varying the pocket length for the first pocket will change the exit land length, while the pocket length of the second pocket is constant. Table 4 shows the characteristics of the double-pocket bearing.

TABLE 4: Characteristics of a double pocket bearing.

Parameters	Data Setting	Unit
First pocket depth h_{d1}	0.1; 0.5; 1.0 and 2.0	μm
Land film thickness h_o	10^{-6}	m
Total length B_o	0.02	m
Inlet length a	0.002	m
Pocket length b	0.0015; 0.006	m
Length between pocket c	0.002	m
Second pocket length d	0.003	m
Exit land length e	0.0115; 0.007	m
Atmospheric pressure P_{atm}	100	kPa
Cavitation pressure P_{cav}	0	kPa
Sliding velocity U	1	m/s
Slip coefficient α	0.02	m^2/kg
Lubrication viscosity μ	0.01	$\text{Pa}\cdot\text{s}$

4.2.1 Effect of first pocket depth

Figure 10 shows the pressure distribution of the double pocket bearing (Fig. 2) for both the slip and the no-slip condition. The figure describes some findings: firstly, the pressure distribution shows a similar trend for both the slip and the no-slip condition. The maximum pressure, P_5 , occurs in the second pocket, the value of the maximum pressure is the same for the slip or no-slip condition ($P_5^* = 0.0034$). The maximum pressure in the first pocket is equal to the atmosphere pressure. Secondly, the pressure distribution in the first or second pocket is not influenced by the first pocket depth, HD_1 . The variation of the first pocket depth, HD_1 , will influence the length of the cavitation area in the first pocket (see inset of Fig. 10). However, it does not affect the cavitation area in the second pocket (see Eq. A.24 in Appendix A). Consequently, the length of the cavitation area is not a function of the first pocket depth, HD_1 . Thirdly, the slip condition will affect the cavitation area in the first or second pocket, in other words it reduces the cavitation area.

FIGURE 10: The pressure distribution as a function of coordinate in sliding direction for the double-pocketed bearing for $h_{d2} = 1 \times 10^{-6}$ m, $h_o = 1 \times 10^{-6}$ m, $a = 0.002$ m, $b = 0.0015$ m, $c = 0.002$ m, $d = 0.003$ m, $e = 0.0115$ m, $Bo = 0.02$ m, and $\alpha = 0.02$ m^2/kg .

Figure 11 depicts the length of the cavitation area in the first pocket as a function of pocket depth. The length of the cavitation area is similar to that of the single-pocketed bearing for both the slip and the no-slip situation. The maximum pocket length (C_{Lb}) occurs when pocket depth $HD_1 = 0.5$, that is 0.0746. The cavitation area decreases as a function of pocket depth, HD_1 . In the slip condition, the length of the cavitation area decreases with increasing pocket depth, HD_1 . If the pocket depth, HD_1 , is equal to zero, the length of the cavitation area still remains large. However, for the no-slip condition, a contrary result is observed; the length of the cavitation area decreases significantly, which means that the cavitation will vanish if the pocket depth is reduced to up nearly the smooth condition.

Table 5 gives the value of the load support of the double pocket bearing with pocket length $b = 0.0015$ m. The load support for the slip condition increases with increasing pocket depth, HD_1 . The maximum dimensionless load support for the slip situation is 0.0006746 when the pocket depth, $HD_1 = 2.0$. Contrary to the slip condition, the no-slip condition has a minimum load support for $HD_1 = 0.5$. The maximum load support for the no-slip condition is 0.0006719 for $HD_1 = 0.1$ and 2.0. The difference in maximum load support between the slip and no-slip condition is 0.40% for $HD_1 = 2.0$. The result is similar to the research of Bayada et al. [12]: there is a small increase in load support if slip is

not provided in the inlet suction. The load support for the slip condition is higher than that for the no slip situation. In the second pocket, the length of the cavitation area for the no slip condition ($C_{Ld}^* = 0.00291$) is larger than for the slip situation ($C_{Ld}^* = 0.00288$).

FIGURE 11: The cavitation area as a function of the pocket depth of the first pocketed bearing for $h_{d2} = 1 \times 10^{-6}$ m, $h_o = 1 \times 10^{-6}$ m, $a = 0.002$ m, $b = 0.0015$ m, $c = 0.002$ m, $d = 0.003$ m, $e = 0.0115$ m, $B_o = 0.02$ m, and $\alpha = 0.02$ m²/s/kg.

TABLE 5: Dimensionless load support of double pocket bearing with $b = 0.0015$ m.

HD_1	Dimensionless load support, W^* (10^{-3})		Change in load support (%)
	Slip	No-slip	
0.1	0.6742	0.6719	0.35
0.5	0.6743	0.6718	0.37
1.0	0.6744	0.6718	0.38
2.0	0.6746	0.6719	0.40

4.2.2 Effect of first pocket length

Figure 12 shows the pressure distribution as a function of coordinate in sliding direction for the double pocket bearing with a first pocket length of 0.006 m. The pressure distribution shows a similar trend for both the slip and the no slip condition. The maximum pressure in the first pocket, P_3 , is equal to the atmosphere pressure. The second pocket, P_5 , is the same for the slip and the no-slip situation ($P_5^* = 0.0023$). A different pocket depth, HD_1 , affects the length of cavitation in the first pocket for the slip and the no slip situation (see insert Fig. 12). However, it does not influence the pressure distribution in the second pocket.

Figure 13 depicts the cavitation area in the first pocket as a function of pocket depth, HD_1 . For

the slip situation, the cavitation area decreases as a function of the pocket depth. However, the cavitation area has a maximum value for the no-slip situation ($HD_1 = 0.5$) and decreases with increasing pocket depth, HD_1 . The difference in the cavitation area for the slip and no-slip condition is around 0.67% for $HD_1 = 4.0$. The slip condition affects the cavitation area in the first and second pocket. However, the first pocket depth does not influence the cavitation area in the second pocket. The length of the cavitation area in the second pocket C_{Ld} is 0.004 for the slip situation and 0.003 for the no-slip condition. As a consequence, this will affect the load support.

Table 6 shows the load support of the double pocket bearing with pocket length $b = 0.006$ m. The load support increases with increasing first pocket depth, HD_1 , for the slip situation. The maximum load support W^* is 0.0000362 for $HD_1 = 2.0$. For the no slip situation, the maximum load support is 0.0000349 in $HD_1 = 2.0$. The difference in maximum load support is 2.56% for the slip and the no-slip situation. The load support for the slip situation is higher than for the no-slip condition. The second cavitation area influences the load support. The cavitation area, C_{Ld} , is 0.00292 for the no-slip situation and is equal to that for the slip condition (see Fig. 12).

FIGURE 12: The pressure distribution as a function of coordinate in sliding direction for the pocketed bearing for $h_{d2} = 1 \times 10^{-6}$ m, $h_o = 1 \times 10^{-6}$ m, $a = 0.002$ m, $b = 0.006$ m, $c = 0.002$ m, $d = 0.003$ m, $e = 0.007$ m, $B_o = 0.02$ m, and $\alpha = 0.02$ m²/s/kg.

FIGURE 13: The cavitation area in the first pocket as a function of pocket depth of the first pocketed bearing for $h_{d2} = 1 \times 10^{-6}$ m, $h_o = 1 \times 10^{-6}$ m, $a = 0.002$ m, $b = 0.006$ m, $c = 0.002$ m, $d = 0.003$ m, $e = 0.007$ m, $B_o = 0.02$ m, and $\alpha = 0.02$ m²/kg.

TABLE 6: Dimensionless load support in double pocket bearing with $b = 0.006$ m.

HD	Dimensionless load support, $W^* (10^{-3})$		Change in load support (%)
	Slip	No-slip	
0.1	0.0358	0.0349	2.60
0.5	0.0359	0.0348	3.12
1.0	0.0360	0.0348	3.27
2.0	0.0362	0.0349	3.56

4.2.3 Comparison of pocket length in double texture

Figure 14 shows a comparison of the pressure distribution as a function of pocket length for the slip and no-slip situation. Reducing the first pocket length improves the maximum pressure distribution, P_5 . The increasing pressure distribution has the same slope gradient. Consequently, the maximum pressure can be predicted as long as the cavitation condition exists. It can be observed that the maximum pressure distribution of the first pocket, P_3 , is equal to the atmospheric pressure.

FIGURE 14: The pressure distribution as a function of the coordinate in sliding direction for the double pocketed bearing for $h_{d1} = 1 \times 10^{-6}$ m, $h_{d2} = 1 \times 10^{-6}$ m, $h_o = 1 \times 10^{-6}$ m, $a = 0.002$ m, $c = 0.002$ m, $d = 0.003$ m, $B_o = 0.02$ m, and $\alpha = 0.02$ m²/kg.

Figure 15 reflects the no cavitation area of the first pocket, X_b , as a function of the first pocket depth for the slip and no slip situation. The figure also depicts that the no cavitation area of the first pocket is not affected by the first pocket length. Slip and pocket depth improve X_b . For the slip situation, X_b increases with increasing pocket depth. For very small HD_1 , the no cavitation area is close to zero. However, the no-slip situation has a minimum X_b for $HD_1 = 0.5$, and it increases with increasing HD_1 . For $HD_1 = 0.0$ (i.e., smooth surface), X_d does not exist. It can be observed that the slip trends for varying b coincide with each other. These trends also occur for the no slip condition.

Figure 16 shows the comparison of the no-cavitation area, X_b , between the single and double pocket bearing for the slip and no slip situation, respectively. It can be observed that X_b in the case of the double pocket is not affected by pocket length for both the slip and the no-slip condition. In other words, such a pocket pattern gives the same X_b profile. Another interesting result is that X_b for the case of the double pocket bearing is smaller than that of the single pocket bearing for both slip and no-slip conditions. The no cavitation area increases with larger first pocket length, HD_1 , and is close to zero for small HD_1 . It can be observed that the no-slip trends for varying b coincide with each other as shown in Fig. 16.

FIGURE 15: The no cavitation area in the first pocket as a function of pocket depth of the double pocketed bearing for $h_{d1} = 1 \times 10^{-6}$ m, $h_{d2} = 1 \times 10^{-6}$ m, $h_o = 1 \times 10^{-6}$ m, $a = 0.002$ m, $c = 0.002$ m, $d = 0.003$ m, $B_o = 0.02$ m, and $\alpha = 0.02$ m²/s/kg.

FIGURE 16: The no cavitation area as a function of pocket depth of the single and double pocketed bearing for $h_{d1} = 1 \times 10^{-6}$ m, $h_{d2} = 1 \times 10^{-6}$ m, $h_o = 1 \times 10^{-6}$ m, $a = 0.002$ m, $c = 0.002$ m, $d = 0.003$ m, $B_o = 0.02$ m, and $\alpha = 0.02$ m²/s/kg.

5. Conclusions

In the present work, the mechanism of the inlet suction using single and double pockets of a lubricated contact (with and without slip) considering cavitation condition was analyzed. A one dimensional analytical model has been developed for both a single and a double pocketed bearing with boundary slip, including cavitation. The following conclusions summarize the results of the present study:

1. The pocket depth has a strong effect on pressure and load support. For the slip condition, increasing the pocket depth increases the load support. However, for the no-slip condition, the optimal pocket depth is observed; when this value is exceeded, the reduced load support is found.

2. When the slip condition is applied on the top edge of the pocket, the slip effect is not as significant as in the no slip condition. The load support improvement is no more than 3% both for single pocket and double pocket, irrespective of the pocket length.
3. The maximum pressure and load support increase with reducing pocket length.
4. The slip condition in the pocket reduces the cavitation area both for the single and double pocket bearing. This slip effect is more pronounced when a high pocket depth is used.
5. The decrease in pocket length enhances the load support and reduces the cavitation area both for the slip and no slip conditions irrespective of the pocket depth.
6. The main interesting finding is that the pocket depth reduces the cavitation area for both the single and double pocket bearing. However, the pocket depth has a more important influence in reducing the effect of cavitation in the case of the single pocket than in the case of the double pocket bearing.

For future research, the experiment will be the main concern to validate the developed lubrication model in terms of the cavitation area and the effect of hydrophobic coating layout.

Nomenclatures

a	= inlet length
b	= first pocket width
B_o	= slider length
c	= length between pockets
C_L	= length of cavitation area
C_{Lb}	= length of cavitation area first pocket
C_{Ld}	= length of cavitation area second pocket
d	= second pocket width
e	= outlet land length
h_d	= height of microtexture pocket (excluding film thickness)
h_{d1}	= height of microtexture first pocket (excluding film thickness)
h_{d2}	= height of microtexture second pocket (excluding film thickness)
h_p	= film thickness in pocket
h_{p1}	= film thickness in first pocket
h_{p2}	= film thickness in second pocket
h_o	= minimum film thickness
P_3	= pressure at C
P_5	= pressure at E
p_{atm}	= atmospheric pressure

p_{cav} = cavitation pressure
 U = sliding velocity
 W = load support
 q = flow rate
 x = coordinate in sliding direction
 X_b = non-cavitated fraction of first pocket
 X_d = non-cavitated fraction of second pocket
 z = coordinate through film thickness
 μ = lubricant dynamic viscosity
 α_s, α_h = slip coefficient at surface s (moving) and h (stationary)

Non dimensional parameters

$$C_{Lb}^* = (b - X_b) / B_o$$

$$C_{Ld}^* = (d - X_d) / B_o$$

$$HD = h_{d1} / h_o$$

$$HD_1 = h_{d1} / h_o$$

$$HD_2 = h_{d2} / h_o$$

$$p^* = ph_o^2 / (U \mu B_o)$$

$$x^* = x / B_o$$

$$X_b^* = X_b / B_o$$

$$X_d^* = X_d / B_o$$

$$W^* = wh_o^2 / (U \mu B_o^2)$$

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Appendix A

In this section, the details of the modified Reynolds equation are presented. Extending the work of Fowell et al. [24], the corresponding flow rate equation can be expressed as

$$q_x = - \left(\frac{h^3}{12\mu} \frac{h^2 + 4\mu h(\alpha_h + \alpha_s) + 12\mu^2 \alpha_h \alpha_s}{h(h + \mu(\alpha_h + \alpha_s))} \frac{\partial p}{\partial x} \right) + \frac{U}{2} \left(\frac{h^2 + 2\mu h \alpha_h}{h + \mu(\alpha_h + \alpha_s)} \right) + \frac{h}{2\mu} \frac{\partial p}{\partial x} \frac{\partial h}{\partial x} \frac{\mu h \alpha_h + 2\mu^2 \alpha_h \alpha_s}{h + \mu(\alpha_h + \alpha_s)} - U \frac{\partial h}{\partial x} \frac{\mu \alpha_h}{h + \mu(\alpha_h + \alpha_s)} \quad (\text{A.1})$$

If slip is present only on the stationary surface ($\alpha_s = 0$ but $\alpha_h \neq 0$), Equation (A.1) will reduce to:

$$q_x = - \left(\frac{h^3}{12\mu} \frac{\partial p}{\partial x} \frac{h^2 + 4\mu h \alpha_h}{h(h + \mu \alpha_h)} \right) + \frac{U}{2} \left(\frac{h^2 + 2\mu h \alpha_h}{(h + \mu \alpha_h)} \right) \quad (\text{A.2})$$

For the first-order Reynolds equation, the flow rate is given:

$$q_x = \frac{U}{2} \frac{h^2 + 2\mu h \alpha_h}{h + \mu \alpha_h} - \frac{h^3}{12\mu} \frac{h + 4\mu \alpha_h}{h + \mu \alpha_h} \frac{dp}{dx} \quad (\text{A.3})$$

A.1 Slip in single pocket

Cavitation in single pocket

Pressure distribution

For the areas of the bearing including the parallel surfaces, the film thickness, h , is set constant, and therefore the pressure gradient must be linear (see Equation A.3). Equation (A.3) can be modified for each region of the bearing:

$$q_{AB} = \frac{U}{2} h_o - \frac{h_o^3}{12\mu} \left(\frac{p_{cav} - p_{atm}}{a} \right) \quad (\text{A.4})$$

$$q_{BC} = \frac{U}{2} \frac{h_p^2 + 2\mu h_p \alpha_h}{h_p + \mu \alpha_h} - \frac{h_p^3}{12\mu} \frac{h_p + 4\mu \alpha_h}{h_p + \mu \alpha_h} \left(\frac{p_3 - p_{cav}}{X_b} \right) \quad (\text{A.5})$$

$$q_{CD} = \frac{U}{2} h_o - \frac{h_o^3}{12\mu} \left(\frac{p_{atm} - p_3}{c} \right) \quad (\text{A.6})$$

with the condition $q_{AB} = q_{BC} = q_{CD}$ based on volume conservation.

When q_{AB} is set to be equal to q_{CD}

$$\frac{U}{2} h_o - \frac{h_o^3}{12\mu} \left(\frac{p_{cav} - p_{atm}}{a} \right) = \frac{U}{2} h_o - \frac{h_o^3}{12\mu} \left(\frac{p_{atm} - p_3}{c} \right) \quad (\text{A.7})$$

$$p_3 = p_{atm} + \frac{c}{a} (p_{atm} - p_{cav}) \quad (\text{A.8})$$

For the case of cavitation in the recess, p_2 is assumed to be equal to p_{cav} , so $q_{AB} = q_{BC}$ gives

$$\frac{U}{2} h_o - \frac{h_o^3}{12\mu} \left(\frac{p_{cav} - p_{atm}}{a} \right) = \frac{U}{2} \frac{h_p^2 + 2\mu h_p \alpha_h}{h_p + \mu \alpha_h} - \frac{h_p^3}{12\mu} \frac{h_p + 4\mu \alpha_h}{h_p + \mu \alpha_h} \left(\frac{p_3 - p_{cav}}{X_b} \right) \quad (\text{A.9})$$

For

$$K_p = \frac{\mu \alpha_h}{h_p + \mu \alpha_h}$$

$$K_{p1} = \frac{\mu\alpha_h}{h_{p1} + \mu\alpha_h}$$

$$K_{p2} = \frac{\mu\alpha_h}{h_{p2} + \mu\alpha_h}$$

$$C_p = h_p + h_p K_p$$

$$C_{p1} = h_{p1} + h_{p1} K_{p1}$$

$$C_{p2} = h_{p2} + h_{p2} K_{p2}$$

$$C_p^* = h_p^3 + 3h_p^3 K_p$$

$$C_{p1}^* = h_{p1}^3 + 3h_{p1}^3 K_{p1}$$

$$C_{p2}^* = h_{p2}^3 + 3h_{p2}^3 K_{p2}$$

Equation (A.9) becomes

$$\frac{U}{2} h_o - \frac{h_o^3}{12\mu} \left(\frac{p_{cav} - p_{atm}}{a} \right) = \frac{U}{2} C_p - \frac{C_p^*}{12\mu} \left(\frac{p_3 - p_{cav}}{X_b} \right) \quad (\text{A.10})$$

where

$$X_b = \frac{C_p^* (p_3 - p_{cav})}{6\mu U (C_p - h_o) - h_o^3 \left(\frac{p_{atm} - p_{cav}}{a} \right)} \quad (\text{A.11})$$

A.2 Slip in double pockets

Cavitation in double pocket

Pressure distribution

$$q_{AB} = \frac{U}{2} h_o - \frac{h_o^3}{12\mu} \left(\frac{p_{cav} - p_{atm}}{a} \right) \quad (\text{A.12})$$

$$q_{BC} = \frac{U}{2} C_{p1} - \frac{C_{p1}^*}{12\mu} \left(\frac{p_3 - p_{cav}}{X_b} \right) \quad (\text{A.13})$$

$$q_{CD} = \frac{U}{2} h_o - \frac{h_o^3}{12\mu} \left(\frac{p_{cav} - p_3}{c} \right) \quad (\text{A.14})$$

$$q_{DE} = \frac{U}{2} C_{p2} - \frac{C_{p2}^*}{12\mu} \left(\frac{p_5 - p_{cav}}{X_d} \right) \quad (A.15)$$

$$q_{EF} = \frac{U}{2} h_o - \frac{h_o^3}{12\mu} \left(\frac{p_{atm} - p_5}{e} \right) \quad (A.16)$$

with the condition $q_{AB} = q_{BC} = q_{CD} = q_{DE} = q_{EF}$ based on volume conservation.

q_{AB} is set to be equal to q_{BC}

$$\left(\frac{p_3 - p_{cav}}{X_b} \right) = \frac{6\mu U}{C_{p1}^*} (C_{p1} - h_o) - \frac{h_o^3}{C_{p1}^*} \left(\frac{p_{cav} - p_{atm}}{a} \right) \quad (A.17)$$

q_{BC} is set to be equal to q_{CD}

$$\left(\frac{p_3 - p_{cav}}{X_b} \right) = \frac{6\mu U}{C_{p1}^*} (C_{p1} - h_o) - \frac{h_o^3}{C_{p1}^*} \left(\frac{p_{cav} - p_3}{c} \right) \quad (A.18)$$

Substitution of Equation (A.17) in Equation (A.18) gives

$$p_3 = p_{cav} + \frac{c}{a} (p_{atm} - p_{cav}) \quad (A.19)$$

$$X_b = \frac{c C_{p1}^* (p_{atm} - p_{cav})}{6\mu U a (C_{p1} - h_o) - h_o^3 (p_{atm} - p_{cav})} \quad (A.20)$$

q_{CD} is set to be equal to q_{DE}

$$\left(\frac{p_5 - p_{cav}}{X_d} \right) = \frac{6\mu U}{C_{p2}^*} (C_{p2} - h_o) - \frac{h_o^3}{C_{p2}^*} \left(\frac{p_{cav} - p_3}{c} \right) \quad (A.21)$$

q_{DE} is set to be equal to q_{EF}

$$\left(\frac{p_5 - p_{cav}}{X_d} \right) = \frac{6\mu U}{C_{p2}^*} (C_{p2} - h_o) - \frac{h_o^3}{C_{p2}^*} \left(\frac{p_{atm} - p_5}{e} \right) \quad (A.22)$$

Substituting Equation (A.21) in Equation (A.22) gives

$$p_5 = p_{atm} + \frac{e}{a} (p_{atm} - p_{cav}) \quad (A.23)$$

$$X_d = \frac{(a+e) C_{p2}^* (p_{atm} - p_{cav})}{6\mu U a (C_{p2} - h_o) - h_o^3 (p_{atm} - p_{cav})} \quad (A.24)$$

Appendix B

In this section, the validation of our model has been presented for the case of a single pocket. All dimensions and boundary conditions have been set to those used in the work of Fowell et al. [24]. The no slip boundary condition is of particular interest by considering the cavitation. Figure B1 presents the load support prediction varying the inlet length for two values of pocket depth, $h_d = 2.5$ and $7.5 \mu\text{m}$. It can be observed that the predictions match well with the reference [24]. Our developed analytical lubrication model is very accurate with zero relative error in comparison with the analytical lubrication model of Fowell et al. [24].

FIGURE B1: The effect of inlet length on the load support. The profiles are calculated for $h_o = 1.0 \times 10^{-6} \text{ m}$, $b = 0.006 \text{ m}$, $B_o = 0.02 \text{ m}$, $\alpha = 0.02 \text{ m}^2/\text{s/kg}$.

Paper E

M. Muchammad, M. Tauviquirrahman, F. Hilmy, J. Jamari, D.J. Schipper
Effect of a Hydrophobic Surface on the Load Carrying
Capacity of a Double Pocket Lubricated
Plan Parallel Sliding Contact
To be submitted: Tribology – Materials, Surfaces & Interfaces.

Effect of a Hydrophobic Surface on the Load Carrying Capacity of a Double Pocket Lubricated Plan Parallel Sliding Contact

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Abstract This paper presents a theoretical investigation of the lubrication mechanism with boundary slip induced by a hydrophobic coating. The double pocketed lubricated sliding contact with or without a hydrophobic coating is of particular interest. The first-order Reynolds equation is modified to include boundary slip and cavitation. The results indicate that introducing a hydrophobic coating in the inlet suction prevents the occurrence of cavitation in the two pockets. It is also found that applying a hydrophobic coating in the leading edge of the two pockets gives two effects, namely minimizing the cavitation and improving the load support. Further, increasing the slip area in each inlet of a texture cell gives greater load support.

Keywords: Cavitation, hydrophobic, lubrication, slip, texture.

1. Introduction

It is interesting to study the cavitation phenomenon in lubricated bearings because cavitation is often disadvantageous. Numerical calculations that ignore cavitation may give questionable results. Cavitation cannot be avoided in the diverging area in a journal bearing or textured areas in bearings [1-3]. Recently, researchers focused on how to reduce the occurrence of cavitation [4-12]. Fowell et al. [11] discussed the cavitation phenomena in single pocket bearings. They showed also that the inlet suction mechanism in textured hydrodynamic bearings that has micropockets on its surface provides enhanced load support and

reduced friction. Later, Muchammad et al. [4] studied the cavitation phenomenon in a slider pocket bearing with boundary slip. The results showed that the cavitation pressure has a huge impact on the performance characteristics both for the boundary slip pocketed bearing and the no-slip pocketed one.

As is known, texturing on the bearing surfaces may lead to the occurrence of cavitation. Surface texturing is developed to obtain good lubrication performance, i.e. increased load support and decreasing friction. Surface texturing has proven to be beneficial experimentally for slider bearings [13-14], mechanical seals [15] and automotive components [16]. The effect of surface texturing on the tribological performance have been analysed analytically and numerically by many researchers [1-2,4-5,7-11,17-28]. Numerically, there are two approaches to investigating surface texturing: using Reynolds [17-26] and using CFD models based on Navier Stokes [27-29]. Sahlin et al. [27] confirmed that by using CFD the introduction of a micro-groove on the stationary surface of a parallel slider leads to a net pressure build-up and therefore hydrodynamic load support. In their research it was found that the effect of inertia on lubrication was dominant. Similarly, Han et al. [29] reported a three-dimensional CFD analysis of a spherical dimple on the stationary surface of a parallel sliding contact. An optimum micro dimple (depth and width) is defined that gives the greatest load support and smallest coefficient of friction. It should be pointed out that these analyses [2,4,11] focused only on single texture cells with centred dimples. In the case of multi textures, Kligerman et al. [17] studied the case

of the contact between a piston ring and cylinder liner by applying multiple texture cells. Using a theoretical model based on a modified Reynolds equation, it was demonstrated that partial texturing has an advantage over full texturing in terms of friction..

In several of the above-mentioned studies, an assumption of an ideal boundary condition was constructed, i.e. no boundary slip occurs at the interface. With the aim of explaining the wall slip of lubricated contacts, several lubrication theories with slip were developed [30-35]. Salant and Fortier [33] studied the slip surface with a recess without considering cavitation. Wu et al. [30], using the finite element method, proposed a modified Reynolds equation to determine the optimal slip zone. Tauviquirrahman et al. [23-25] explored by an optimization approach the optimal slip zone for untextured surfaces. Only a few papers [2,4-5,23-26] refer to the generation of engineered slip combined with the advantages of surface texturing with respect to tribological performance. Rao et al. [2] studied slider and the journal bearings with boundary slip and surface texturing using a cavitation model. Muchammad et al. [4] investigated the combination effect of texturing and slip on the load support and friction including cavitation. Tauviquirrahman et al. [23] explored the possibility of slip and texture in order to improve the performance characteristics. Their results indicate that the combined texture/slip pattern increase the load-carrying capacity and decrease friction.

This paper proposes a theoretical lubrication equation for parallel slider bearings with a double pocket for the no-slip and boundary slip condition including cavitation. With the purpose of maximizing the load support, boundary slip is placed in the inlet and the edge between the two pockets. The pressure distribution, load support

and cavitation area are of particular interest with the respect to boundary slip.

2. Mathematical model

The textured bearing configuration is shown in Fig. 1. The textured surface is composed of a double recess. It is hypothesized that the textured bearing with double recess will generate more “inlet suction” phenomena than the bearing with one recess. The bearing contains the no-slip condition on most of surfaces except the leading edge of the contact as well as the edge between pockets. On these edges, the boundary slip is applied. In hydrodynamic lubrication, the governing equations are described by the well-known Reynolds equation, which is a simplification of the Navier-Stokes equation. Thus, in the present work, an analytical equation based on the first-order Reynolds theory is derived. In this way, the Reynolds equation is extended by considering boundary slip and the possible occurrence of cavitation. The methodology used in the derivation of the extended Reynolds equation at the bearing surface is quite similar to that employed by Fowell [11] and is given in detail in the Appendix.

For the double pocketed bearing with no-slip as illustrated in Fig. 1a, the expressions of the pressure and flow rate can be obtained. Applying pressure continuity at boundary points, a set of governing equations can be derived for the flow rate and for constants of integration in the pressure expressions, see the Appendix for details. In the Appendix, the derivation of the flow rate and pressure, which are the basis of this analysis, are presented in Eqs. A.1-A.21. Solving the set of governing equations and employing the quantities given in the nomenclature, the analytical solution for the no-slip case is:

$$\left(\frac{p_3 - p_{cav}}{X_b} \right) = \frac{6\mu U}{h_p^3} (h_p - h_o) - \frac{h_o^3}{h_p^3} \left(\frac{p_{cav} - p_{atm}}{a} \right) \quad (1)$$

$$\left(\frac{p_5 - p_{cav}}{X_d} \right) = \frac{6\mu U}{h_p^3} (h_p - h_o) + \frac{h_o^3}{h_p^3} \left(\frac{p_{cav} - p_3}{c} \right) \quad (2)$$

For the analysis of the double pocketed bearing with no-slip condition, the pockets are situated at the inlet of the stationary surface (see Fig. 1). The no-cavitation area in first pocket X_b , no-cavitation area in second pocket X_d , pressure distribution p and load support W are of particular interest. The derivation of pressure distribution and load support with either cavitation or no-cavitation condition is given in the Appendix, i.e. Eqs. A.6-A.21. For all the following simulations, the standard one-dimensional bearing geometry is investigated, with values for the parameter as presented in Table 1.

TABLE 1: Characteristics of double pocket bearing analysed.

Parameters	Data Setting	Unit
Land film thickness h_o	1.0	μm
Total length B_o	0.02	m
Hight of microtexture	0.02	m
pocket h_d		
Inlet length a	0.002; 0.003; 0.004	m
First pocket length b	0.0015; 0.0025; 0.0035	m
Length between pocket c	0.002; 0.003; 0.004	m
Second pocket length d	0.0015; 0.0025; 0.0035	m
Atmospheric pressure P_{atm}	100	kPa
Cavitation pressure P_{cav}		
Sliding velocity U	0	kPa
Slip coefficient α	1	m/s
Lubrication viscosity μ	0.02 0.01	$\text{m}^2\text{s/kg}$ Pa.s

In the present study, solving the lubrication problem for a pocketed slider bearing from numerical point of view, the solution method is proposed as follows:

1. For all computations, the cavitation phenomenon is assumed to occur in the bearing. Here, the cavitation equations (Eqs. A.6 and A.13 in the Appendix) to obtain X_b , X_d (non-cavitated region in first and second pockets as seen in Figure 1) are used. Then, the X_b and X_d will be used as an indicator whether cavitation occurs or not. It should be noted that the cavitation occurs in first pocket if $0 < X_b < b$ where b is first pocket length and the cavitation in second pocket if $0 < X_d < d$ where d is second pocket length.
2. If cavitation is present in the pocketed bearing, then other cavitation equations (Equation A.9. and A.13. in the Appendix) are implemented to predict the pressure as well as the load support.

(a) No-slip condition.

FIGURE 1: Schematic of parallel sliding double pocketed bearing considering (a) No-slip condition (b) Slip at inlet (c) Slip at inlet and between pockets.

3. Results and discussion

As discussed by Tauviquirrahman, et al. [4, 23-25] the existence of boundary slip and texturing with respect to the hydrodynamic lubrication has a significant effect in a textured bearing. In this paper the combined effects of texture and wall slip (inlet length and length between pockets) as well as cavitation will be explored in more detail. The main characteristics of the bearing are given in Table 1 and the details of slip boundaries can be seen in Figure 1.

3.1. The Variation of dimensionless inlet length A

Figure 2 shows the effect of the dimensionless inlet length A on the dimensionless pressure distribution of parallel sliding pocket bearings.

In the no-slip condition, the variation of dimensionless inlet length does not have an effect on the dimensionless maximum pressure distribution of the slider pocket bearing. However, it tends to make a shift in the dimensionless maximum pressure distribution. The maximum dimensionless pressure distribution in the first pocket P_3 is 0.0005 and the second pocket P_5 is 0.0038; the value is nearly the same for all inlet lengths A . Increasing the dimensionless inlet length A leads to a negative effect on the load support, thus a decrease in W is found (see Table 2). This is due to a decrease in the out length land e (see Figure 2a). Cavitation occurs in the pockets and the cavitation length increases in the pockets (see insert Figure 2a). However, in contrast to the previous case, for the condition in which boundary slip is present in the inlet length, no

cavitation takes place. The effect of slip in the inlet increases the pressure distribution. A pressure drop still occurs when the fluid enters the second pocket. The pressure drop does not occur when slip is present between the pockets as well (see Figure 2c). So, boundary slip in the inlet and the area before the second pocket can eliminate cavitation. In the case of boundary slip, the extension of the inlet length has a positive effect, namely of increasing the maximum pressure and the load support (see Table 2). The maximum load support is 0.0417 for the inlet length A of 0.2 for the condition in which boundary slip is present in the inlet and between the pockets.

TABLE 2: Dimensionless load-carrying capacity in double pocket bearing, varying A .

A	Dimensionless load-carrying capacity, W		
	No-Slip	Slip a	Slip a and c
0.1	0.000940	0.0204	0.0320
0.15	0.000458	0.0260	0.0371
2.0	0.000218	0.0313	0.0417

(a) No-slip condition.

(b) Slip at inlet.

(c) Slip at inlet and between pockets.

FIGURE 2: The effect of non-dimensional inlet length A on the dimensionless pressure P .

(Note: $h_o = 1.0 \times 10^{-6}$ m, $h_d = 1.0 \times 10^{-6}$ m, $b = 0.0015$ m, $c = 0.002$ m, $d = 0.0015$ m, $B_o = 0.02$ m).

3.2. The variation of dimensionless length between pocket C

For the no-slip condition, cavitation exits in all pockets for the variations in length applied between the pockets (see Figure 3a). However, the cavitation length in the first pocket decreases with the increase of the dimensionless length between the pockets. However, the cavitation length in second pocket increases (see insert Figure 3a). Increasing the length between the pockets does not improve the pressure distribution. The maximum in the pressure distribution tends to shift by varying the length between the pockets. The load support decreases with increasing the length between the pockets (see Table 3). With boundary slip in the inlet, and increasing the dimensionless length between the pockets C , the maximum dimensionless pressure in the second pocket P_5 decreases and as a result the load support also decreases (see Table 3). For the case in which slip is located at the inlet and at the surface between the pockets, an increase of the dimensionless length of the area between the pockets C gives an increase in the maximum dimensionless pressure in the second pocket P_5 (see Figure 3c), resulting in an increase in load support (see Table 3). It should be noted that the highest load support achieved is 0.0450 for the dimensionless length between the pockets C of 0.20 for the case in which boundary slip is

present at the inlet length and the area between the pockets.

TABLE 3: Dimensionless load-carrying capacity in double pocket bearing with variation C .

C	Dimensionless load carrying capacity, W		
	No-Slip	Slip a	Slip a and c
0.10	0.000458	0.0260	0.0371
0.15	0.000358	0.0253	0.0413
0.20	0.000274	0.0246	0.0450

(a) No-slip condition.

(b) Slip at inlet.

(c) Slip at inlet and between pockets.

FIGURE 3: The effect of non-dimensional length between pockets C on the dimensionless pressure P . (Note: $h_o = 1.0 \times 10^{-6}$ m, $h_d = 1.0 \times 10^{-6}$ m, $a = 0.003$ m, $b = 0.0015$ m, $d = 0.0015$ m, $B_o = 0.02$ m).

3.3. The variation of dimensionless pocket lengths B and D

In this section, the dimensionless length B of the first pocket and D of the second pocket are varied, while remaining equal in size. Figure 4 shows the hydrodynamic dimensionless pressure distribution as a function of the dimensionless pocket length. Cavitation takes place in the first and second pocket and affects the dimensionless pressure distribution and dimensionless load support. For the no-slip condition, increasing the dimensionless pocket length increases the cavitation area at the first pocket (see insert Figure 4a). It can be observed that the maximum dimensionless pressure distribution in the first pocket P_3 and second pocket P_5 is the same. However, the dimensionless load support decreases with increasing dimensionless pocket length (see Table 4). There will even be a lubrication failure due to the existence of negative load support for $B = D = 0.175$. In this condition, the resulting dimensionless pressure distribution is unable to resist the atmospheric pressure (see Equation A14 in Appendix). For boundary slip in the inlet situation, increasing the dimensionless pocket length gives a rise in the maximum dimensionless pressure in the first pocket P_3 and second pocket P_5 (see Figure 4b), thus the dimensionless load support also increases (see Table 4). For the case in which the slip is located at the inlet and at the surface between pockets, the maximum dimensionless pressure

in the second pocket P_5 and dimensionless load support increase with increasing dimensionless pocket length B and D . The maximum dimensionless load support W is 0.0468 for the dimensionless pocket length $B = D$ is 0.175.

TABLE 4: Dimensionless load-carrying capacity in double pocket bearing with variation B and D .

$B=D$	Dimensionless load carrying capacity, W		
	No-Slip	Slip a	Slip a and c
0.075	0.000358	0.0253	0.0413
0.125	0.000140	0.0290	0.0449
0.175	-0.000044	0.0314	0.0468

(a) No slip condition.

(b) Slip at inlet.

(c) Slip at inlet and between pockets.

FIGURE 4: The effect of non-dimensional pocket length B on the dimensionless pressure P .

(Note: $h_o = 1.0 \times 10^{-6}$ m, $h_d = 1.0 \times 10^{-6}$ m, $a = 0.003$ m, $c = 0.003$ m, $B_o = 0.02$ m).

3.4. The variation on dimensionless pocket depth HD

The dimensionless pocket depth of the first pocket HD_1 and of the second pocket HD_2 are taken in the calculations to be the same, i.e. HD . This value is varied as 1.0, 2.0 and 3.0 respectively. The effect of the dimensionless pocket depth HD on the dimensionless pressure P is shown in Figure 5. In the case of the no-slip condition, increasing the dimensionless pocket depth HD does not not affect the maximum dimensionless pressure in the first pocket P_3 and second pocket P_5 . The increasing dimensionless pocket depth HD decreases the cavitation area in the first and the second pocket (see insert Figure 5a). The decreasing cavitation area has a positive effect on the load support (see Table 5). For the slip in the inlet, increasing the dimensionless pocket depth HD decreases the maximum dimensionless pressure in the first pocket P_3 and the second pocket P_5 (see Figure 5b), resulting in a decrease in dimensionless load support (see Table 5). For the case in which slip is located at the inlet and the surface between the pockets, increasing the dimensionless pocket depth decreases the maximum dimensionless pressure in the first pocket P_3 and in the second pocket P_5 (see Figure 5c), decreasing the dimensionless load support.

TABLE 5: Dimensionless load-carrying capacity in double pocket bearing varying HD .

HD	Dimensionless load carrying capacity, W		
	No-Slip	Slip a	Slip a and c
1.00	0.000140	0.0290	0.0449
2.00	0.000142	0.0261	0.0421
3.00	0.000145	0.0244	0.0402

(a) No-slip condition.

(b) Slip at inlet.

(c) Slip at inlet and between pockets.

FIGURE 5: The effect of non-dimensional texture depth HD on the dimensionless pressure P . (Note: $a = 0.003$ m, $b = 0.0025$ m, $c = 0.003$ m, $d = 0.0025$ m, $e = 0.009$ m, $B_o = 0.02$ m).

4. Conclusions

This paper focuses on the effect of the boundary slip in a double pocket slider bearing on pressure distribution (load support) and cavitation. The following conclusions summarize the results of the present study:

1. For the no-slip condition, cavitation occurs for all variations of the dimensionless inlet length A , dimensionless length between pockets C , dimensionless pocket length B , D and dimensionless pocket depth HD .
2. For the no-slip condition, increasing dimensionless inlet length, as well as dimensionless length between pockets and pocket length, decrease the dimensionless load support. It is also found that the dimensionless load support increases with increasing dimensionless pocket depth.
3. Boundary slip in the inlet shows an increase of the dimensionless pressure distribution in the first pocket and reduces cavitation in the first and second pocket. However, the maximum dimensionless pressure in the second pocket decreases.
4. With boundary slip at the inlet and between the pockets, this maximum dimensionless pressure in the first and second pocket increase for all geometry variations studied. As a consequence no cavitation occurs and a maximum for the dimensionless load support is obtained.

Nomenclatures

a	= inlet length
b	= first pocket width
B_o	= slider length
c	= length between pockets
d	= second pocket width
e	= outlet land length
h_d	= height of microtexture pocket (excluding film thickness)
h_p	= film thickness in pocket
h_o	= minimum film thickness
p_3	= pressure at position C
p_5	= pressure at position E
p_{atm}	= atmospheric pressure
p_{cav}	= cavitation pressure
U	= sliding velocity
w	= load support
q	= flow rate
x	= coordinate in sliding direction
X_b	= non-cavitated fraction of first pocket
X_d	= non-cavitated fraction of second pocket
z	= coordinate through film thickness
μ	= lubricant dynamic viscosity
α_h	= slip coefficient at surface h (stationary)

Non dimensional parameters

$$A = a / B_o$$

$$B = b / B_o$$

$$C = c / B_o$$

$$D = d / B_o$$

$$HD = h_d / h_o$$

$$P = \rho h_o^2 / (U \mu B_o)$$

$$X = x / B_o$$

$$W = w h_o^2 / U \mu B_o^2$$

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Appendix A

The analysis below is based on continuity of flow through the bearing and uses the terminology as shown in Figure 1. The corresponding flow rate equation can be expressed as by Fowell et al., [11]:

A. No-slip condition and cavitation in double recess

A.1. Pressure distribution

$$q_{AB} = \frac{U}{2} h_o - \frac{h_o^3}{12\mu} \left(\frac{P_{cav} - P_{atm}}{a} \right) \quad (A.1)$$

$$q_{B'C} = \frac{U}{2} h_p - \frac{h_p^3}{12\mu} \left(\frac{p_3 - p_{cav}}{X_b} \right) \quad (A.2)$$

$$q_{CD} = \frac{U}{2} h_o - \frac{h_o^3}{12\mu} \left(\frac{p_{cav} - p_3}{c} \right) \quad (A.3)$$

$$q_{D'E} = \frac{U}{2} h_p - \frac{h_p^3}{12\mu} \left(\frac{p_5 - p_{cav}}{X_d} \right) \quad (A.4)$$

$$q_{EF} = \frac{U}{2} h_o - \frac{h_o^3}{12\mu} \left(\frac{p_{atm} - p_5}{e} \right) \quad (A.5)$$

with one $q_{AB} = q_{B'C} = q_{CD} = q_{D'E} = q_{EF}$ condition based on volume conservation.

When q_{AB} is set as equal to $q_{B'C}$

$$\left(\frac{p_3 - p_{cav}}{X_b} \right) = \frac{6\mu U}{h_p^3} (h_p - h_o) - \frac{h_o^3}{h_p^3} \left(\frac{p_{cav} - p_{atm}}{a} \right) \quad (A.6)$$

When $q_{B'C}$ is set as equal to q_{CD}

$$\left(\frac{p_3 - p_{cav}}{X_b} \right) = \frac{6\mu U}{h_p^3} (h_p - h_o) + \frac{h_o^3}{h_p^3} \left(\frac{p_{cav} - p_3}{c} \right) \quad (A.7)$$

Substituting Equation (A.6) in Equation (A.7) gives

$$p_3 = p_{cav} + \frac{c}{a} (p_{atm} - p_{cav}) \quad (A.8)$$

$$X_b = \frac{ch_p^3 (p_{atm} - p_{cav})}{6\mu U a (h_p - h_o) - h_o^3 (p_{atm} - p_{cav})} \quad (A.9)$$

When q_{CD} is set as equal to $q_{D'E}$

$$\left(\frac{p_5 - p_{cav}}{X_d} \right) = \frac{6\mu U}{h_p^3} (h_p - h_o) + \frac{h_o^3}{h_p^3} \left(\frac{p_{cav} - p_3}{c} \right) \quad (A.10)$$

When $q_{D'E}$ is set as equal to q_{EF}

$$\left(\frac{p_5 - p_{cav}}{X_d} \right) = \frac{6\mu U}{h_p^3} (h_p - h_o) - \frac{h_o^3}{h_p^3} \left(\frac{p_{atm} - p_5}{e} \right) \quad (A.11)$$

Substituting Equation (A.10) in Equation (A.11) gives

$$p_5 = p_{atm} + \frac{e}{a} (p_{atm} - p_{cav}) \quad (A.12)$$

$$X_d = \frac{(a+e)h_p^3 (p_{atm} - p_{cav})}{6\mu U a (h_p - h_o) - h_o^3 (p_{atm} - p_{cav})} \quad (A.13)$$

A.2. Load Support

$$w = w_{AB} + w_{BB'} + w_{B'C} + w_{CD} + w_{DD'} + w_{D'E} + w_{EF} - p_{atm}(a+b+c+d+e) \quad (A.14)$$

$$w_{AB} = \left(\frac{p_{atm} + p_{cav}}{2} \right) a \quad (A.15)$$

$$w_{BB'} = \left(\frac{p_{cav} + p_{cav}}{2} \right) (b - X_b) \quad (A.16)$$

$$W_{B'C} = \left(\frac{p_{cav} + p_3}{2} \right) X_b \quad (\text{A.17})$$

$$W_{CD} = \left(\frac{p_3 + p_{cav}}{2} \right) c \quad (\text{A.18})$$

$$W_{DD'} = \left(\frac{p_{cav} + p_{cav}}{2} \right) (d - X_d) \quad (\text{A.19})$$

$$W_{D'E} = \left(\frac{p_{cav} + p_5}{2} \right) X_d \quad (\text{A.20})$$

$$w_{EF} = \left(\frac{p_5 + p_{atm}}{2} \right) e \quad (\text{A.21})$$

$$X_d = \frac{(a+e)C_{p_2}^*(p_{atm} - p_{cav})}{6\mu Ua(C_{p_2} - h_o) - h_o^3(p_{atm} - p_{cav})} \quad (\text{A.22})$$

Paper F

M. Muchammad, M. Tauviquirrahman, A.W. Pratomo,
J. Jamari, D.J. Schipper, 2015

The Beneficial Effect of Superhydrophobic Layer
Addition on Lubrication Behavior
in Bearing Application

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The Beneficial Effect of Superhydrophobic Layer Addition on Lubrication Behavior in Bearing Application

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Abstract Recently, superhydrophobic materials have gained a lot of attention for special applications, such as anti-stiction and energy conversion. In this paper, details are given of a theoretical investigation that was carried out to study the effect of adding a superhydrophobic layer on the tribological performance of bearings, further comparing it with textured bearings. In doing so, different texture depths were also studied and compared with each other to find the optimal textured bearing. The superhydrophobic behaviour was characterized by the Navier slip model. The results showed that the presence of a superhydrophobic layer in a bearing could result in a significant improvement in the hydrodynamic pressure distribution and in the shear stress, when compared with a textured bearing.

Keywords: Analytical method, bearing, lubrication, superhydrophobic.

1. Introduction

Recently, hydrodynamic lubricated sliding contacts with a surface texture and surface slip have gained a lot of attention. The sliding step bearing is one example of texturing, leading to a high load support. Regarding slip, Choo et al. [1] showed that liquid slip could be used to considerably reduce friction under full film, hydrodynamic lubrication. Tauviquirrahman et al. [2] demonstrated that partial texturing gave even better results than full texturing. However, in comparison with a flat complex slip surface, a partially textured surface is still less efficient for

enhancing load support, even if this textured configuration is combined with a slip condition. Rahmani et al. [3] examined the step bearing analytically by considering the variations of hydrostatic pressure on the bearing borders in order to find circumstances in which an optimum Rayleigh step could be introduced concerning load capacity, friction force, friction coefficient, and lubricant flow rate. Shukla and Deheri showed [4] that for a better performance of the bearing system, the slip velocity should be minimized. Naduvanamani and Siddangouda [5] showed that a negatively skewed surface roughness pattern increased the load-carrying capacity and decreased the coefficient of friction, whereas the opposite effects were found in the positively skewed surface roughness pattern. Rao et al. [6] stated that partially textured slip was effective in the case of a parallel slider and a journal bearing with respect to load capacity. Yu et al. [7] studied theoretically the influence of recess depth and rotating velocity on the dynamic pressure of a bearing according to the lubricating theory. They showed that an improved characteristic would be affected by recess depth and rotating velocity. Farmer and Shepherd [8] used singular perturbation methods to analyse the isothermal operation of the Rayleigh step slider bearing of narrow geometry, when the bearing number was moderate and the gas lubricant was rarefied, so that 'slip flow' occurred.

In most of the aforementioned works, the hydrodynamic analysis ignored cavitation, which might affect the results significantly. In the present study, the beneficial effect of slip

phenomenon resulting from the superhydrophobic layer will be explored in the performance of step slider bearings. The cavitation is also of particular interest in terms of obtaining more realistic results. The modified form of the Reynolds equation containing the cavitation effect is derived, in which the wall slip is allowed to occur in a textured bearing. In this case, three slip patterns are compared.

2. Mathematical model and its solution

In the present paper, the mixed slip surface and the step bearing are explored (Figs. 1 and 2 respectively). For the analysis of the mixed slip bearing in this study, boundary slip is applied at the leading edge of the contact and its length is 0.65 times the bearing length [2]. In addition, the step bearing with different patterns are analysed, i.e. no slip (Fig. 2a), partial slip (Fig. 2b), and full slip (Fig. 2c). In hydrodynamic lubrication, the governing equation is described by the well-known Reynolds equation. The methodology used in the derivation of the extended Reynolds equation considering boundary slip is quite similar to that employed by Fowell [9] and is given in detail in Appendix A.

$$P_2 \left[(h_p^3 + 3h_p^3 K_p) \frac{ab + bc}{ab^2} + \left(\frac{h_o^3 + 3h_o^3 K_o}{a} \right) \right] =$$

$$P_{atm} \left[(h_p^3 + 3h_p^3 K_p) \frac{ab + bc}{ab^2} + \left(\frac{h_o^3 + 3h_o^3 K_o}{a} \right) \right]$$

$$-6\mu U \left[(h_p + h_p K_p) - (h_o + h_o K_o) \right] \quad (1)$$

For the parallel and Rayleigh step bearing with slip as well as the mixed slip as illustrated in Figs. 1 and 2, expressions for the pressure and flow rate can be obtained. Applying pressure continuity at boundary points, a set of governing equations can be obtained for the flow rate and constants of integration in the pressure expressions, see Appendix A for details. In Appendix A, the derivation of the flow rate and pressure, which are the basis of this analysis, are presented in Eqs. A-1 – A-11. Solving the set of governing equations and applying the quantities in the nomenclature, the analytical solution for

the slip case is shown in the figures below. It needs to be noted that for the following computations, parameters as shown in table 1 are used.

FIGURE 1: A schematic representation of parallel slider bearing with optimal slip [2].

(a) No-slip condition.

(b) Partial slip condition.

(c) Full slip condition.

FIGURE 2: A schematic representation of Rayleigh step bearing with different boundary slip configurations.

TABLE 1: Main Characteristics of the bearing analysed.

Parameters	Data Setting	Unit
Bearing length B_o	0.02	m
Length of textured region a	0.013	m
Exit land length b	0.007	m
Atmospheric pressure p_{atm}	100	kPa
Min. film thickness h_o	1	μm
Moving surface velocity U	1	m/s
Slip coefficient α	0.02	$\text{m}^2\text{s/kg}$
Dynamic viscosity μ	0.01	$\text{Pa}\cdot\text{s}$

3. Results and discussion

Figure 3 depicts the pressure distribution for the partial slip Rayleigh step bearing with $HD = 0$ and flat slip [2] parallel slider bearing for the length of slip region $a = 0.65*B_o$. For the latter, the dimensionless length ($A=\mu\alpha/h_o$) of 50 is considered. It shows a comparison between present model and that of Tauviquirrahman et al. [2]. From the figure, it was shown that the value pressure distribution of partial slip was the same with flat slip [2]. It should be noted that the calculated results are obtained using the same slip parameter and the length of slip region. The partial and flat slips produce the same load support, ($W^+ = wh_o^2 / \mu UB_o = 0.3297$).

FIGURE 3: A schematic Pressure distribution of partial slip Rayleigh step and flat slip [2]. (Note: $x^+ = x / B_o$, $HD = h_d / h_o$ and $p^+ = ph_o^2 / \mu UB_o$).

Figures 4 and 5 show the pressure profile as well as the shear stress for the Rayleigh step bearing by varying the texture depth ($HD = h_d/h_o = 0.25$; 0.50; 0.75 and 1.00) and the length of texture region $a = 0.65*B_o$. It can be seen that the pressure distribution increases along with the increase of HD and achieves the maximum value

when $HD = 0.75$ ($p^+ = 0.405$). Then it decreases by the increase of HD . Therefore, the highest load support ($W^+ = 0.203$) is achieved when $HD = 0.75$.

Regarding the shear stress profile, Fig. 4, the shear stress has a similar trend with the increasing HD and has a maximum shear stress when $HD = 0.75$ ($\tau_{AB}^+ = \tau h_o^2 / \mu U = 1.110$).

FIGURE 4: Pressure distribution of Rayleigh step bearing with no slip.

FIGURE 5: Shear stress distribution of Rayleigh step bearing no slip.

Figures 6 and 7 show the pressure distribution as well as the shear stress for the Rayleigh step bearing with partial slip (see Fig. 2b). It can be seen that the pressure distribution decreases with the increase of HD . The maximum pressure ($P^+ = 0.664$) is achieved when $HD = 0$. The load support reaches the highest value (i.e. $W^+ = 0.332$) when $HD = 0$. Adding the partial slip and an increase in HD results in the increase in the shear stress along with the AB region (see Fig. 8) and finally reaches a maximum value ($\tau_{AB}^+ = 1.171$) when $HD = 0.5$. The minimum

shear stress ($\tau_{AB}^+ = 1.024$) at the AB area occurs when $HD = 0$. For BC area, the shear stress increases with an increase in HD . The minimum shear stress $\tau_{BC}^+ = 0.051$ when $HD = 0$. The lowest shear stress $\tau_{BC}^+ = 0.683$ when $HD = 0$.

FIGURE 6: The pressure distribution of the Rayleigh step bearing with partial slip.

FIGURE 7: Shear stress distribution of Rayleigh step bearing with partial slip.

Figures 8 and 9 are the pressure profiles and shear stress of the Rayleigh step bearing with the full slip condition (see Fig. 2c). Introducing full slip in the Rayleigh step bearing gives the same behaviour with the Rayleigh bearing without slip, in other words the increase in HD will increase the pressure profile ($P^+ = 0.202$), and reach a maximum value when $HD = 0.75$. However, subsequently the pressure profile tends to decrease again. The greatest load support W^+ is 0.101 when $HD = 0.75$. From Fig. 8, it can be observed that the shear stress profile has the same trend with HD as for the pressure profile as described before. The shear stress is

maximal (i.e. $\tau_{AB}^+ = 0.610$) when $HD = 1.00$ and minimal ($\tau_{AB}^+ = 0.250$) when $HD = 0.25$. For the BC area, it results in a negative shear stress. The friction force increases with an increase in HD . The friction force is minimal (i.e. $F^+ = 0.683$) when $HD = 0.25$ and maximal (i.e. $F^+ = 0.201$) when $HD = 1.00$.

FIGURE 8: Pressure distribution of Rayleigh step bearing with full slip.

FIGURE 9: Shear stress distribution of Rayleigh step bearing with full slip.

TABLE 2: Optimized lubricated contact characteristics for different slip patterns.

Slip condition	Load optimal		Friction optimal	
	HD	W^+	HD	F^+
Rayleigh step bearing	0.75	0.203	1.00	0.873
Partial slip Rayleigh step bearing	0.00	0.332	0.00	0.684
Full slip Rayleigh step bearing	0.75	0.101	0.25	0.037

The optimum condition for the slider bearing is summarized in Table 2; the highest load support is found in the Rayleigh step bearing with $HD = 0$ (parallel slider bearing). The load support is maximum when boundary slip is introduced at the inlet when $0.65*Bo$ with load support $W^+ = 0.332$. Whilst for the friction force (i.e. $F^+=0.037$), the minimum value is achieved when $HD = 0.25$ in full slip condition. Slip can be used to reduce friction under full film, hydrodynamic lubrication Choo et al. [1].

3. Conclusions

In the present paper, a comparison is made of several slip patterns for Rayleigh step bearing. The cavitation effect has been taken into account by varying the pocket depth. The following conclusions can be drawn:

The highest load support is achieved for a Rayleigh step bearing when $HD = 0$ (parallel slider bearing) with boundary slip at the leading edge of the contact, with a length of 0.65 times the bearing length ($W^+ = 0.332$). This value is 63.87% higher than for the Rayleigh step bearing without slip for an optimal HD (in this case, $HD = 0.75$). The load support is 228% higher than for the Rayleigh step bearing with full slip, and HD is optimal (i.e. $HD = 0.75$).

Introducing full slip in a Rayleigh step bearing will decrease the friction force. The friction force will also decrease with the decrease of HD . The minimum friction force is $F^+ = 0.038$ when $HD = 0.25$.

Nomenclatures

a	= length of textured region
A	= slip length
b	= exit land length
Bo	= slider length
F	= friction
h_d	= height of textured region
h_o	= minimum film thickness

p_2	= pressure at B
p_{atm}	= atmospheric pressure
U	= sliding velocity
W	= load support
q	= flow rate
x	= coordinate in sliding direction
z	= coordinate through film thickness
μ	= lubricant dynamic viscosity
α	= slip coefficient
τ	= shear stress

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Appendix A:

A.1. Parallel slider bearing with optimal slip

A.1. Pressure

$$q_{AB} = \frac{U}{2} C_o - \frac{\left[h_o^3 + \left(3h_o^3 \frac{\mu\alpha}{h_o + \mu\alpha} \right) \right]}{12\mu} \left(\frac{p_2 - p_{atm}}{a} \right) \quad (A.1)$$

$$q_{BC} = \frac{U}{2} h_o - \frac{h_o^3}{12\mu} \left(\frac{p_{atm} - p_2}{b} \right) \quad (A.2)$$

Continuity equation $q_{AB} = q_{BC}$

$$\frac{U}{2} C_o - \frac{\left[h_o^3 + \left(3h_o^3 \frac{\mu\alpha}{h_o + \mu\alpha} \right) \right]}{12\mu} \left(\frac{p_2 - p_{atm}}{a} \right) = \frac{U}{2} h_o - \frac{h_o^3}{12\mu} \left(\frac{p_{atm} - p_2}{b} \right) \quad (A.3)$$

$$p_2 = p_{atm} + \frac{6\mu U (C_o - h_o) ab}{ah_o^3 + b \left[h_o^3 + \left(3h_o^3 \frac{\mu\alpha}{h_o + \mu\alpha} \right) \right]} \quad (A.4)$$

A.1.2. Shear Stress

$$\tau = -\frac{\mu U}{h + \mu\alpha_h} - \frac{h}{2} \frac{\partial p}{\partial x} \frac{h + 2\mu\alpha_h}{h + \mu\alpha_h} \quad (A.5)$$

$$\tau_{AB} = -\frac{\mu U}{h_o + \mu\alpha_h} - \frac{C_o}{2} \left(\frac{p_2 - p_{atm}}{a} \right) \quad (A.6)$$

$$\tau_{BC} = -\frac{\mu U}{h_o} - \frac{h_o}{2} \left(\frac{p_{atm} - p_2}{b} \right) \quad (A.7)$$

A.2. Rayleigh step bearing no-slip condition

A.2.1. Pressure

$$\frac{U}{2}h_p - \frac{h_p^3}{12\mu}\left(\frac{p_2 - p_{atm}}{a}\right) = \frac{U}{2}h_o - \frac{h_o^3}{12\mu}\left(\frac{p_{atm} - p_2}{b}\right) \quad (\text{A.8})$$

$$p_2 = p_{atm} + \frac{6\mu U(h_p - h_o)ab}{ah_o^3 + bh_p^3} \quad (\text{A.9})$$

A.1.2 Shear Stress

$$\tau_{AB} = -\frac{\mu U}{h_p} - \frac{h_p}{2}\left(\frac{p_2 - p_{atm}}{a}\right) \quad (\text{A.10})$$

$$\tau_{BC} = -\frac{\mu U}{h_o} - \frac{h_o}{2}\left(\frac{p_{atm} - p_2}{b}\right) \quad (\text{A.11})$$

Paper G

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The Effect of Boundary Slip and Cavitation on Hydrodynamic Pressure Generation in Pocket Bearings

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Abstract Surface texturing, as well as boundary slip, is an effective approach to increasing the tribological performance, friction and load support of lubricated mechanical components. In a textured bearing with boundary slip, however, cavitation – which is generally ignored – is important and it alters the flow behaviour. Therefore, in this paper a novel analytical equation is proposed based on a modified Reynolds equation considering boundary slip and cavitation. The modified Reynolds equation is derived by following the general approach to deducing the Reynolds equation from the Navier-Stokes equation by assuming classical assumptions, except that boundary slip is present. A theoretical investigation was carried out to study the effect of boundary slip including cavitation on the pressure distribution of a pocket bearing. Different slip configurations (full slip, single slip and semi slip) were studied and compared with various pocket depths. The results indicate that the presence of boundary slip both on the leading edge of the contact and the pocket area (i.e. the “semi slip” case) could result in a significant improvement in the hydrodynamic pressure generation; thus, it causes an increased load support. The above improvement depends on the pocket depth of the bearing.

Keywords: Cavitation, pocket bearing, boundary slip.

1. Introduction

During recent years, in classical lubrication mechanics the assumption of no-slip at the interface of solid and fluid has been used as the boundary condition of the fluid solid interaction. This assumption has been widely used in various engineering designs and experiments. However, along with the significant development of the measurement technology, this assumption may no longer be accurate. Control of the boundary slip will allow a degree of control to increase the hydrodynamic pressure in confined systems and to be important in a lubricated sliding contact like the micro-electro-mechanical system (MEMS). How to control the boundary slip in an application like a bearing is going to be one of the challenging tasks in the future.

In addition, various researchers have focused attention on the study of tribological effects obtained by introducing surface texturing. Various surface texture models and area of texturing have been investigated to conclude that the surface texture enhanced the hydrodynamic performance.

Generally speaking, both surface texturing and boundary slip are effective approaches in increasing the performance of a bearing. In literature [1-8], some have studied the lubricant flow in a bearing. The purpose of lubrication is to separate surfaces in relative motion in order to reduce friction and wear. Regarding the improvement of the lubrication performance, applying boundary slip on one or both of the opposing surfaces is introduced. It is shown that

Recently, the presence of cavitation in surface textured slider bearings has been researched [4-8]. The present paper extends the analytical method in parallel pocket slider bearings including boundary slip (full slip, single slip and semi slip) of the texture cell.

The pocket bearing configuration is shown in Fig. 1. In this paper, different boundary slip patterns are introduced. In detail, Figs. 1a-c present a schematic illustration of a pocket sliding bearing with different boundary slip configurations: (a) full slip, (b) single slip, and (c) semi slip. As reflected in Fig. 1a, “full slip” means that the boundary slip is applied over the whole surface of the bearing including the pocket surface. “Single slip” refers to the condition in which the boundary slip is applied only at the leading edge of the contact (Fig. 1b). When the boundary slip is applied both at the leading edge of the contact and at the pocket surface, the term “semi slip” is used. In the present paper, the pocket depth of the bearing is varied. The dimensionless pocket depth HD (where $HD = h_d / h_o$) is introduced with variations of HD of 0.25, 0.50, 0.75 and 1.00. It should be noted for all following configurations of slip textured bearing (see Fig. 1) that the cavitation effect is taken into account by introducing the cavitated zone ($b - X_b$). From the physical point of view, the aim of using this approach is to investigate whether or not the slip application of boundary on a specific area could minimize (or even prevent) the cavitation phenomena.

lubricated bearing including slip, a modified Reynolds equation is developed by taking cavitation into account (Eq. 1). The derivation of the analytical equation for full slip condition is presented in detail in the Appendix. For other slip patterns, the derivation of lubrication equations is similar.

In the present study, the lubricated pocket bearing is operating under steady-state conditions. The hydrodynamic pressure generation, and thus the load support generation, is of particular interest. The load support is determined by integrating the calculated hydrodynamic pressure field along the surface contact.

FIGURE 1: A schematic representation of pocket bearing with boundary slip for different configurations.

TABLE 1: Main Characteristics of the bearing analysed.

Parameters	Data Setting	Unit
Bearing length B_o	0.02	m
Inlet length, a	0.002	m
Pocket length, b	0.003	m
Exit land length, c	0.015	m
Atmospheric pressure, p_{atm}	100	kPa
Cavitation pressure, p_{cav}	0	kPa
Min. film thickness, h_o	1	μm
Moving surface velocity, U	1	m/s
Slip coefficient, α	0.02	$\text{m}^2\text{s/kg}$
Dynamic viscosity, μ	0.01	Pa.s

3. Results and discussion

As discussed by Fowell et al. [4], the existence of inlet suction with respect to the hydrodynamic lubrication mechanism has a significant effect on a pocket bearing. In this paper, the texture parameter pocket depth and the boundary slip on one surface will be explored in more detail.

Figures 2-4 show the hydrodynamic pressure distribution of the parallel pocket bearing with different slip configuration for several pocket depths. The hydrodynamic pressure profile for the full slip configuration of the pocket bearing is shown in Fig. 2. As depicted in Fig. 2, the pressure initially tends to decrease to the cavitation pressure. After decreasing, the pressure increases to the maximum pressure and finally decreases. This trend prevails for all values of pocket depth. It should be noted that the greatest difference of the dimensionless load support W^+ (where $W^+ = Wh_o^2 / \mu UB_o^2$) among the different pocket depth configurations is only

0.6%. The highest W^+ resulted in the bearing with $HD = 0.50$ is 1.159×10^{-4} , whereas the lowest W^+ is 1.166×10^{-4} (produced by bearing with $HD = 1.00$). It indicates that boundary slip has a more dominant effect than the texturing parameter (in this case, pocket depth) with respect to the hydrodynamic pressure behaviour. This finding matches well with the published result of Tauviquirrahman et al. [6]. It can also be observed from Fig. 2 that the cavitation phenomena are found along the pocket area of the bearing for all pocket depths considered here. Therefore, applying the full slip does not lead to a positive effect with respect to the prevention of cavitation.

FIGURE 2: Hydrodynamic pressure distribution of full slip pocket bearing for different pocket depths. (Note: $x^+ = x / B_o$ and $p^+ = ph_o^2 / U \mu B_o$).

Figure 3 shows the hydrodynamic pressure profile for the single slip configuration of the pocket bearing (see Fig. 1). Unlike the full slip characteristics, the introduction of single slip is able to prevent the existence of cavitation in the bearing for all pocket depths. There is an increase in pressure in the pocket area; P_3 (see Fig. 1) is higher than the P_2 . However, the increase in P_3 becomes less sensitive for HD that is higher than 0.5. From Fig. 3, it is also observed that combining single slip with pocket depth leads to an improved pressure generation, and thus load support. When $HD = 0.25$, $W^+ = 0.2161$, and for $HD = 1.00$, the lubrication is able to produce a W^+ of 0.2422, which is 12% higher.

FIGURE 3: Hydrodynamic pressure distribution of single slip pocket bearing for different pocket depths.

Figure 4 shows the hydrodynamic pressure profile for the semi slip configuration of the pocket bearing. It can be seen that the pressure of P_2 tends to slightly increase for all pocket depths, whereas the P_3 tends to decrease. However, the increase in P_2 is smaller than the decrease in P_3 . Consequently, the generated load support also decreases due to the increase of pocket depth. The decrease in W^+ is up to 21% (from 0.280 down to 0.232).

By comparing the hydrodynamic pressure profiles for all slip configurations (Figs. 2-4), some findings are synthesized. Firstly, the maximum P_2 (i.e. $P_2^+ = 0.2851$) is found for the semi slip condition with $HD = 1.00$, whereas the full slip configuration gives the lowest P_2 , in other words zero pressure or cavitation pressure for all HD values. Secondly, the highest P_3 (i.e. $P_3^+ = 0.5483$) occurs for the semi slip configuration with $HD = 0.25$, whereas the lowest P_3 ($P_3^+ = 0.0081$) is found for the full slip condition. Fig. 5 shows the hydrodynamic pressure distributions for the optimal pocket depths for the different slip configurations with respect to the load support as shown in Table 2. From Fig. 5 and Table 2, it is clear that the semi slip pattern with optimal HD gives the best improvement of the lubrication performance with respect to load support.

FIGURE 4: Hydrodynamic pressure distribution of semi slip pocket bearing for different pocket depths.

FIGURE 5: Hydrodynamic pressure distribution for different slip configurations with optimum pocket depth.

TABLE 2: Optimized lubricated contact characteristics for different slip patterns.

Slip condition	HD optimal	W^+
Full slip	1.00	0.000116
Single slip	0.75	0.245
Semi slip	0.25	0.280
No slip	1.00	0.000115

4. Conclusions

This paper focuses on enhancing the hydrodynamic pressure as a result of the load support of a pocket bearing by boundary slip including cavitation. The conclusions based on the analysis presented in this paper are as follows.

Applying different boundary slip configurations affects the lubrication performance of the pocket bearing with respect to the load support. The best slip configuration is semi slip with $HD = 0.25$.

Introducing a correct slip pattern (semi slip and single slip) in the pocket bearing reveals that cavitation is prevented.

Nomenclatures

A	= inlet length
b	= pocket width
B_o	= slider length
c	= outlet land length
F	= friction
h_d	= height of microtexture pocket (excluding film thickness)
h_o	= minimum film thickness in inlet land at B
p_2	= pressure at B
p_3	= pressure at C
p_{atm}	= atmospheric pressure
p_{cav}	= cavitation pressure
U	= sliding velocity
W	= load support
q	= flow rate
x	= coordinate in sliding direction
X_b	= non-cavitated fraction of pocket breadth

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Appendix A:

A.1. Full slip condition

A.1.1 Cavitation in recess

For the areas of the bearing containing parallel surfaces, the film thickness h is constant so the pressure gradient must be linear (see Eq. A.1). The flow (shear and pressure) equation (A.1) can be modified for each region of the bearing:

$$q_{AB} = \frac{U}{2} \frac{h_o^2 + 2\mu h_o \alpha_h}{h_o + \mu \alpha_h} - \frac{h_o^3}{12\mu} \frac{h_o + 4\mu \alpha_h}{h_o + \mu \alpha_h} \left(\frac{p_{cav} - p_{atm}}{a} \right) \quad (A.1)$$

$$q_{BC} = \frac{U}{2} \frac{h_p^2 + 2\mu h_p \alpha_h}{h_p + \mu \alpha_h} - \frac{h_p^3}{12\mu} \frac{h_p + 4\mu \alpha_h}{h_p + \mu \alpha_h} \left(\frac{p_3 - p_{cav}}{X_b} \right) \quad (A.2)$$

$$q_{CD} = \frac{U}{2} \frac{h_o^2 + 2\mu h_o \alpha_h}{h_o + \mu \alpha_h} - \frac{h_o^3}{12\mu} \frac{h_o + 4\mu \alpha_h}{h_o + \mu \alpha_h} \left(\frac{p_{atm} - p_{cav}}{c} \right) \quad (A.3)$$

Based on the volume conservation $q_{AB} = q_{BC} = q_{CD}$, see Fig. A1.

$$p_3 = p_{atm} + \frac{c}{a} (p_{atm} - p_{cav}) \quad (A.4)$$

$$X_b = \frac{\left(h_p^3 + 3h_p^3 \frac{\mu \alpha_h}{h_p + \mu \alpha_h} \right) (p_3 - p_{cav})}{6\mu U \left[\left(h_p + h_p \frac{\mu \alpha_h}{h_p + \mu \alpha_h} \right) - \left(h_o + h_o \frac{\mu \alpha_h}{h_o + \mu \alpha_h} \right) \right] - \left(h_o^3 + 3h_o^3 \frac{\mu \alpha_h}{h_o + \mu \alpha_h} \right) \left(\frac{p_{atm} - p_{cav}}{a} \right)} \quad (A.5)$$

FIGURE A.1: Schematic representation of couette flow and poiseuille flow in pocket slider bearing.

We know that to ensure the volume concentration, the shear stress must consist of two terms, namely the couette shear (due to the velocity difference) and the poiseuille term (due to the pressure gradient). Due to the existence of the pocket, the flow based on couette flow shear rate only never yields the volume concentration. In this case, $h_o U$ (in the area of AB) $\neq (h_o + h_p) U$ (in the area of BC) $\neq \frac{1}{2} h_o U$ (in the CD area). Therefore, again to ensure the volume concentration in the flow, the pressure generation has to be compensated by the pressure flow (i.e. poiseuille flow).

Paper H

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CFD Analysis of Texture Depth and Length on the Performance of Hydrodynamic Lubricated Bearings

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Abstract Surface modification of a lubricated bearing by texturing has proven to improve the hydrodynamic performance. The present paper examines the effect of texture depth as well as the texture length on the tribological performance using a computational fluid dynamic (CFD) approach. The cavitation model is also considered to obtain more realistic characteristics of the bearing. It is shown that by increasing texture depth of the surface of the bearing, enhanced hydrodynamic pressure is achieved. Moreover, another interesting result is that the length of the cavitation region decreases with the increase in texture depth and decrease in texture length.

Keywords: Texture, CFD, cavitation, hydrodynamic pressure.

1. Introduction

For decades, the research into texturing of lubricated contacts has been of interest to researchers around the world. It is well known that the texturing technique has proven to improve the lubrication characteristics. Based on the hydrodynamic lubrication theory, numerous researchers have introduced several mechanisms of pressure on textured surfaces.

Using a mass-conserving numerical analysis, Fowell et al. [1] concluded that any convergence between the bearing surfaces provides a significant mechanism of pressure generation. Gherca et al. [2] evaluated the effects of the texture geometry using a mass-conserving model. It was found that the load support is strongly dependent on the geometrical features

such as size, density, and shape. Henry et al. [3] studied experimentally the effect of surface texturing on the steady-state behaviour of bearings. They concluded that the textured bearings reduce friction up to 30% at low loads, while for heavy loads their performance is equivalent or even lower than that of an untextured bearing. Yagi et al. [4] studied the value of load support enabled by a textured pattern in the hydrodynamic lubrication regime. They concluded that the mechanism of pressure generation was strongly affected by the boundary conditions. Zhang et al. [5] investigated the effect of a rectangular array of dimples on the film thickness. It was concluded that the pressure generation could be improved through appropriate partial arrangement of textures. Later, Shinde and Pawar [6] proposed the optimal surface texturing parameters to improve the lubrication performance of a bearing.

Based on published works, it can be highlighted that texturing produces more positive effects with respect to the load support. However, from a literature survey it emerges that most of the solutions of the lubrication problems have been solved by using the Reynolds equation solely, which may be questionable.

Following this, in the present paper a CFD method to solve the Navier-Stokes equation has been applied, along with cavitation, to investigate the texturing parameter, in particular texture depth on the hydrodynamic pressure.

2. Methods

2.1 Theory

In the present study, the lubrication problem is solved by the Navier-Stokes equation and continuity equations. The Navier-Stokes (N-S) equations are solved over the domain using a finite-volume method with the commercial CFD software package FLUENT®. The Navier-Stokes and the continuity equations can be expressed, respectively.

$$\rho \frac{Du_i}{Dt} = -\frac{\partial p}{\partial x_i} + \rho G_i + \frac{\partial}{\partial x_j} \left[2\eta e_{ij} - \frac{2}{3}\eta(\nabla \cdot \mathbf{u})\delta_{ij} \right] \quad (1)$$

$$\nabla \cdot \mathbf{u} = 0 \quad (2)$$

Unlike the previously published works in which the cavitation model is ignored, in the present study, the cavitation effect is taken into account. In FLUENT®, there are three available cavitation models: the Schneer and Sauer model, the Zwart-Gelber-Belamri model and the Sigbal et al. model [7]. In this study, the Zwart-Gelber-Belamri is employed due to its capability (less sensitive to mesh density, robust and converges quickly [7]).

In addition to the Navier-Stokes equation, in this work the lubrication problem is also solved by a modified Reynolds equation (Eq. 3 as below).

For more details, the derivation of the equation used here is described in [8, 9]. It should be noted that the isoviscous Newtonian one-dimensional Reynolds equation with slip is derived from the x -component of the Navier-Stokes equation that assumes an incompressible flow, ignoring the inertia effects in the film. However, for the analysis studied here, the slip coefficient is set to zero.

$$\begin{aligned} P_2 \left[(h_p^3 + 3h_p^3 K_p) \frac{ab+bc}{ab^2} + \left(\frac{h_0^3 + 3h_0^3 K_0}{a} \right) \right] = \\ P_{atm} \left[(h_p^3 + 3h_p^3 K_p) \frac{ab+bc}{ab^2} + \left(\frac{h_0^3 + 3h_0^3 K_0}{a} \right) \right] \\ - 6\mu U \left[(h_p + h_p K_p) - (h_0 + h_0 K_0) \right] \end{aligned} \quad (3)$$

2.1 CFD Model

Figure 1 gives the schematic illustration of a parallel textured slider bearing. The assumption of the no-slip boundary condition is adopted.

The main characteristics of the bearing and the lubricant properties studied are presented in Table 1 and 2 respectively.

FIGURE 1: Schematic of a parallel textured bearing.

TABLE 1: Characteristics of the main parameters of the bearing analysed.

Parameters	Data Setting	Unit
Bearing length, B_o	2.00	mm
Length of inlet, a	0.75	mm
Length of texture, b	0.25; 0.50; 0.75	mm
Length of outlet, c	0.75	mm
Minimum film thickness, h_o	4.00	μm
Depth of texture, h_d	2.00;	μm
Inlet pressure, p_a	100	kPa
Cavitation pressure, p_{cav}	50.0	kPa
Moving surface velocity, U	1.00	m/s

In the present study, the grid consists of three blocks with a grid system as shown in Fig. 2. The number of grids in the longitudinal direction (Nx) and transverse (Nz) is 1000 x 100. These grids result in 90,000 cells, 18,112 faces and

91,121 nodes based on the independent mesh study. In the present work, the texture depth is of particular interest. The pocket depth is categorized in a low texture depth and high texture depth, as shown in Fig. 3. The hydrodynamic pressure is used as an indicator to describe the texture depth effect using the Reynolds equation and Navier-Stokes equation.

TABLE 2: Characteristics of lubricant properties.

Parameters	Data Setting	Unit
Density oil, ρ_o	962	kg/m^3
Density oil vapour, ρ	0.02556	kg/m^3
Viscosity oil, μ_o	0.0135	Pa.s
Viscosity oil vapour, μ	0.0000126	Pa.s

FIGURE 2: Grid system analysed in the numerical analysis.

(a) $h_d = \frac{1}{2} h_o$ (low texture depth).

(b) $h_d = 2 h_o$ (high texture depth).

FIGURE 3: Pocket depth analysed.

3. Results and discussion

In this section, the results for different texture depths with respect to pressure distribution are shown in Fig. 4. In this figure the pressure distribution with $b = 0.25$ mm with a low and high texture depth is presented. It can be observed from Fig. 4a that the N-S solution gives a larger prediction of the pressure

compared to the Reynolds approach for low texture depth. This indicates that the inertia effect exists in the case of low texture depth. As a note, this inertia term cannot be predicted by the Reynolds equation. The interesting finding is that when texture depth is increased, the inertia effect disappears. As indicated in Fig. 4b, the predictions of the hydrodynamic pressure by the

N-S equation and by the Reynolds approach are exactly the same.

Related to the cavitation behaviour, a textured contact with a low texture depth leads to the generation of a cavitation area. However, the cavitation area is generally predicted to be smaller when using the N-S equation than with the Reynolds approach. On the other hand, the Reynolds theory underestimates the hydrodynamic pressure when the inertia effect has a dominant role.

(a) Low texture depth.

(b) High texture depth.

FIGURE 4: The pressure distribution as a function of coordinate in sliding direction of interest with $b = 0.25$ mm.

Figure 5 shows the hydrodynamic pressure predicted by the N-S approach and the Reynolds equation for the case of a texture length of $b = 0.5$ mm. It can be seen that for the case of a high texture depth, the pressure distribution predicted by the N-S approach is larger than that predicted

by the Reynolds equation. This means that the inertia effect exists in the texture contact with larger texture length b . For low texture depth, the prediction of the pressure by the two approaches is the same, in particular with respect to the cavitation region. In other words, the inertia and the cavitation term are independent. The profile of the hydrodynamic pressure is strongly affected by the texture depth as well as by the texture length. This finding is strengthened by the simulation result as indicated in Figure 6. For high b , in this case $b = 0.75$ mm, the prediction by the two approaches is similar both for low and high texture depth.

(a) Low texture depth.

(b) High texture depth.

FIGURE 5: The pressure distribution as a function of coordinate in sliding direction with $b = 0.5$ mm.

(a) Low texture depth.

(b) High texture depth.

FIGURE 6: The pressure distribution as a function of coordinate in sliding direction with $b = 0.75$ mm.

FIGURE 7: The load support as a function of texture depth, b in mm.

Figure 7 shows the load support for each texture depth and bearing texture length configuration, obtained by the CFD method either with the Navier-Stokes equation or with the Reynolds equation. In general, load support decreases with increasing texture length and increases with increasing texture depth except for the case of high texture depth with texture length of 0.25 mm. Under these conditions the support load decreases. Apart from that phenomenon, generally the high texture depth gives a positive effect on bearing performance. This is influenced by the higher cavitation region at large texture length. Conversely, a high texture depth actually makes the cavitation region smaller. This indicates that the occurrence of cavitation causes a negative effect on bearing performance. Thus, this study indicates that the maximum bearing performance corresponds to the smallest texture length configuration of 0.25 mm with low texture depth.

4. Conclusions

In the present study, the lubrication analysis of the single textured contact was solved by two approaches: the Navier-Stokes equation and the Reynolds equation. Based on the explanation discussed earlier, the conclusions that can be drawn are as follows:

1. Increasing texture depth leads to enhanced hydrodynamic pressure.
2. Increasing texture depth decreases the cavitation area.
3. Decreasing texture length decreases the cavitation area.

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