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RESEARCH ARTICLE

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-surfaces bearings can be achieved just by alternating slip and non-slip regions in the direction of fluid flow. The amount of slip at the wall depends on the surface tension at the liquid–solid interface, which in turn depends on the chemical state of the surface and its roughness. In the present study a heterogeneous surface was obtained by coating half of a circular glass disc with a coating repellant to glycerol. A rotating glass disc was placed at a known/calibrated distance and the gap was filled with glycerol. With the mobile surface moving from the direction of slip to non-slip region it can be theoretically shown that a pressure build up can be achieved. The pressure gradient in the two regions is constant, similar to that in a Rayleigh step bearing, with the maximum pressure at the separation line. The heterogeneous disc was placed on a holder supported by a load cell thus the force generated by this pressure increase can be measured accurately. Tests were carried out at different sliding speeds and gaps and the load carried was measured and subsequently compared with theoretical calculations. This allowed the slip coefficient to be evaluated.

Keywords: bearing; slip; heterogeneous; load

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 Eq. 1.

$$\frac{\partial p}{\partial x}=\frac{\partial \tau}{\partial z} \tag{1}$$

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 assumptions regarding the conditions of the interaction between the fluid and solid at the two boundaries are made. For example, in the classical case analyzed by Reynolds, one surface is at rest (e.g., the lower surface) and the other moves at a known velocity U, thus the velocity profile becomes:

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

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 remains 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].

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

In this equation U is the speed of the sliding surface, η 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 clear 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 is because it would have important practical implications, as recognized by Watanabe and Udagawa [4]. They observed a reduction of drag of water flowing on a water-repellent pipe surface. They also found experimentally that the shear stress at the wall where slip occurs is proportional to the slip velocity. Pit et al. [5] used an internal reflection–fluorescence recovery after photobleaching (TIR–FRAP) experimental technique 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 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 140°. In a theoretical and experimental study Zhu and Granick [7] quantifed the relative importance of molecular interactions and roughness upon the hydrodynamic force in a converging conjunction. They concluded that with very smooth surfaces the molecular interactions (liquid slip) dominate the behaviour of the bearing, however for asperities larger than about 6 nm RMS, the roughness dominate the behaviour. Spikes and Granick [8] derived equations for slip of a simple liquid, considering that slip occurs when the shear stress at the wall reaches a critical value. Bayada and Meurisse [9] focused their numerical analysis on the cavitation occuring at the slip/non-slip boundary showing the importance of the cavitation model upon the behaviour of a heterogeneous slip/non-slip bearing surface. Vinogradova [10] and Rohstein [11] published comprehensive up to date reviews of the slip at wall phenomenon, covering the fundamentals of the non-slip condition and the behaviour and applications of slip at hydrophobic surfaces.

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 and 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 [12]. The concept of half-wetted bearing was later confirmed experimentally by Choo et al. [13] 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 Fig. 1. In these experiments hexadecane was the chosen lubricating fluid. In this

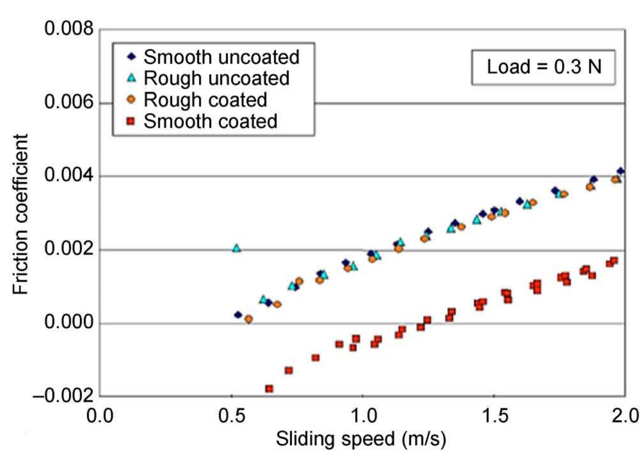

Fig. 1 The effect of slip and roughness on friction coefficient [13].

work slip is defined by a two-component model; a critical shear stress, which if exceeded causes 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 [14]. They evaluated the slip in terms of a critical shear stress, which if exceeded causes 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. Wu and Ma [15] carried out a numerical analisys of a hydrodynamic bearing with slip at one surface, pointing out the instability that slip may cause.

Fatu and co-workers [16] investigated numerically the effect of liquid slip in hydrodynamic bearings finding that the slip zone geometry must be carefully chosen, otherwise drastic reduction of bearing performance occurs, especially as the load carrying capacity is concerned. They also extend their analysis to highly-loaded, compliant bearings showing that slip/non-slip patterns can considerably improve bearing performance.

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 [17] who tested a bearing featuring a heterogeneous surface of water repellent and non-water repellent regions. He found a reduction of the friction coefficient by over one order of magnitude, as seen in Fig. 2, which is an indication of the load carrying capacity of the bearing and the presence of a thick fluid film.

Pascovici [18] analysed the load carrying capacity of a heterogeneous, slip/non-slip pin sliding aganst 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. [19] have confirmed this theoretical approach

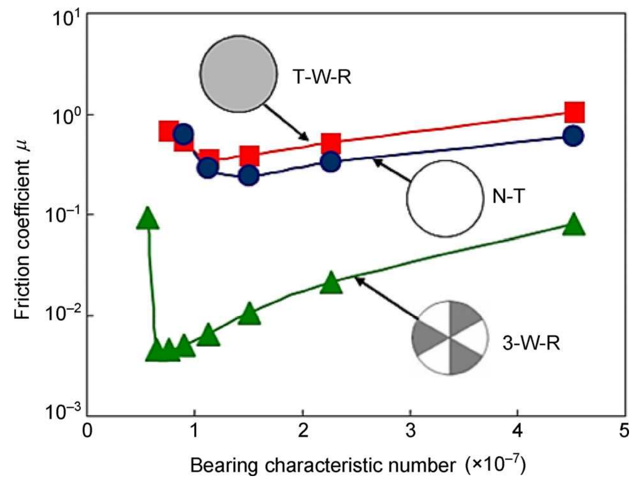

Fig. 2 Friction reduction in a heterogeneous bearing [17].

and found that heterogeneous surfaces are able to carry loads even if they are parallel. In the present paper this experimental approach is taken further and the load carrying capacity of the heterogenous slip/non-slip surface is measured for different speeds and separations between the solid surfaces.

2 Experimental method and materials

2.1 Test rig

A schematic of the test rig used in the present study is seen in Fig. 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.

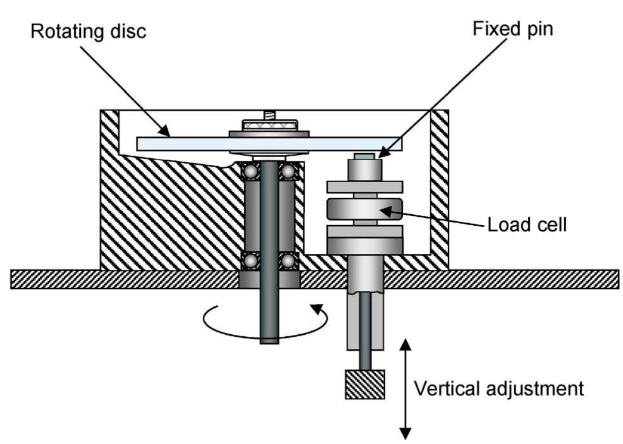

Fig. 3 Schematic of the test rig.

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 seen on a digital display. 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 vertical direction, thus allows setting the gap between the two specimens. 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 an octadecyltrichlorosilane (OTS) layer, while the other half is left uncoated. The fluid used in this study was glycerol, with a dynamic viscosity, at the room temperature, at which the tests were conducted of 0.632 Pa·s. The viscosity was measured before and after each test and no significant difference was found, which means that no water absorption took place during the tests. The viscosity of the lubricant, at a range of temperatures is shown in Table 1. The lubricant temperature was measured in the bath and no significant changes were observed during the tests. The temperature of the contacting surfaces was not measured in these experiments.

The contact angle at the interface between glycerol, OTS coating and air was between 100° and 110° while for the un-coated region about 15°–20°. 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 glass and coated surfaces.

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,

Table 1 Lubricants' properties.

Lubricant	Viscosity at 30 °C (Pa·s)	Viscosity at 40 °C (Pa·s)	Viscosity at 100 °C (Pa·s)
Glycerol	0.612	0.283	0.153

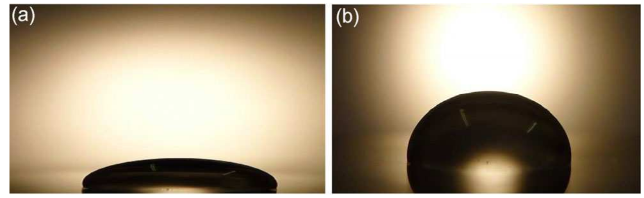

Fig. 4 Contact angle of glycerol on two surfaces; (a) uncoated and (b) coated.

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. The roughness of coated surfaces was not measured as non-contact instruments were not available at the time and use of a stylus instrument would risk damaging the coating. 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.

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. The force values shown were calculated from the measured data for a heterogeneous pin surface by subtracting the force obtained for a non-coated pin (at the same speed).

As seen load carried by the bearing strongly depends on the gap between the two solid surfaces and on 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 Fig. 6.

A simple, qualitative analysis in which there is sliding between the fluid and the fixed surface gives the velocity profile and the load carried by the bearing. Going back to Eq. 2 it is now assumed that there is slip between the stationary surface and the fluid such

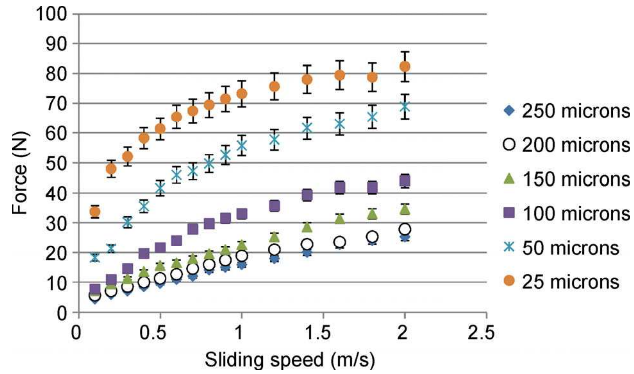

Fig. 5 Load support function of sliding speed.

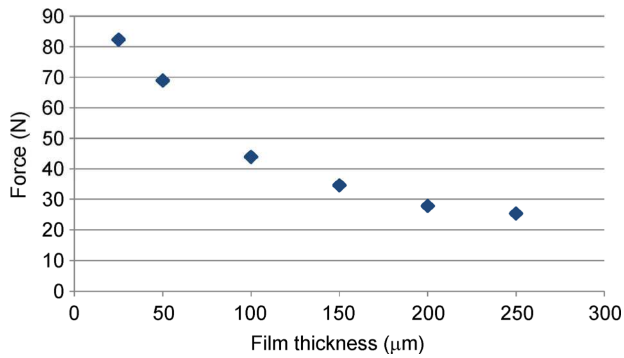

Fig. 6 Load support function of film thickness at 2 m/s sliding speed.

that the shear stress at this surface is a fraction α of the shear stress in the absence of slip. With this the velocity profile is given by:

$$u = \frac{1}{2\eta}\frac{\partial p}{\partial x}\left[z^2 - \alpha hz + h^2(\alpha - 1)\right] + \alpha U\frac{z}{h} - U(\alpha - 1) \quad (4)$$

The slip coefficient α takes values between 0 (total slip) and 1 (non-slip). It follows that for the non-slip condition Eq. 4 reduces to Eq. 2.

For the surface of the fixed pin in these experiments there is clearly a discontinuity at the separation between the coated (partial slip) and uncoated (non-slip) regions. By analogy to the pressure distribution in a step bearing, it is assumed a linear variation of the pressure in each of the two regions. This makes the pressure gradient p_{max}/R in the entry region (partial slip) and $-p_{max}/R$ in the exit region (non-slip). By writing the condition of continuity of flow at the separation, the maximum pressure can be found. Subsequently, it is assumed that the pressure distribution is conical over the whole area of the pin thus the load carried by the bearing can be obtained. This results in a relationship of the force which is proportional to h^{-2} as given by Eq. (5). It can be seen that when $\alpha = 1$ (non-slip condition) the load carried becomes zero.

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

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 Fig. 7. In this figure a value of 0.3 for parameter α was chosen.

A good fit is not to be expected as Eq. 5 was derived ignoring the flow perpendicular to the direction of sliding and the circular shape of the pin, however it is clear that the slip coefficient cannot be constant if the theoretical values were to fit better with the experimental results. Indeed as shown by Brochard and de Gennes [2], Craig et al. [20] and Zhu and Granick [21], slip in systems with hydrophobic surfaces does depend on the shear rate that is, on the film thickness. Due to difference in geometry and kinematics it is not intended to carry out a quantitative comparison with those studies, however a similar dependence of the slip upon shear rate 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 Fig. 8, where the two curves are normalized by dividing by the largest value.

This result prompts to a numerical analysis of the fluid flow in this system, which could reveal the dependence of the slip coefficient on the shear rate

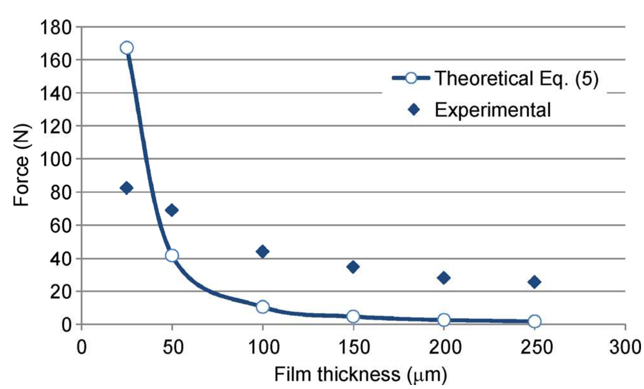

Fig. 7 Comparison between experimental and theoretical results at 2 m/s sliding speed.

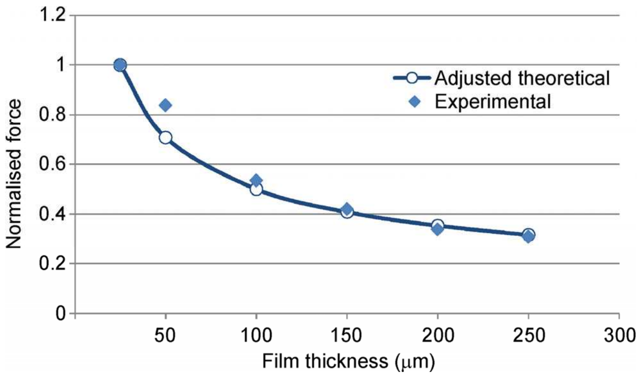

Fig. 8 Adjusted theoretical force which best fit the experiment.

and subsequently that of the force support on the film thickness however this is not the objective of this paper.

4 Conclusions

A novel experimental study on the load carrying capacity of a heterogeneous surface bearing has been performed. A bearing system was obtained by sliding an untreated glass disc against a pin half coated with a layer 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 results 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.

Nomenclature

F – Load carried by the bearing
h – Separation between surfaces (lubricant film thickness)
p – Pressure
R – Radius of pin specimen
U – Velocity of the moving surface (sliding speed)
x, z – Coordinates (x in flow direction)
α – Slip coefficient
η – Lubricant viscosity
τ – Shear stress

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