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# LOAD CARRYING CAPACITY OF A HETEROGENEOUS SURFACE BEARING*

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**Abstract**

It has been shown before that liquids can slip at a solid boundary, which prompted the idea that parallel-surface bearings can be obtained by the use of alternating slip and non-slip regions in the direction of fluid flow. The amount of slip at the wall depends upon the surface tension at the liquid - solid interface, which in turn depends on the chemical composition of the surface and its roughness. In the present study a heterogeneous surface was created by coating half of a circular glass disc with a coating repellent to glycerol. A rotating glass disc was placed at a known distance above the coated stationary surface and the gap was filled with glycerol. With the top surface moving in the direction of slip to non-slip region, a pressure build up can be theoretically predicted. The pressure gradient in the two regions is constant, similar to that in a Rayleigh step bearing, with the maximum pressure at the boundary. In order to accurately measure the force generated by the pressure increase, a load cell was attached to the heterogeneous disc. Experiments were conducted by varying both sliding speed and distance, and the resultant load carrying capacity was measured and compared with theoretical calculations. This allowed the slip coefficient of the coated surface to be evaluated.

**Keywords:** Bearing; Slip; heterogeneous; Load.

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# 1 INTRODUCTION

In lubricating systems, where the bounding solid surfaces are very close together and one of the dimensions of the fluid column is much smaller than the other two, a number of simplifying assumptions can be made which reduce Navier – Stokes equations to the form given by equation 1.

$$\frac{\partial p}{\partial x} = \frac{\partial \tau}{\partial z} \qquad (1)$$

It is assumed that the flow takes place along direction x and axis z is perpendicular to the bounding surfaces. Integrating twice with respect to *z* gives the velocity profile across the film thickness, with the approximation of two constants. Finding those constants and thus the full velocity profile can be done if some conditions of the interaction between the fluid and solid surfaces, at the two boundaries are assumed. For example in the classical case, analyzed by Reynolds, one surface is at rest (for example the lower surface) and the other moves at a known velocity *U*, thus the velocity profile becomes:

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

One of the hypotheses made in deriving this equation is that there is no slip between the fluid and the solid surfaces. This hypothesis is a cornerstone of lubrication and stays at the foundation of Reynolds equation for lubrication. Once the velocity profile is known the fluid flow can be derived and using the continuity of flow principle, the pressure gradient can be derived [1].

$$\frac{\partial p}{\partial x} = 6\eta U\left(\frac{h-\bar{h}}{h^3}\right) \qquad (3)$$

In this equation *U* is the speed of the sliding surface, $\eta$ is the viscosity of the fluid, *h* is the current separation between the solids and $\bar{h}$ is the separation at the position of maximum pressure. It is obvious that if the separation between surfaces is constant (surfaces are parallel) then $h = \bar{h}$ and no pressure is generated by the bearing. In other words, the load carrying capacity is zero in this case.
It has been found however, that the condition of no-slip at wall is not always fulfilled. Brochard and de Gennes [2] have shown that slip at the solid surface can occur in the case of polymer melts, when the shear stress near the wall exhibits a critical value. Leger *et al* [3] using near-field velocimetry proved experimentally the existence of slip between polymer melts and solid surfaces. The slip regime ensues above a critical slip velocity due to a progressive dynamic decoupling of the surface and the bulk chains of the polymer. The polymer melts studied in the previously mentioned papers are evidently non – Newtonian fluids. The question that researchers started asking was whether liquid slip at wall can be acheived in simple, Newtonian fluids, this can have important practical implications as recognized by Watanabe and Udagawa [4]. They observed an important reduction in drag of water flowing in a water-repellent pipe surface. They also find experimentally that the shear stress at the wall where slip occurs is proportional to the slip velocity. Pit *et al* [5], used an experimental technique of total internal reflection – fluorescence recovery after photobleaching (TIR-FRAP) to investigate the velocity of a simple, Newtonian fluid near a solid wall. The fluid tested was hexadecane and the solid boundary was treated with a classical organic friction modifier additive, stearic acid. Their tests demonstrated that simple Newtonian fluids can develop slip at the wall. Barrat and

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Bocquet [6] have also demonstrated, by using extensive molecular dynamics simulations that large slip lengths of about 30 molecular diameters are obtained if the contact angle of a liquid to a solid surface exceeds 140º. In a theoretical and experimental study Zhu and Granick [7] quantify the relative importance of molecular interactions and roughness upon the hydrodynamic force in a converging conjunction. They conclude that with very smooth surfaces the molecular interactions (liquid slip) dominates the behaviour of the bearing, however for asperities larger than about 6 nm RMS, the roughness dominates the behaviour.
The problem of slip at wall is important for the effect that this phenomenon may have upon the operation of sliding bearings. Exploiting the low shear stress at wall can result in bearings with lower friction thus better efficiency. The effect of slip upon the friction generated in a bearing was approached theoretically by Spikes who showed that bearings with half the friction of normal ones can be created by allowing slip to occur at one of the surfaces [8]. The concept of half – wetted bearing was later confirmed experimentally by Choo *et al* [9] who showed that friction reduction can indeed be achieved in a low-load bearing which has one of the surfaces treated as to slip against the lubricating fluid, as seen in Figure 1.

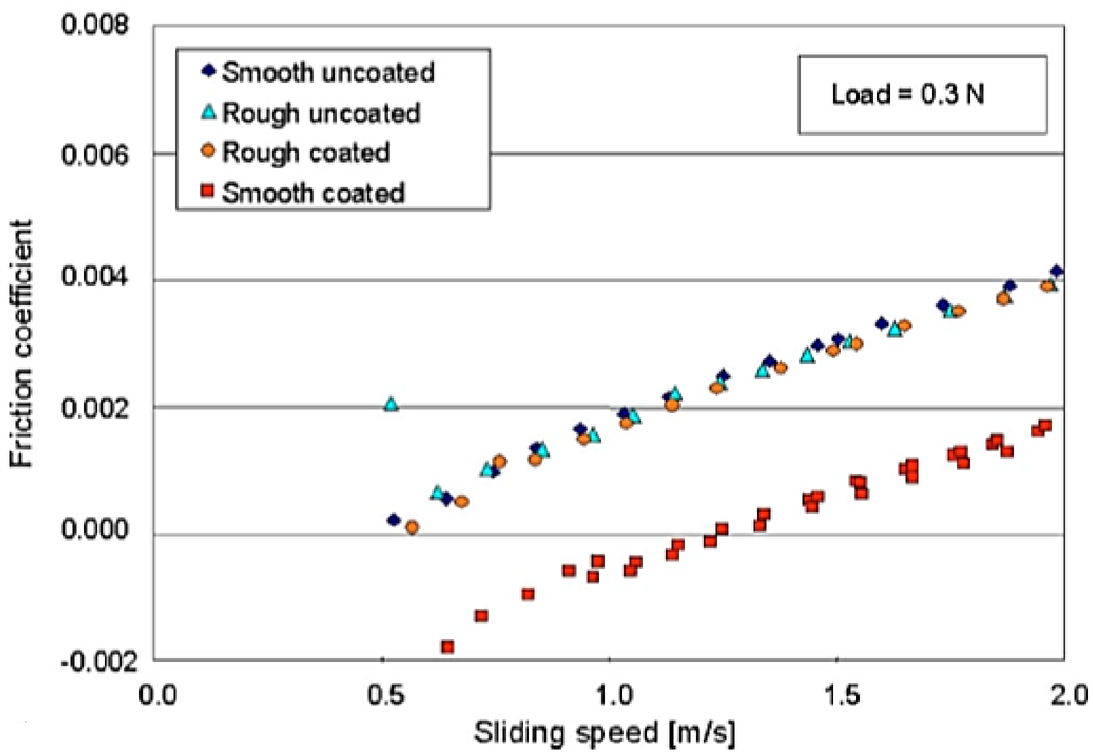

Figure 1. The effect of slip and roughness on friction coefficient [9]

In this work slip is defined by a two – component model; a critical shear stress, which if exceeded casues slip to occur between the fluid and the boundary. If shear stress is increased further the slip velocity increases in a linear fashion.
Significant friction reduction in a plane pad with regions of slip and non-slip was predicted theoretically by Salant and Fortier [10]. They evaluated the slip in terms of a critical shear stress, which, if exceeded caused the liquid to slip against the solid surface. They also define a slip velocity which is proportional to the shear stress through a dimensional factor of proportionality defined as slip coefficient. Their numerical simulations showed that not only a reduction in friction is obtained but also an increase of the load carrying capacity of the bearing.

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Reynolds' equation shows that a classical plane pad, with zero tilting angle cannot support a load, if thermal distortions are avoided. It has been shown however that when one of the surfaces has regions of both slip and non-slip, these act as geometrical discontinuities and create pressure gradients even when the surfaces are parallel. This feature was exploited by Takeuchi who tested a bearing featuring a heterogeneous surface of water repellent and non-water repellent regions [11].

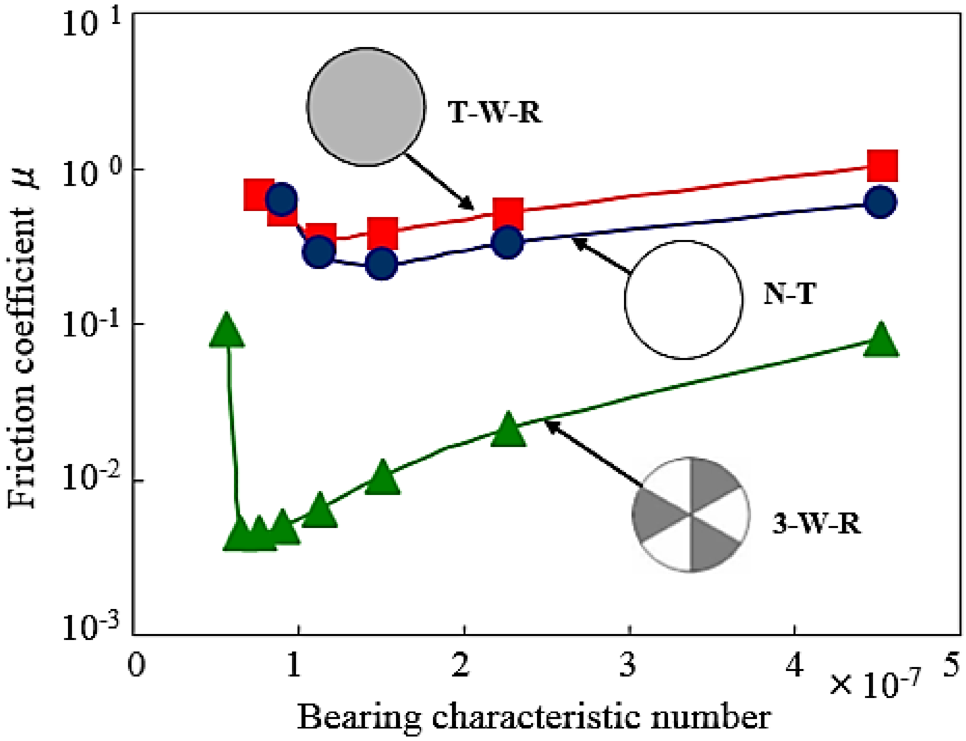

Figure 2. Friction reduction in a heterogeneous bearing [11]

He found a reduction of the friction coefficient by over one order of magnitude, as seen in Figure 2, which is an indication of the load carrying capacity of the bearing and the presence of a thick fluid film. Pascovici [12] analysed the load carrying capacity of a heterogeneous, slip/non slip pin sliding against a flat disc. He showed that a linear pressure variation can be obtained, similar to that found in step bearings if the fluid flows in the direction from the slip towards non slip region of the bearing. Experiments by Thomas *et al* [13] have confirmed this theoretical approach and found load carrying capacity forces of a magnitude similar to those predicted theoretically.

## 2 EXPERIMENTAL METHOD AND MATERIALS

### 2.1 Test Rig

A schematic of the test rig used in the present study is seen in Figure 3. The test specimens are a fixed glass pin (disc of 10 mm diameter, 5 mm thickness) and a rotating glass disc, 100 mm in diameter. The larger disc is fixed to a shaft which is attached to the end of a gearbox and receives motion from a DC electrical motor.

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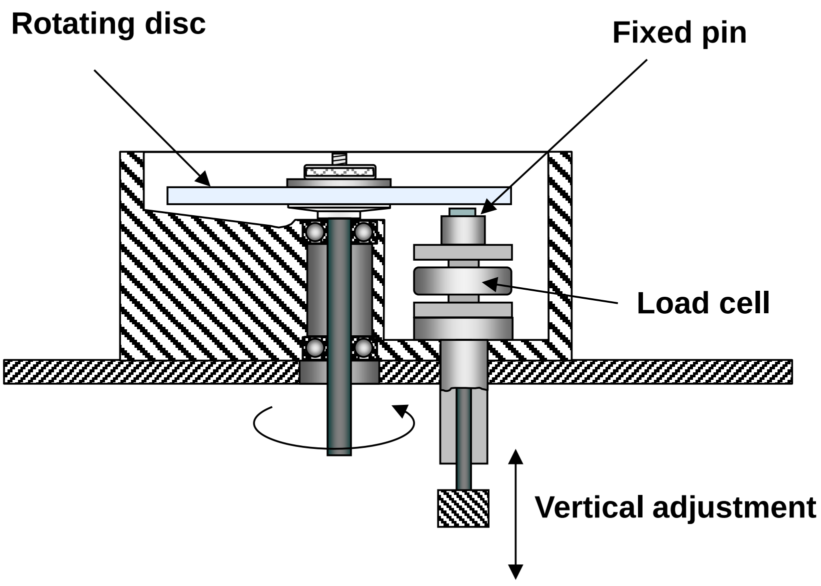


Figure 3. Schematic of the test rig

The fixed glass disc (pin) is supported in a holder, which in turn is attached to a push/pull load cell. The readings of the load cell are displayed on a digital panel meter. The load cell is calibrated prior to the tests and a reading versus load curve is constructed. The other end of the load cell is rigidly attached to a disc/plunger assembly free to move in a vertical direction, thus allowing the gap between the two specimens to be set. A lever and weights system, not shown in the picture, applies a force to the plunger such that the fixed and mobile specimens come into contact. Subsequently a micrometer is used to push the plunger, and the load cell/pin assembly downward thus setting the distance between the pin and rotating disc at a desired value. The precision of the micrometer is 5 microns.

## 2.2 Materials and Test Parameters

The pin has half of the flat surface coated, using a dip – coating method, with a octadecyltrichlorosilane (OTS) layer, while the other is left uncoated. The fluid used in this study was glycerol, with a viscosity of 0.632 Pa·s at test temperature. The contact angle at the interface between glycerol, OTS coating and air was between 100º and 110º while for the un-coated region about 15º - 20º. This creates a heterogeneous surface as the fluid wets the non – coated surface but slips against the OTS coating. Figure 4 shows images of two drops of glycerol on the bare galls and coated surfaces.

Figure 4. Contact angle of glycerol on two surfaces

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The roughness of the flat surfaces of the disc and pin was in the region of 10 – 12 nanometers $R_a$. No roughness measurement was carried out after the experiments, as it was assumed that because the gap between the two surfaces is fixed and thus the discs do not touch, there is no reason for the roughness to be altered during the tests. To be noted that the roughness stated for the pin was measured without coating. It was presumed that a stylus instrument for measuring the roughness would damage the coating and the laboratory does not have non – contact instruments for measuring roughness.
The kinematic condition in the gap between specimens was pure sliding, with the mobile disc rotating such that the sliding velocity was varied between 0.1 m/s and 2 m/s. The gap between the surfaces of the discs, in other words, the fluid film thickness was set to values between 25 and 250 microns. The same disc was used throughout all tests, following a thorough cleaning. Each result (data point) was an average of readings taken over a period of 60 seconds and the full experiment was repeated three times before the use of a new freshly coated pin.

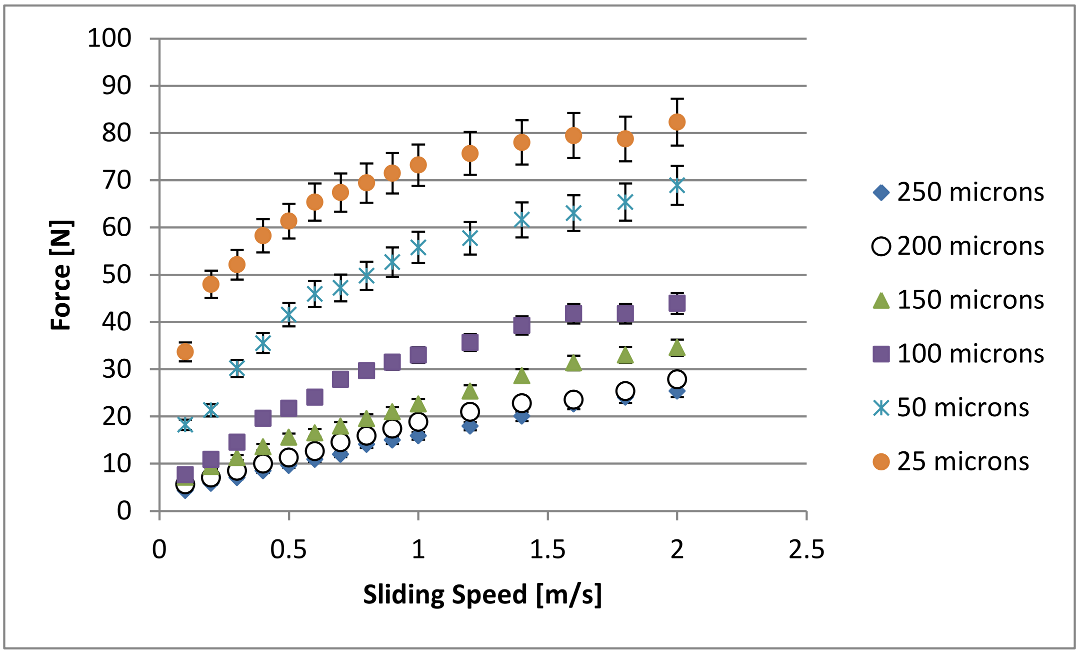

Figure 5. Load support function of sliding speed

## 3 RESULTS AND DISCUSSION

In this study the load support of the bearing formed by the un-coated glass surface sliding against the heterogeneous surface was measured and the results compared with theoretical values. Figure 5 shows the variation of the load carried function of the sliding speed, for various values of the film thickness. As seen the load carried by the bearing strongly depends of the gap between the two solid surfaces and of the sliding speed. The trend is consistent for the whole range of parameters tested.
As the film thickness increases the force carried decreases in a non-linear fashion as seen in Figure 6.

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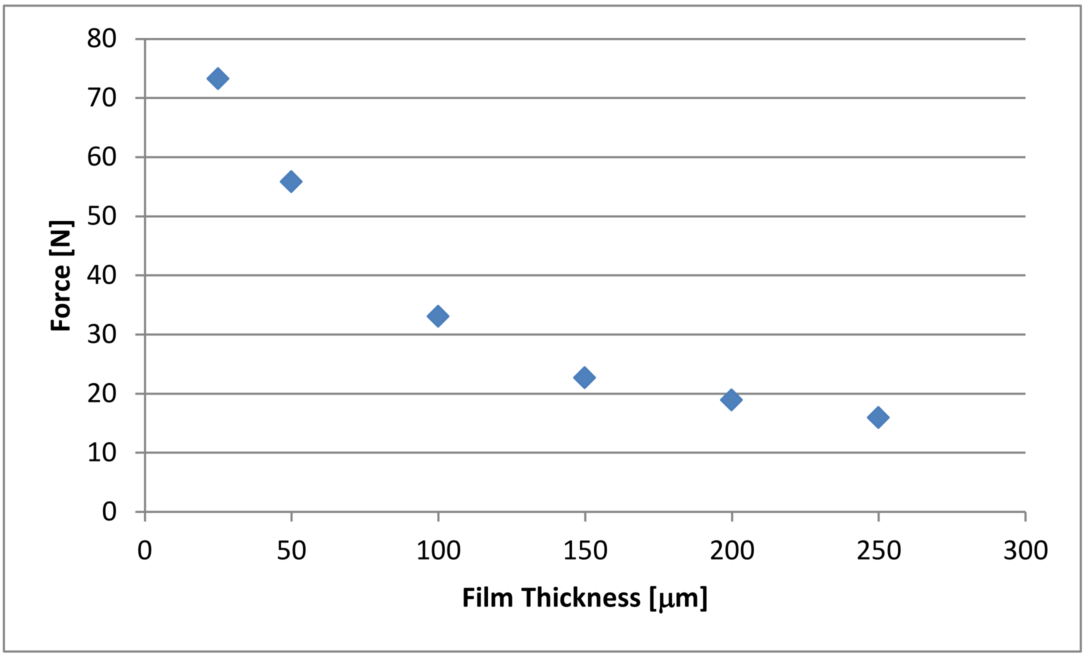

Figure 6. Load support function of film thickness

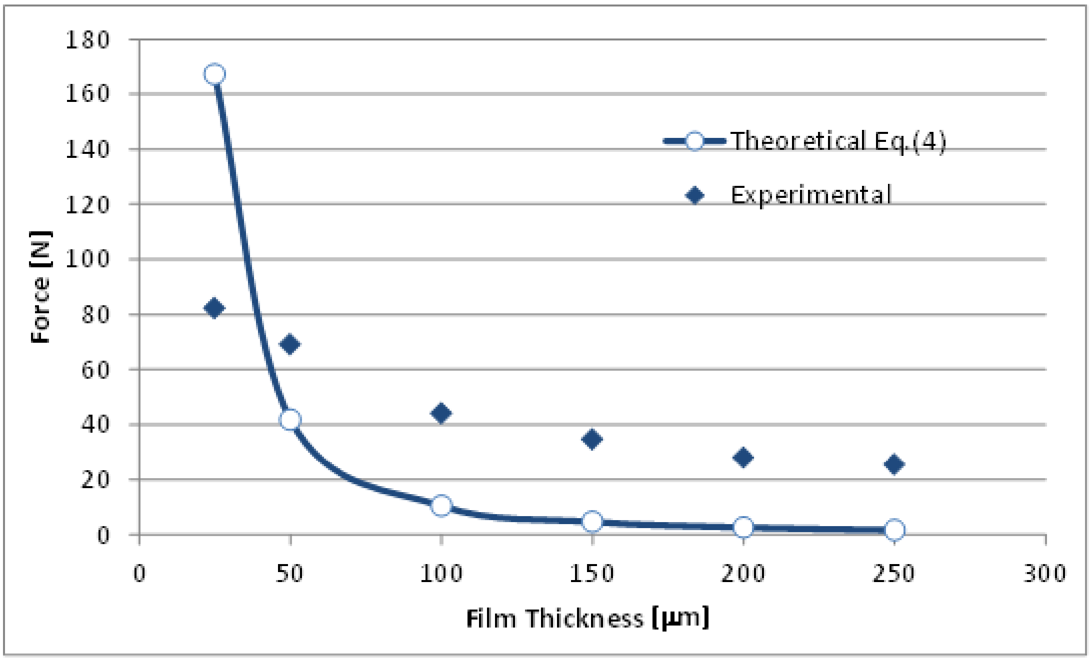

Figure 7. Comparison between experimental and theoretical results

A simple analysis, as carried in [12, 13] results in a relationship of the force which is proportional to $h^{-2}$ as given by relationship (4).

$$F = \frac{4(1-\alpha)U\eta R^3}{(5-3\alpha)h^2} \quad (4)$$

In this equation $\eta$ is the viscosity of the fluid, $R$ the radius of the pin, and $h$ the film thickness and $\alpha$ is the slip coefficient. If the slip coefficient is assumed constant then the variation of the force, function of the film thickness deviates strongly from the experimental trend, as seen in Figure 7.

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It is clear that the slip coefficient does not remain constant if the theoretical values were to fit the experimental results. Indeed, as shown by Brochardt and de Gennes [2], Craig *et al* [14] and Zhu and Granick [15], slip in systems with hydrophobic surfaces does depend on the shear rate, that is, on the film thickness. A quantitative analogy with those papers it is not intended to be carried out in this study, as the geometry and kinematics are different, but the dependence of the slip upon shear rate it is noted. If for example the slip coefficient depends on the shear rate ($U/h$) in such a way that the force resulted is overall proportional to $h^{-0.5}$ then the theoretical and experimental curves are very similar. This is illustrated in Figure 8, where the two curves are normalized by dividing by the largest value.

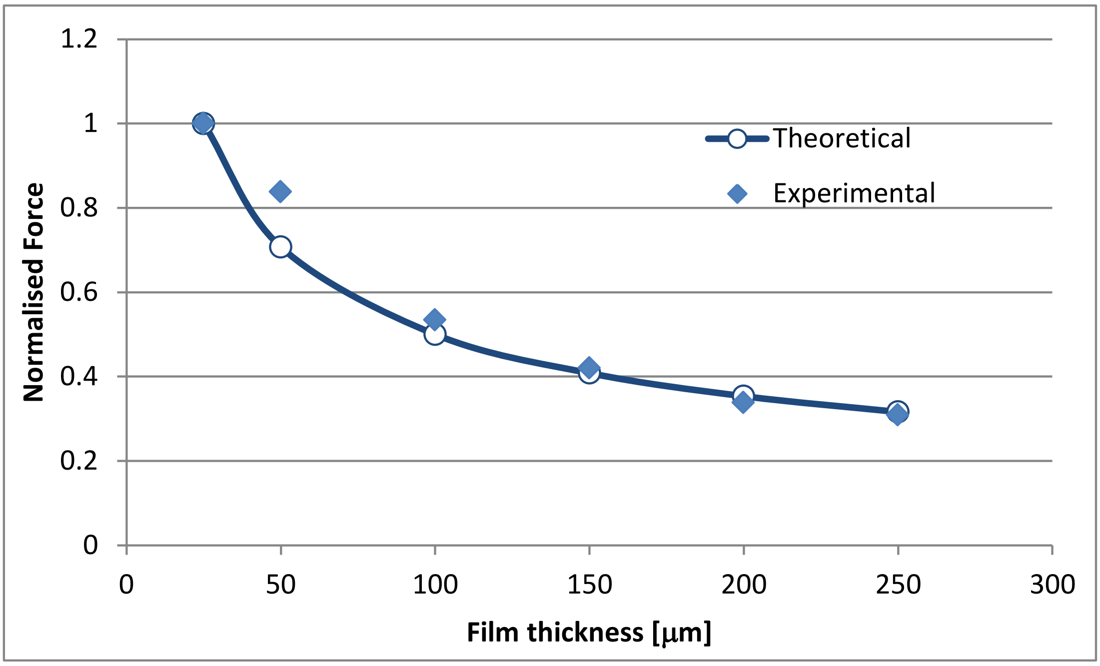


Figure 8. Comparison between experimental and theoretical force

The results motivate the need for a full numerical analysis of the fluid flow in this system, which could reveal the dependence of the slip coefficient on the shear rate and subsequently that of the load carrying capacity on the film thickness.

## 4 CONCLUSIONS

An experimental study on the load carrying capacity of a heterogeneous surface bearing has been carried out. A bearing system was obtained by sliding an untreated glass disc against a pin half coated with a self-assembled-monolayer of OTS, which is not wetted by glycerol, the fluid used in this study. The results showed that the bearing can carry considerable loads even if the solid – boundary surfaces are parallel. The force supported by the bearing was found to depend on the sliding velocity and the gap between the solid surfaces (that is the film thickness). Comparison with theoretical values obtained from a simple analysis showed values of the same order of magnitude, but of a different dependence of the load support of the film thickness. A full numerical analysis is required in order to reveal the relationship between the load carried by the bearing and the shear rate.

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