

Fluid distribution in a friction gap during mixed lubrication

Timo Fräger[✉], Cassandra Kampen[✉], Michael Müller[✉]

Institute for Acoustics and Dynamics, Technische Universität Braunschweig, 38106 Braunschweig, Germany.

✉ Corresponding author. E-mail: t.fraeger@tu-bs.de (Timo Fräger), mi.mueller@tu-bs.de (Michael Müller)

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Abstract:

To enable a robust measurement of the lubricant distribution between two rough topographies under starved regimes, measurements on a scaled friction gap with transparent topographies and coloured fluid were carried out. Therefore, the disc-to-disc tribometer “FLUCS” was developed, as a further development of the Wear Debris Investigator. Highspeed videos through the transparent topography are used to calculate the fluid height from the colour intensity of the fluid. In order to quantify the share of solid and fluidic friction, the shear area of the fluid is determined quasi in-situ. For this purpose,

tracer particles were added to the fluid and its flow was analysed using particle tracking velocimetry.

By varying the amount of fluid, the influence of the friction regime on the fluid distribution, fluid height, gap height, coefficient of friction and lubricant flow was analysed. The results show the formation of characteristic fluid lines, which are favored by high viscosities, relative velocities and topographic roughnesses. The measured coefficient of friction over the fluid quantity shows a Stribeck-like character and is interpreted in terms of the friction regimes.

Keywords:

Mixed lubrication, disc-to-disc tribometer, particle tracking velocimetry (PTV), lubricant distribution, gap height, Stribeck curve

1 Introduction

In technical contacts the coefficient of friction varies greatly with the process parameters, which include the normal force, relative speed, viscosity of the lubricant and surface geometry – this relationship can be represented in the so-called Stribeck curve [1, 2]. Depending on the process parameters, either boundary friction, mixed friction or (elasto-)hydrodynamic friction occurs. A further influencing factor is the amount of fluid in the gap, which is orthogonal from the process parameters [3, 4]. While a gap without fluid supply obviously runs under dry conditions, a hydrodynamic state can be achieved with sufficient lubricant supply. The coefficient of friction in dependence of fluid amount also exhibits a Stribeck-like character [3–5] as schematically illustrated in Fig. 1. The optimal range, in which friction is low, lies in the range of mixed lubrication. The minimum coefficient of friction was found in a fill level range of 10...20 % [3–6], whereby the fill level corresponds to the ratio of the fluid volume to the gap volume. Previous studies have shown that not only the fluid amount but also the fluid distribution in the friction gap has a decisive influence on the coefficient of friction

[3, 4]. Especially in the area of mixed lubrication, the fluid is distributed unevenly in the friction gap [3] and the shear area of the lubricant no longer corresponds to the surface area.

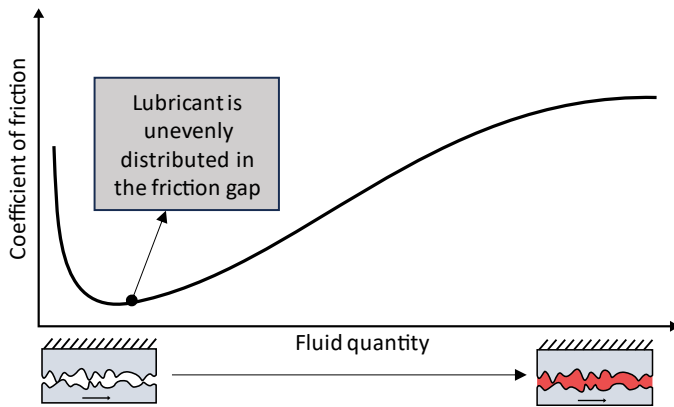


Fig. 1 Stribeck-like dependence of the coefficient of friction on fluid quantity

In these terms, there is still a lack of experimental work that investigates the fluid distribution between rough surfaces under starved conditions. Experimental measurements of the fluid distribution are difficult because the lubricating film thicknesses of tribological contacts are in the sub-nanometer range [7] and the topographies optically obscure the actual contact. The first measurements of lubricant film thickness were therefore based on electrical methods [8, 9]. By measuring the electrical resistance [10, 11] or the voltage drop [12] between the topographies, a rough estimate can be made as to whether the plates are in contact or separated by a lubricating film. Since the resistance also depends on the temperature and calibration is complicated, it is difficult to determine the film thickness quantitatively [11]. Using electrical capacitance measurements, the average film thickness can be determined quantitatively [13]. The disadvantage of this method is that it is intrusive, as electrodes have to be added [7]. Methods have also been developed to determine the lubricating film thickness using ultrasound; however, these can lead to measurement errors under mixed lubrication conditions [14]. For this reason, optical methods have been developed to measure the fluid distribution using a

transparent topography. A widely used optical method is interferometry [15, 16]. The idea is to illuminate the contact with a monochromatic light beam. One part reflects at the glass surface, another at the second topography, and the lubricating film thickness is determined from the interference. The limitation of this method is that measurements on rough surfaces are not possible [7, 13]. Another optical method involves the use of a fluorescent fluid, the laser induced fluorescence (LIF) [17–21]. By measuring the emitted radiation this method can also be used to determine the spatially resolved thickness of the lubricating film [22–25].

These methods are used, for example, in the investigation of piston ring-liner contacts [20, 21]. According to Obert et al., the thickness and distribution of the oil lubricant film in such a contact depend crucially on the supply of lubricant [20]. An insufficient quantity of lubricant led to starved lubrication and a different distribution of the lubricant within the contact. Wigger et al. observed a dependency of the fluid distribution within different friction regimes too. There the friction gap of a piston ring-liner contact is only partly filled with lubricant, distributed in fluid-lines in direction of topographic movement [21]. Such path-like fluid distributions could also be observed in the ball-on-ring contact of ball bearings at high speeds [26].

In previous studies, therefore a disc-to-disc-tribometer with transparent topographies and macroscopic roughnesses was used to investigate the fluid distribution within different friction regimes [3, 4]. The macroscopic roughnesses of the topographies were designed to represent a microscopic friction gap on an enlarged scale (see section 2.1). It has been shown that the scaled macroscopic friction gap reproduces essential microscopic phenomena and that the flow behaviour is comparable with the microscopic scale [3]. Accordingly it was also observed that, in the regime of mixed lubrication, the friction gap was only partly filled, and that path-like fluid distributions were present in this system.

The present paper attempts to identify the conditions under which these fluid lines form and how the fluid distribution influences friction. Therefore the disc-to-disc tribometer “FLUCS” was developed. Using transparent glass discs with macroscopic roughness and coloured lubricant, the fluid distribution

is calculated quantitatively based on the colour intensity. In order to quantify the shear area of the fluid in the partly filled friction gap, tracer particles are added to the fluid in a second step and the flow is quantified using particle tracking velocimetry. This allows a distinction to be made between areas in which fluid is sheared and areas in which fluid is deposited on the topographies. The aim is to gain a basic understanding of how fluid is distributed in a representative friction gap with rough topographies, what this distribution depends on and how it influences friction.

2 Material and Methods

2.1 Scaled disc-to-disc tribometer

The FLUId Contact triboScope (FLUCS) is a disc-to-disc tribometer whose contact surfaces consist of two ornamental glass discs (Fig. 2). This is a further development of the Wear Debris Investigator, which was presented in [3, 27]. The ornamental glass discs have roughnesses in the order of several tens to some hundreds of micrometres and represent a geometrically scaled surface topography of technical contacts, whose roughnesses are usually a few micrometres. The scaled roughness not only makes individual asperities more easily visible, but the hard glass discs also mean that topographical changes caused by wear can be disregarded when normal forces are low. A detailed discussion of scalability was provided in [6, 21] and is not the subject of this paper. The lower glass disc is rotated by a gear motor and fixed vertically, while the upper glass disc is fixed rotationally but can move vertically (Fig. 4). Vertical forces of between 0 and 120 N can be applied to the upper glass disc, creating a normal force on the friction gap. The upper glass disc is not completely rigidly supported, but can tilt slightly relative to the lower glass disc due to an elastomer coupling. A sensor measures the torque and the normal force acting on the upper disc. As the ornamental glass discs are transparent, the friction gap is visible through a 45° mirror and can be recorded with a camera. The lower disc is illuminated with a ring light and a dark room around the entire test stand ensures uniform light distribution that is independent from ambient light.

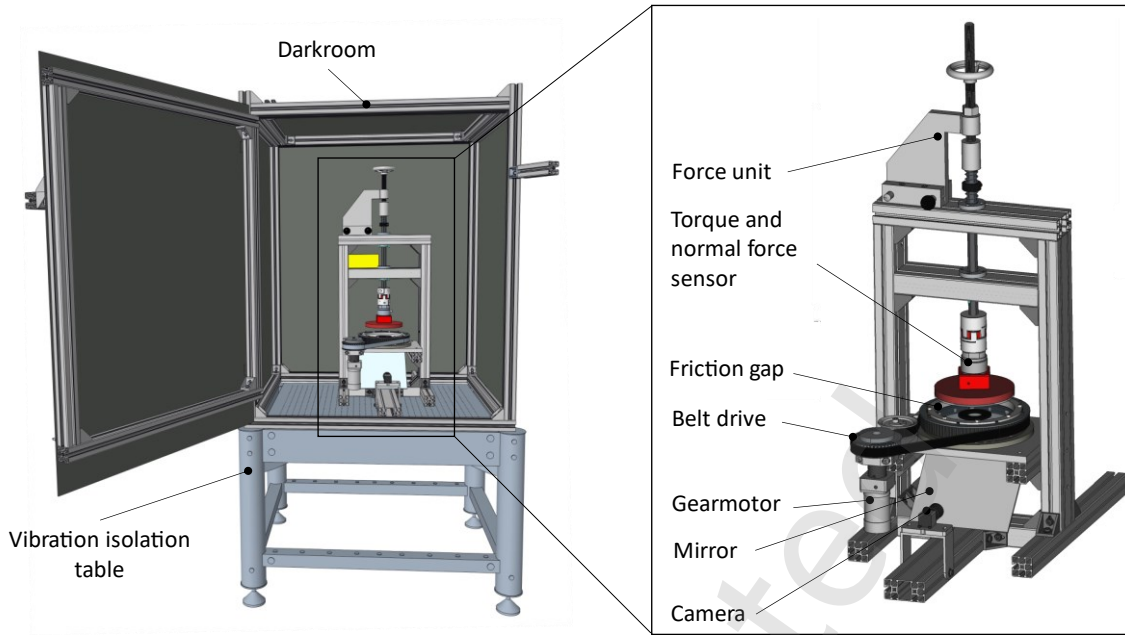


Fig. 2 Experimental setup of the FLUId Contact triboScope (FLUCS)

Two ornamental glass discs with the trade name ORNAMENT 504 were used for the friction tests (Fig. 3). The mean roughness is $18.87 \mu\text{m}$, the square roughness is $23.35 \mu\text{m}$ and the maximum roughness depth is around $130 \mu\text{m}$ (see Fig. 3c). The lateral distance between local profile asperities is in the range between 1 mm and 2 mm . The upper glass disc has a white coating on the top.

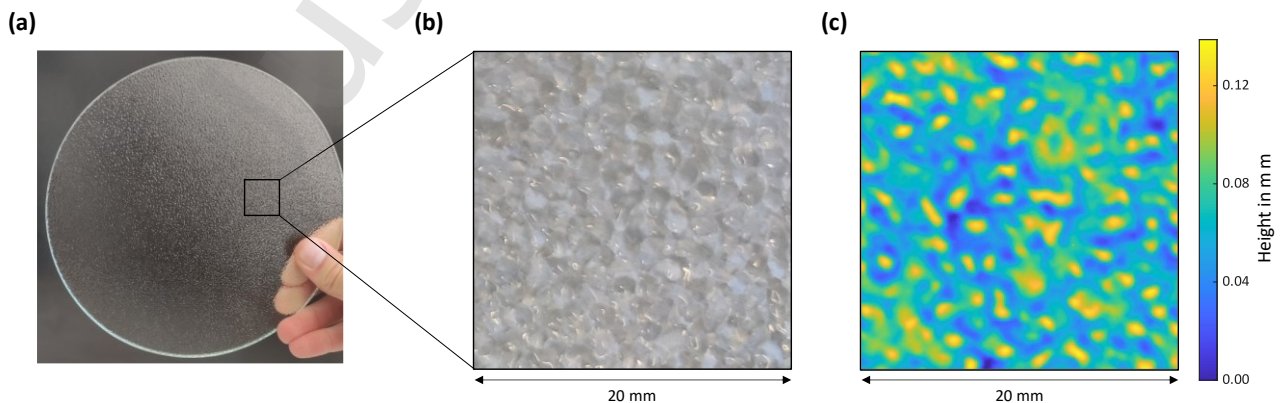


Fig. 3 Ornamental glass disc: (a) image of the entire disc, (b) section of 20 mm, (c) height profile

The contact surface has a diameter of 168 mm and glycerin as a rather high-viscous fluid is applied as a lubricant between the ornamental glass discs. Glycerin has a dynamic viscosity of $\eta \approx 1.5 \text{ Pa} \cdot \text{s}$ at 20°C [28], which is mixed with 2.5 weight % natural red dye ("PureRubin" by Trotec GmbH) to make the fluid visible. As the viscosity of glycerine is highly temperature-dependent, additional tests were carried out using temperature sensors in the friction gap of the FLUCS. They showed, that the temperature did not change during the friction test as a result of friction. The tests were conducted at a temperature in a range of 19 to 21 °C.

Fig. 4 shows the configuration of the glass discs schematically. During the experiment, the high-speed camera films a section of the glass discs with a width of $b_x = 0.047 \text{ m}$ and a height of $b_y = 0.035 \text{ m}$. The section is at a distance of $b_{\Delta y} = 0.035 \text{ m}$ from the center of the discs in the y-direction, see Fig. 4b. In a second step, tracer particles are added to the glycerin to determine the shear area of the fluid (see section 2.4 for details). Using the high-speed camera with a frame rate of 200 fps and a resolution of 640x480 pixels, the particles are filmed and their movements are detected and analysed using particle tracking velocimetry (PTV).

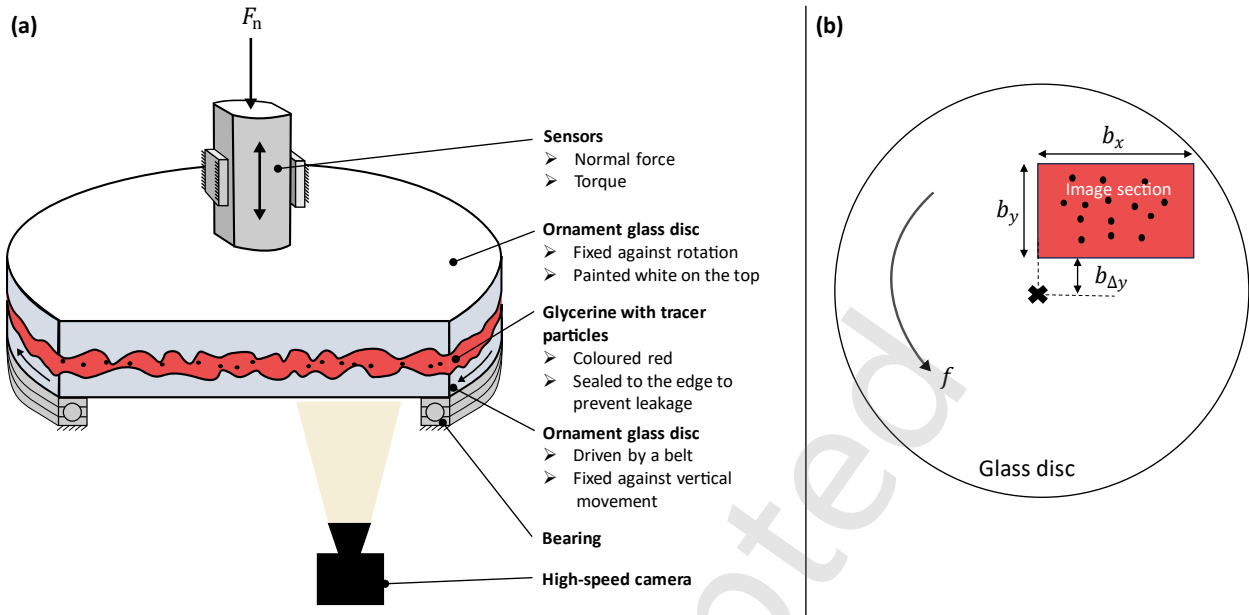


Fig. 4 Schematic sketch of the experimental setup: (a) Top view, (b) perspective of the high speed camera with the image section

2.2 Test procedure

In the experiment, the friction regime is varied by changing the amount of fluid. The test is carried out first with pure glycerine without tracer particles to determine the lubricant distribution and afterwards with tracer particles to quantify the fluid movement. The test procedure for a measurement is as follows: For the test with tracer particles, 0.1 g of particles is first weighed using a precision balance and distributed evenly on the lower glass disc. Then the second glass disc is placed on top and the measurement is started. At the start of a measurement, a run-in process is carried out for 20 seconds at a speed of 30 rpm to ensure even distribution of the particles in contact. Then the speeds are increased successively from low to high for a few seconds at a time (at least one full revolution) (**Fig. 5**). During this process, the normal force and torque are measured and the friction gap is recorded with the camera.

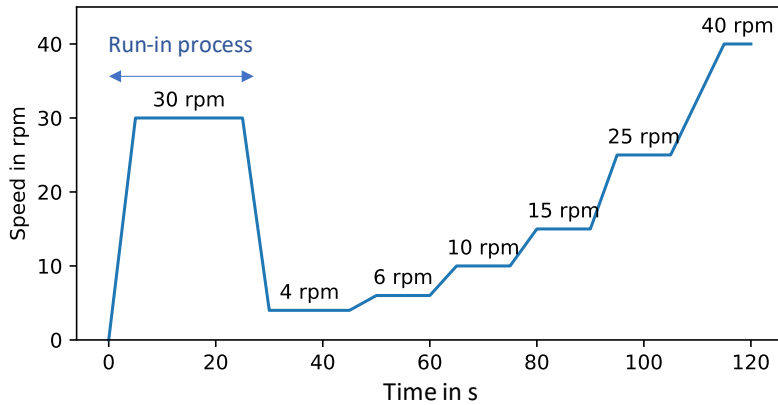


Fig. 5 Nominal speed over time for one measurement

Once all speeds have been reached, the measurement is complete. The upper glass disc is removed, new fluid (glycerin) is added with a pipette and the measurement is restarted. To achieve friction regimes from dry to purely fluid friction, the fluid quantity is varied from $0 \mu\text{l}$ to $4500 \mu\text{l}$, where the gap is completely filled with fluid (full filling). Since the coefficient of friction varies greatly with the amount of fluid in the transition between dry friction and mixed friction, a high resolution of the fluid quantity is used in this regime. Once the measurement has been repeated for all desired fluid quantities, the measurement series is complete.

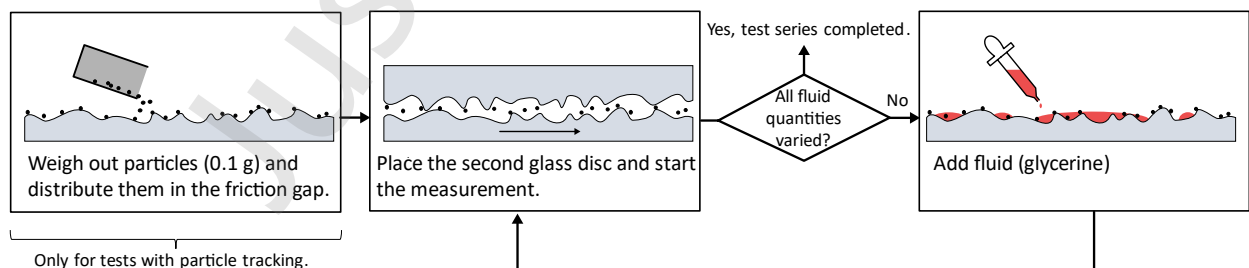


Fig. 6 Test procedure

For evaluation, the torque at each speed step is averaged over time. The coefficient of friction is defined as $\mu = \frac{\tau}{\sigma}$ and cannot be directly calculated from the torque. In order to calculate a parameter which is similar to the coefficient of friction, following assumptions are made:

1. The shear stress is proportional to the radius r from the centre of rotation: $\tau = C r$
2. The normal stress σ is distributed homogeneously across the disc surface A_{ges} : $\sigma = \frac{F_n}{A_{ges}}$

Integrating the friction torque over the disc surface results in $C = \frac{2M_R}{\pi R^4}$ where M_R is the measured torque and R is the disc radius. The coefficient of friction is then given by:

$$\mu = \frac{2M_R}{F_n R^2} r. \quad (2.1)$$

And averaged over the friction surface:

$$\bar{\mu} = \frac{4}{3} \frac{M_R}{R F_n}. \quad (2.2)$$

2.3 Fluid distribution deduced by the color value

The aim is to calculate the distribution of the fluid height from the colour intensity of the coloured lubricant. The intensity of a light beam after passing through a fluid depends exponentially on the height of the fluid film. The relationship can be described by the Lambert-Beer law [29]:

$$I = I_0 e^{-k h_{\text{Fluid}}}, \quad (2.3)$$

where I is the light intensity after passing through the fluid height h_{Fluid} , I_0 is the light intensity without fluid and k is the extinction coefficient, a material parameter that depends on the wavelength of the passing light (**Fig. 7**) [29].

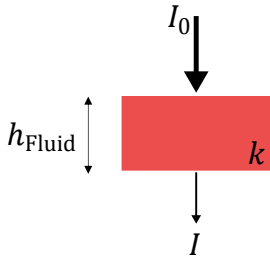


Fig. 7 Intensity of a light beam I after passing through the fluid height h_{Fluid}

The coloured glycerine particularly influences the green value - the green colour channel of the image G is therefore evaluated to determine the fluid height. The friction gap in the FLUCS is illuminated from the side by a ring light. While part of the light is reflected directly off the lower glass disc, the remaining light passes through the fluid in the friction gap and is reflected off the upper white-painted glass disc. According to the Lambert-Beer law, it is assumed that the intensity of the green value g of the reflected light, after passing through the fluid film, depends exponentially on the fluid height. The reflections from the lower glass disc, on the other hand, are accounted for by a constant shift g_1 in the green value (Fig. 8):

$$g = g_0 e^{-k h_{\text{Fluid}}} + g_1 . \quad (2.4)$$

This assumption neglects reflections from the rough underside of the upper glass disc. Although these reflections are significantly lower than those from the white upper surface, they nevertheless cause light to scatter, resulting in lateral blurring of the measurement.

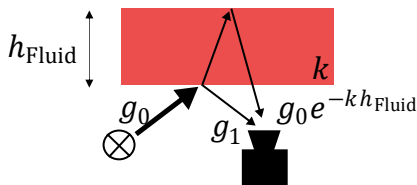


Fig. 8 Green value of a light beam with lateral illumination in FLUCS

According to these assumptions, the fluid height can be deduced from the green value if the absorption coefficient is known:

$$h_{\text{Fluid}} = -\frac{1}{k} \ln \left(\frac{g - g_1}{g_0} \right). \quad (2.5)$$

Here, the absorption coefficient is no longer characterised by the fluid but must be calibrated for the given measurement setup. In order to obtain the fluid height from the green colour channel of the image, the coefficients g_0 , k and g_1 are calibrated as described in the following.

Calibration

The coefficients are not constant across the image due to uneven lighting, camera errors and refraction of the rough glass surfaces. Therefore, to determine these coefficients, reference images with known fluid heights are taken starting from a defined image section (**Fig. 9**). The coefficients for each pixel at the position x, y in the image are then fitted with the least square method using the green color channel for the known fluid heights of the reference images:

$$G(x, y) = g_0(x, y)e^{-k(x, y)h_{\text{Fluid}}} + g_1(x, y) \quad (2.6)$$

with $x \in [1, 480] \cap \mathbb{N}$ and $y \in [1, 640] \cap \mathbb{N}$.

Then for each pixel in the image plane one parameter set $[g_0, g_1, k]$ was calculated. To overcome individual optical errors, these parameters are fitted with two-dimensional polynomial in x and y direction of the image plane:

$$\begin{aligned} g_0(x, y) &= \sum_{i=0}^3 \sum_{j=0}^{3-i} g_{0ij} \cdot x^i y^j, \\ g_1(x, y) &= \sum_{i=0}^3 \sum_{j=0}^{3-i} g_{1ij} \cdot x^i y^j, \\ k(x, y) &= \sum_{i=0}^3 \sum_{j=0}^{3-i} k_{ij} \cdot x^i y^j. \end{aligned} \quad (2.7)$$

With these one-time determined polynomials, the fluid height for this particular image section at a position x, y can be calculated using equation (2.6). This procedure can be understood as a fluid-height-dependent flat-field correction.

Images with two smooth glass discs were used as the reference images. With spacers, different defined constant fluid heights of $h_{\text{Fluid}} = (0 ; 0.25 ; 0.30 ; 0.55 ; 0.63)$ mm were set (Fig. 9).

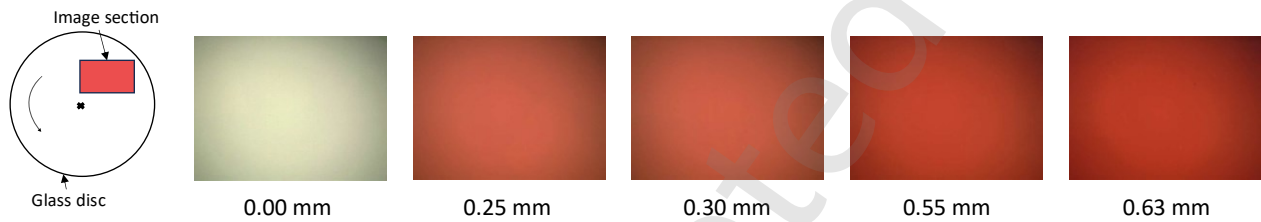


Fig. 9 Image section at different fluid heights with smooth glass-discs

The images with $h_{\text{Fluid}} = (0 ; 0.30 ; 0.63)$ mm were used to fit the coefficients of the bivariate polynomials (verification images). The images with $h_{\text{Fluid}} = (0.25 ; 0.55)$ mm are used for validation (validation images). The fluid heights in the friction tests are below 0.63 mm.

Verification and validation

To examine the suitability of this method, the fluid height of all reference images was calculated using equation (2.6). The calculated fluid distributions are flat and hardly fall off towards the image corners (Fig. 10). Vignetting was therefore well corrected in the reference images. The planes are also horizontal, which indicates that the lighting errors have been properly corrected. Qualitatively, the method is therefore suitable for determining the fluid distribution.

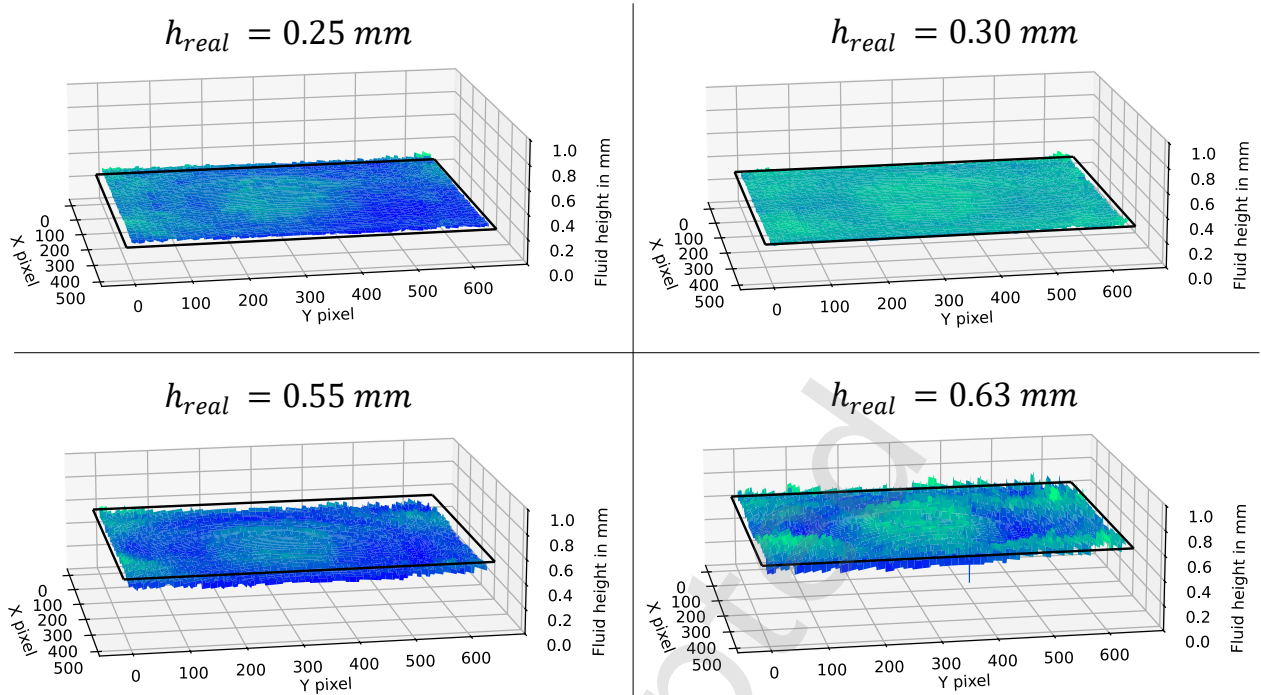


Fig. 10 Calculated height profile of the reference images with defined actual fluid height h_{real} (black frame)

The mean calculated fluid heights compared to the real fluid heights are shown in **Table 1**. The deviation in the verification images is negligible. The calculated fluid heights of the validation images deviate a good 10 % from the real heights. The standard deviation is more than order of magnitude below the real fluid height.

Table 1 Mean calculated fluid height compared to the real fluid height

Real fluid height	Calculated fluid height	
	arithmetic mean	standard deviation
0.00 mm	0.005 mm	0.0015 mm
0.25 mm	0.285 mm	0.0096 mm
0.30 mm	0.300 mm	0.0077 mm
0.55 mm	0.491 mm	0.0205 mm
0.63 mm	0.630 mm	0.0340 mm

In order to better identify areas where there is no fluid in the height profiles, it has proven useful to set calculated liquid heights of less than 0.055 mm to zero in the graphs of the results. The fluid distribution in this study is calculated and averaged from all images taken during a velocity interval. It is also important for the measurements that the colour of the fluid remains stable over time and that the absorption properties do not change either during the course of a single measurement or between recorded measurement series. The colour values during three measurement repetitions with a fully filled friction gap over time are shown in **Fig. 11**. The colour values remain constant both over the course of a single measurement and between measurement repetitions. Accordingly, a single calibration is sufficient for one measurement series.

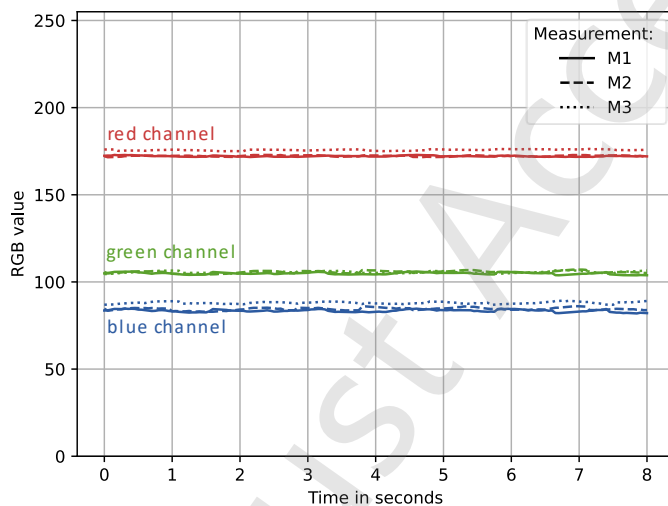


Fig. 11 RGB values for three measurement repetitions over time, with a fully filled friction gap and a rotational speed of 15 *rpm*

2.4 Fluid movement detection based on particle tracking

In the regime of mixed lubrication, the normal force is carried partly by solid-state contacts between the asperities and partly by the lubricating fluid. Accordingly, the frictional force also arises partly by

the shearing of the fluid. The magnitude of this shear force depends not only on the shear rate but also on the shear area.

In order to quantify the fluid movement and the shear area, tracer particles are added to the fluid in a second experiment. For this purpose, commercially available glitter platelets of hexagonal shape with an outer diameter of around 250 μm and a thickness of around 60 μm (Fig. 12) are used (Manufacturer: Max Bringmann KG, Glitter Set No. 578).

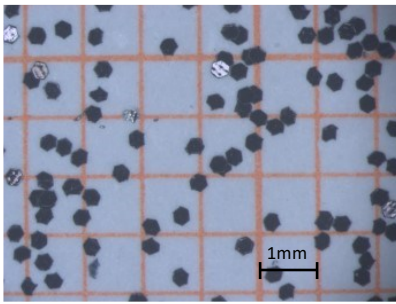


Fig. 12 Microscope image of the tracer particles

To quantify the particle movement, particle tracking velocimetry (PTV) is used with the Python library Trackpy v0.6.4 [30, 31]. In PTV, particle positions are detected in each frame. It then analyses the particle positions in successive frames and calculates the paths (trajectories) for all particles. Subsequently, for the evaluation, a vector field is created from the averaged velocities at a particular location.

Of particular interest in this work is in which areas fluid is sheared between the two topographies and in which areas fluid is transported on only one topography. A suitable measure to distinguish this is the ratio V of the particle velocity v_{particle} , related to the disc velocity v_{disc} :

$$V = \frac{v_{\text{particle}}}{v_{\text{disc}}}. \quad (2.8)$$

If the particles are deposited on one of the topographies, their ratio is 0 if they adhere at the upper disc or 1 if they are moving with the lower disc (**Fig. 13**). If they move with the fluid inbetween, their V is

between 0...1. In particular, if the fluid fills the gap locally and is sheared, it can be assumed that the V of many particles is in the range around 0.5.

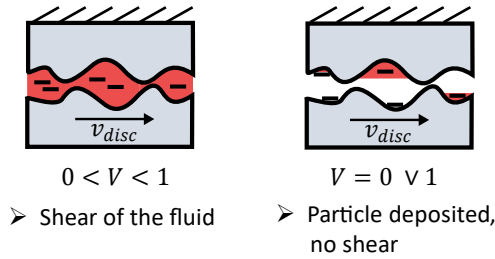


Fig. 13 Identification of locations where fluid is sheared

As there is a scattering of the detected velocities in the real test setup, the areas in the test are categorised as follows:

$0.2 < V < 0.8$: Shear of fluid

Else: Deposited on a topography

At the start of the measurement, the particles are randomly distributed across the topography, and a sufficient sample size with more than 10^6 particle movements are detected during one measurement. Therefore, the share of particles moving relatively between the topographies can be used as an estimate for the area over which the fluid is sheared.

Validation

With the PTV analysis, both, deviations in the test setup and errors in the image analysis, can lead to incorrect results.

Firstly, the detection of the particle positions is validated. A test image with a high particle density is selected for this purpose. The detected particles in this image apparently correspond to the true particle position (Fig. 14a). As an indication that the positions are also correct to an accuracy of less than one pixel, it is common practice to specify the frequency distribution of the decimal part of the particle positions in pixel (Fig. 14b) [32]. As an example, a detected particle with the position $x = 20.45 \text{ pixel}$ has a decimal part of 0.45. If the number of randomly distributed particles is large enough, the decimal

part of the positions should be evenly distributed. An incorrect analysis results in a valley in the centre

- this is not the case here.

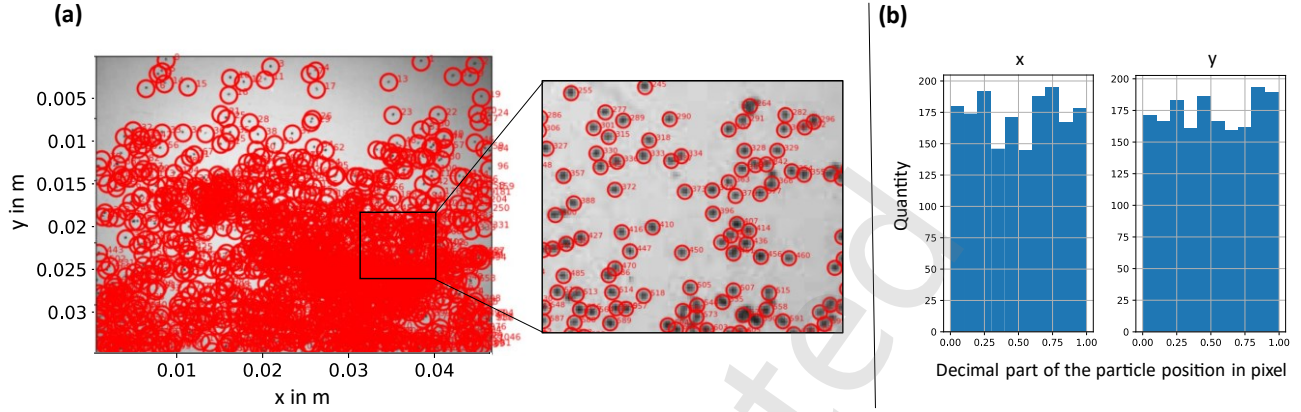


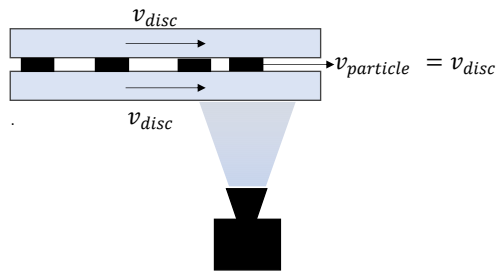
Fig. 14 Validation of the detected particle position: (a) Visual marking of the detected particles in a frame, (b) Frequency distributions over decimal part of the particle position

To validate the entire PTV, the test is carried out on the FLUCS with one modification: The upper glass disc is not fixed against a rotational movement but fixed to the lower glass disc (**Fig. 15**). This causes the two glass discs and the particles between them to rotate at the same determined speed:

$$v_{disc} = v_{particle} \rightarrow V = 1. \quad (2.9)$$

Without measurement errors and deviations in the experimental setup, only particles with $V = 1$ should therefore be detected. As expected, the result shows a peak at $V = 1$ that falls off symmetrically on both sides. The area around $V = 1$ represents the uncertainty of this method for detecting particle movement.

(a)



(b)

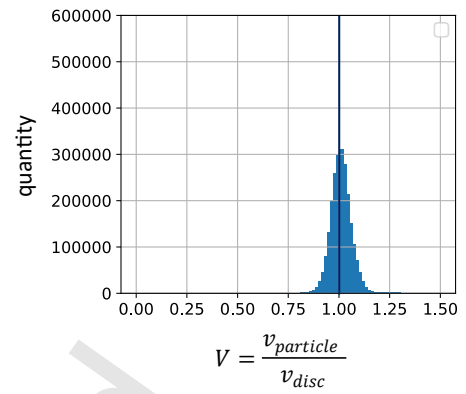


Fig. 15 Test setup for validation of the PTV (a) and result of the validation (b)

3 Results

3.1 Friction

Friction tests were carried out with FLUCS, varying the amount of fluid and speed of the glass disc. The static normal force was set to 27 N in all tests, whereby the experiment was carried out three times under the same conditions. As an example, Fig. 16 shows the time curve of the measured values for a single experiment with a constant amount of fluid. In this test the gap is completely filled with glycerine (4540 μl). The individual speeds are set consecutively and only measurement intervals of full revolutions are evaluated. The measured speed within a speed interval only slightly deviates from the respective nominal value. As a result of differences in surface topography, the normal force varies with the rotational speed of the disc, while the mean normal force fits the nominal 27 N. Due to the viscosity of the fluid, the torque increases with the disc speed and varies only slightly in one speed interval. The test setup and test procedure are therefore suitable for measuring the friction torque at different disc speeds.

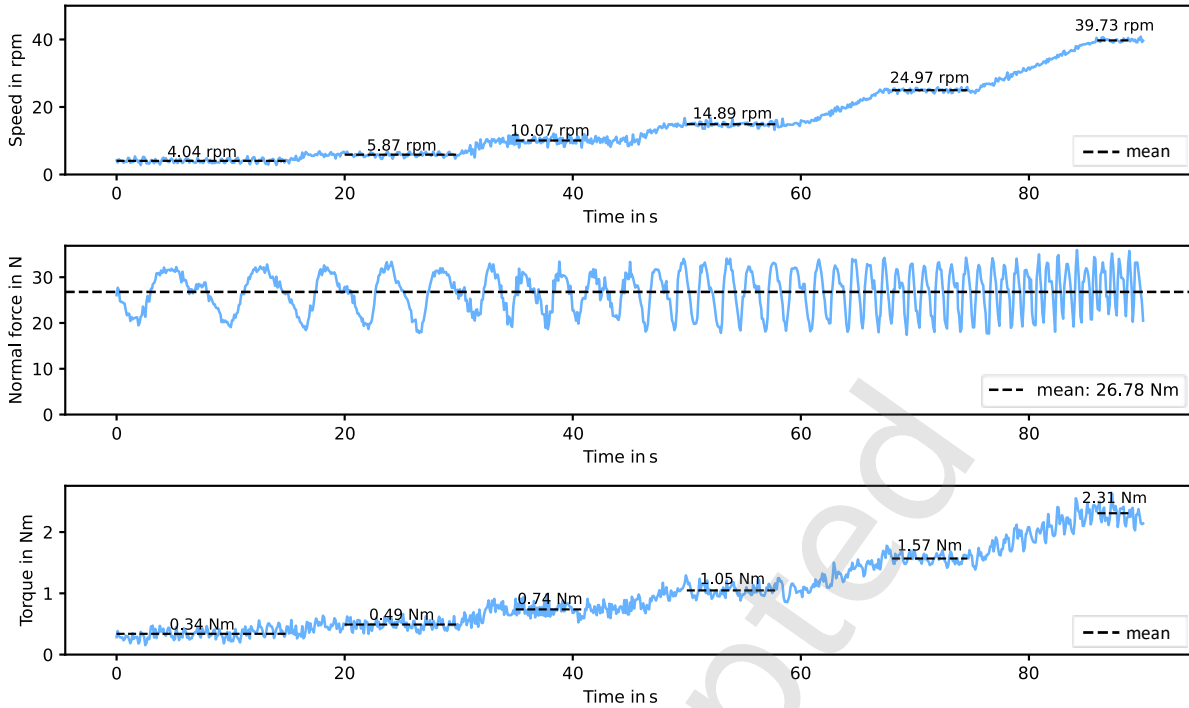


Fig. 16 Measured data with FLUCS over time with a full filling ($4540 \mu\text{l}$)

The calculated coefficient of friction out of the mean torque (see eq. (2.2)) for different fluid quantities is shown in **Fig. 17**. The representation of the coefficient of friction over the fluid quantity results in a Stribeck-like curve (**Fig. 17a**), with a high friction without fluid, the lowest friction in mixed lubrication (here at around $200 \mu\text{l}$) and again a high friction in the regime of purely hydrodynamic friction. A hydrodynamic effect arises due to the macroscopic roughness of the topographies. This qualitative behaviour is independent of the speed and could be observed similar in [3] with a different topography. The coefficient of friction over the speed without fluid is shown in **Fig. 17b**. This shows, that in dry friction is no quantitative correlation between speed and coefficient of friction, as assumed in Coulomb friction. When the gap is completely filled with fluid, the coefficient of friction over the disc speed is shown in **Fig. 17c**. Therefore, in the area of purely hydrodynamic friction the coefficient of friction increases almost linearly with increasing speed. The scaled friction gap thus exhibits behaviour that is also known from microscopic friction gaps.

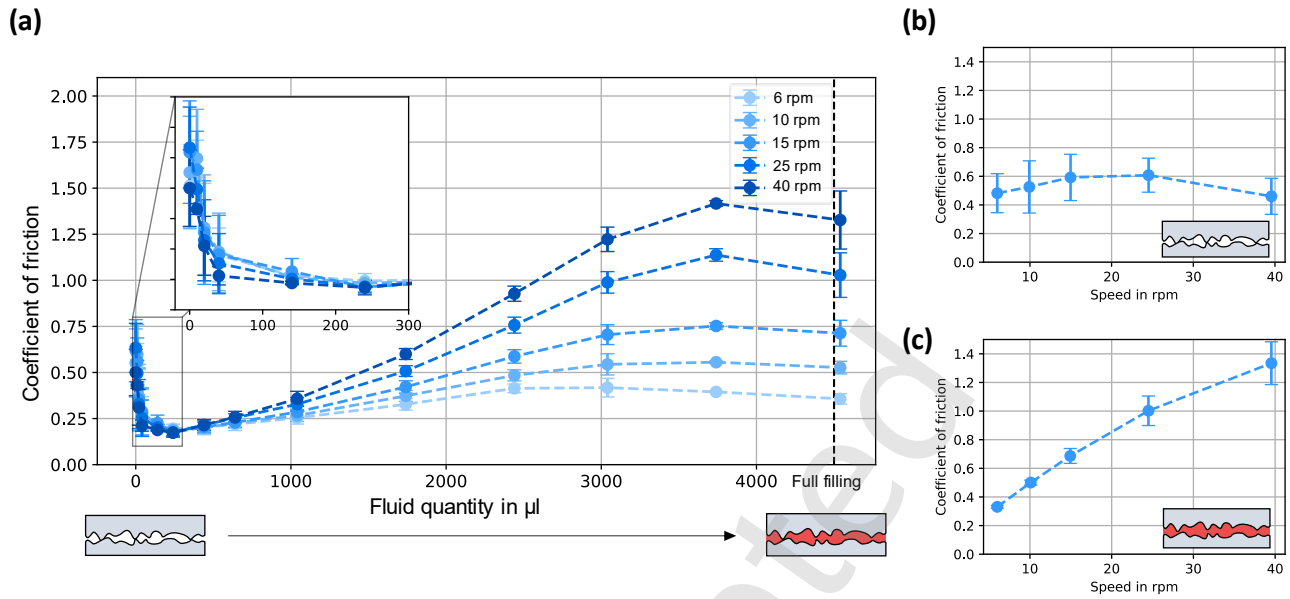


Fig. 17 Average coefficient of friction and standard deviation of 3 test series (a) over the fluid quantity, (b) over the velocity with dry friction (0 μl), (c) over the velocity with a full filling (4540 μl)

3.2 Fluid distribution

A section of the contact surfaces was recorded with the high-speed camera through the transparent topographies. Representative snapshots at 15 rpm and the fluid height distribution calculated from the colour intensity (eq. (2.4)) are shown in **Fig. 18** for three selected fluid quantities. The standard deviation of the fluid distribution is small compared to the mean fluid height and increases with the amount of fluid. While below 140 μl the fluid is distributed in very thin islands, first lines of fluid form from the minimum of the coefficient of friction. Fluid lines therefore in this paper refer to continuous linear accumulations of fluid in the direction of disc movement. The fluid lines in the snapshots are in the same position as those shown by the averaged calculated fluid distribution. A relatively steady-state fluid distribution therefore emerges even with a partly filled friction gap. As the friction gap is not airtight, areas without fluid are filled with ambient pressure air. Similar fluid distributions are found in [3]. The number and height of the concentric fluid lines increases with the

amount of fluid. The distance between fluid lines that are close together is in the range of 1.5 mm. This corresponds to the average distance between two asperites of the topographies (**Fig. 3**). The distance between fluid lines that are further apart is in the range of 3.5 mm - which corresponds to about twice the average distance between two asperites. At small radii, where the relative velocity is smaller, fluid is also deposited outside the fluid lines while at high radii, the fluid is located almost exclusively in the fluid lines. Even after the tests, no visible wear was visible on the glass discs and in the area of the fluid lines. With a fluid amount of 4540 μl the mean fluid height is around 0.2 mm. This corresponds to the maximum height of two asperites (**Fig. 3**) - the contact surfaces are then almost completely separated from each other by the fluid. By calculating the fluid volume out of the mean fluid height one obtains: $\pi r^2 \cdot \bar{h}_{\text{full filling}} = 4522 \mu\text{l}$, which is close to the actual fluid amount and validates the used method.

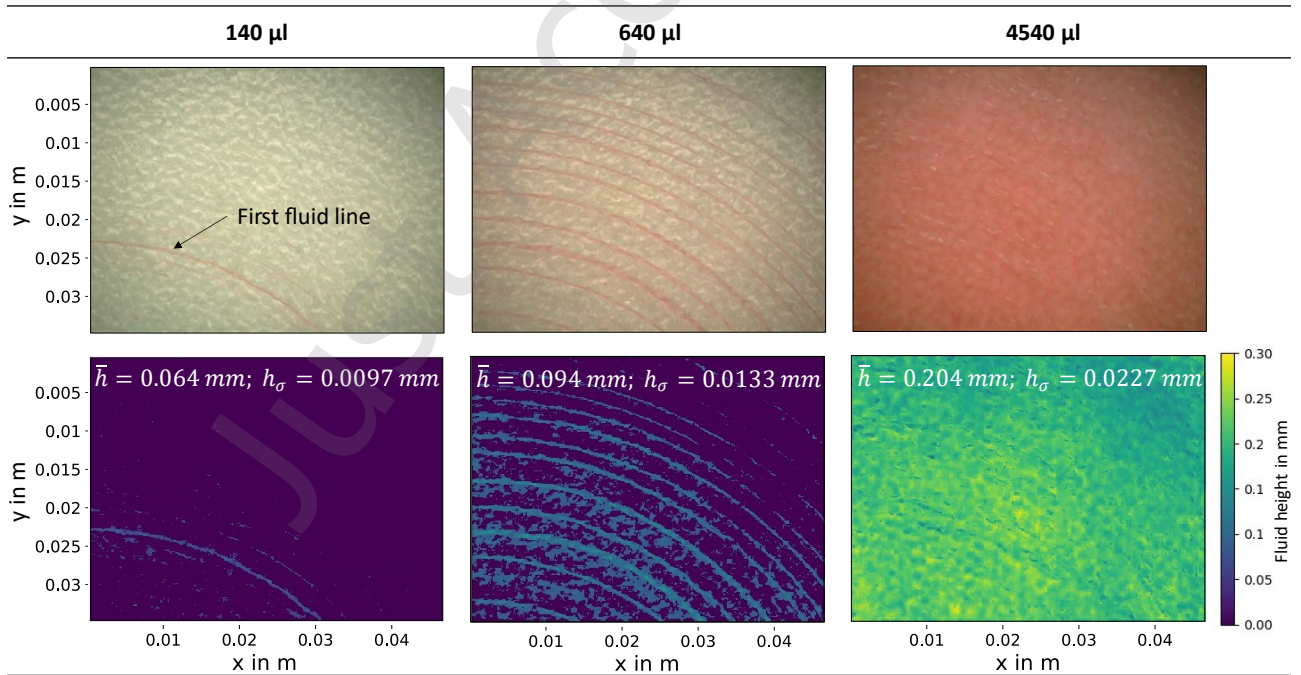


Fig. 18 Snapshots of the fluid distribution at 15 rpm (top) and calculated fluid height distribution (bottom): $\bar{h} \rightarrow$ average height of fluid lines, $h_{\sigma} \rightarrow$ standard deviation of fluid height

To investigate the influence of the fluid quantity on the gap height more precisely, the maximum fluid height was compared with the change in gap height (**Fig. 19**). The maximum fluid height is derived from the calculated fluid distribution while the gap height is measured with a laser at the upper glass disc of the FLUCS. The gap height without fluid serves as a reference from which the change is calculated. The fluid height increases degressively over the fluid volume up to 0.26 mm. The gap height increases much less strongly and the increase varies greatly between the different series of measurements. While the gap height increases sharply with the smallest fluid quantities, it remains almost constant from then on.

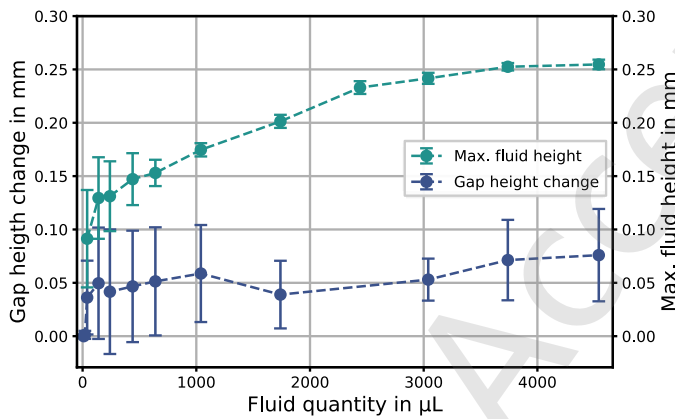


Fig. 19 Mean gap height change and mean maximum fluid height with standard deviation of 3 test series

3.3 Shear area

The calculated fluid distributions already show that fluid lines form in the friction gap. It remains to be analysed whether the fluid is actually sheared in these paths, i.e. whether it is in contact with both topographies and thus contributes to the frictional force. For this purpose and as described in the previous section, 0.1 g of the tracer particles are evenly distributed in the friction gap and the fluid quantity is varied while the amount of particles remains constant.

First, it has to be shown that the particles themselves do not significantly influence the friction contact.

To check this, the previous friction test without particles was repeated with the addition of tracer particles. **Fig. 20** shows the corresponding comparison. The course of the friction coefficient remains qualitatively unchanged. The particles therefore do not significantly change the frictional behaviour and are suitable for tracking fluid movements.

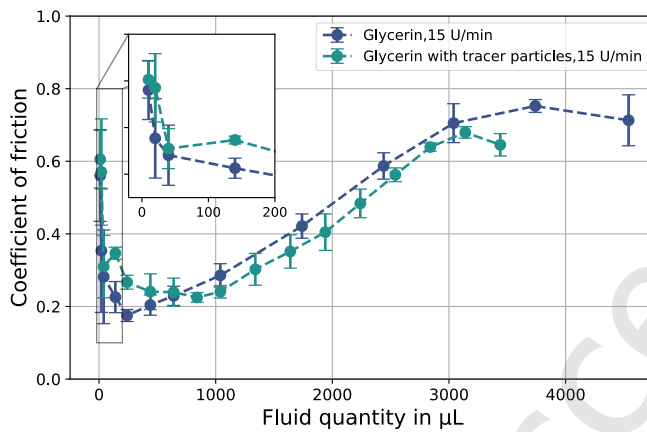


Fig. 20 Average coefficient of friction over the fluid volume with and without tracer particles (standard deviation of 3 test series)

The result of the particle tracking velocimetry is shown in **Fig. 21** for three fluid quantities. In the top pictures, the trajectories of the particles are depicted, where each line represents one track of a particle. Out of the averaged velocities at a particular location the velocity field is calculated, which is shown at the bottom. The fluid quantity of 140 μl represents the transition between boundary friction and mixed lubrication, with the formation of the first fluid lines. The location of the fluid lines matches the blue-green areas in the velocity field, which corresponds to half the plate speed ($V = 0.5$). In the areas without fluid lines, the particles are deposited on the topographies. Due to gravity, the deposited particles are transported on the lower plate that corresponds to $V = 1$ (yellow areas in the top images). The fluid quantity of 640 μl represents the regime of mixed lubrication, where the fluid is present

almost exclusively in the fluid lines. Again the particles in the fluid lines are transported with half the plate speed and are deposited on the topographies outside of the fluid lines ($V = 1$). As the velocity field is time-averaged, it also shows how constant and stable these fluid paths are over time. Since the fluid paths do not migrate outwards over time, this also indicates that centrifugal forces are not a factor at these rotational speeds. The fluid quantity of $3440 \mu\text{l}$ represents the regime, where the fluid fills the gap to a large extent. Therefore nearly all particles are transported with the fluid at half the plate speed. This answers the first question: The fluid in the fluid lines is sheared and is in contact with both topographies.

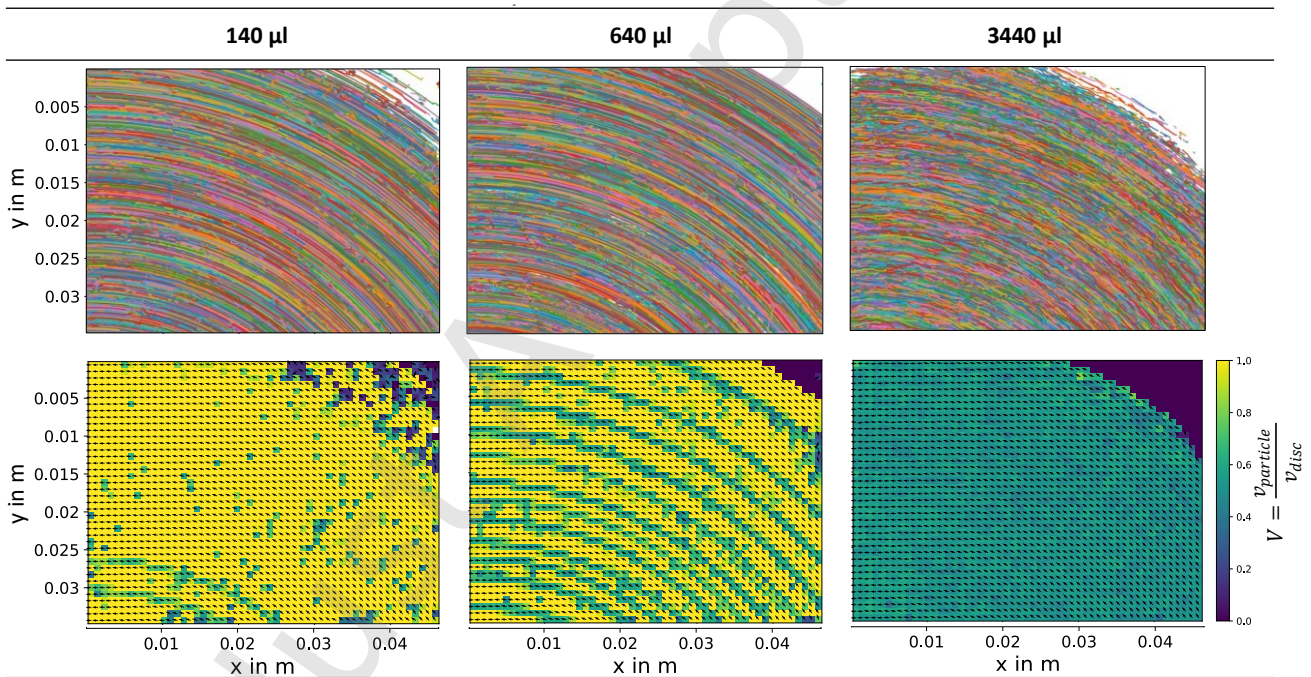


Fig. 21 PTV analysis for three fluid quantities: Trajectories (top), velocity field (bottom)

Decisive for the share of the fluid in the frictional force is the real shear area. In order to obtain an rough estimate for the development of this shear area between the friction regimes, the portion of particles moving relative to the plates can be used. Therefore the frequency distribution of the detected particle movements is shown in **Fig. 22a** for different fluid quantities. The distribution in generally show three symmetrical maxima: The first at $V \approx 0$, where particles are deposited on the upper plate,

the second at $V \approx 0.5$, where the particles flow in the fluid and the third at $V \approx 1$, where the particles are deposited on the lower plate. By sum up all particles, which are deposited and all particles, which flow in the fluid as categorized in section 2.4, ones get **Fig. 22b**. For fluid quantities that are to the left of the minimum on the Stribeck-like curve (**Fig. 20**), almost all particles are deposited and less than 5 % move between the topographies (**Fig. 22b**). With the formation of the first fluid paths at higher fluid amounts, the share of moving particles and thus the real shear area increases approximately linearly (**Fig. 22b**). From 2500 μl around 90 % of the particles are in motion, leading to the maximum shear area. The fluid lines differ slightly in width, which causes them to entrain particles as the fluid flows. Consequently, the maximum number of moving particles is reached before the gap is completely filled with fluid.

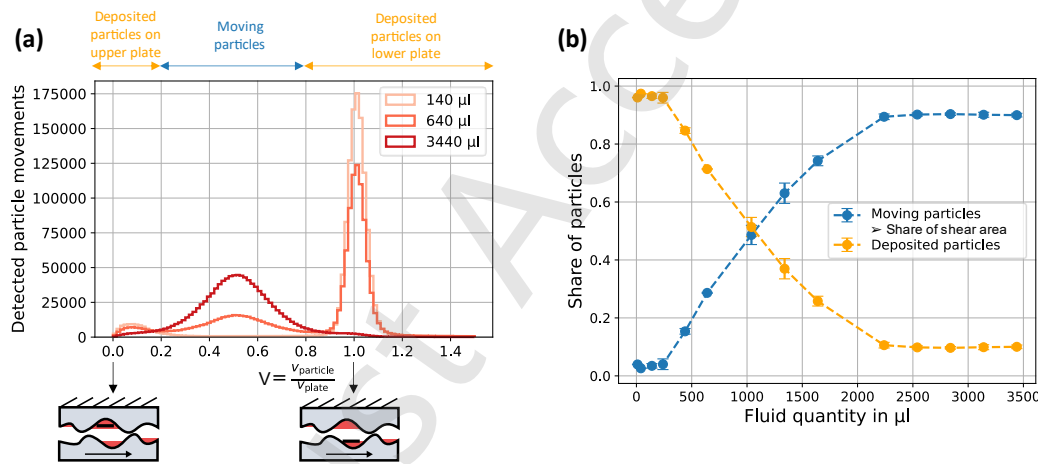


Fig. 22 (a) Frequency distribution of the detected particle movements and (b) derived from this the share of moving and deposited particles

3.4 Initiation of fluid lines in the friction gap

In the results presented, the formation of fluid lines in the friction gap was observed. This section studies the conditions under which these fluid lines form and their influence on the coefficient of

friction. Because the fluid lines formed in the area of mixed lubrication, the following tests were carried out with a fluid amount of $640 \mu\text{l}$.

The viscosity of the fluid was varied by mixing glycerin with water (**Fig. 23**). Three variants are tested: Pure water ($\sim 10^{-3} \text{ Pa} \cdot \text{s}$), 80% glycerine by volume ($\sim 0.1 \text{ Pa} \cdot \text{s}$), and pure glycerine ($\sim 1.5 \text{ Pa} \cdot \text{s}$). The viscosity of the mixture is approximated using the model presented by Cheng [33]. With pure water, the fluid distribution is initially uniform over the radius and small fluid islands form repeatedly for a short time, which immediately dissolve again. At a higher viscosity in the order of $0.1 \text{ Pa} \cdot \text{s}$, individual lines of fluid begin to form. With pure glycerin, the fluid lines become thinner and their number increases. The coefficient of friction decreases with increasing viscosity.

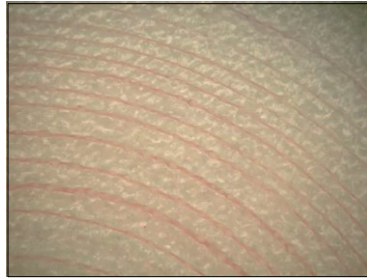
$\sim 10^{-3} \text{ Pa} \cdot \text{s}$ (water)	$\sim 0.1 \text{ Pa} \cdot \text{s}$ (glycerin + water)	$\sim 1.5 \text{ Pa} \cdot \text{s}$ (glycerin)
		
$\mu = 0.870$	$\mu = 0.255$	$\mu = 0.229$

Fig. 23 Snapshots of the fluid distribution and mean coefficient of friction for three viscosities at 15 rpm, $F_n = 27 \text{ N}$ and $640 \mu\text{l}$

In the studies performed, independent of the relative disc speed, no fluid lines form with pure water and fluid lines always form with pure glycerin. With the water-glycerin mixture, however, the fluid distribution depends on the speed: As the speed increases, more fluid lines formed (**Fig. 24**). The share of the surface covered with fluid decreases as a result and the coefficient of friction is nearly constant with increasing speed.

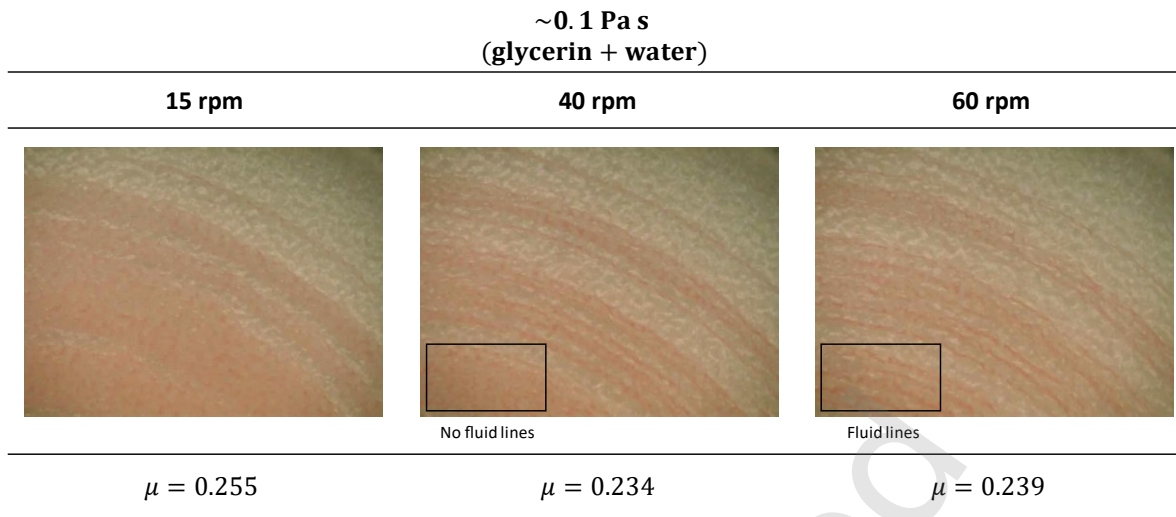


Fig. 24 Snapshots of the fluid distribution and mean coefficient of friction for three disc speeds with 640 μl , $F_n = 27\text{ N}$ and a water-glycerin mixture

Lastly, the influence of the topography was investigated (**Fig. 25**) by combining macroscopically smooth glass discs and the ornamental glass discs (**Fig. 3**). The fluid distribution was also calculated from the snapshots using equation (2.6) and the 90th percentile was evaluated from the fluid heights. With two smooth topographies, the fluid is distributed over a wide area and the coefficient of friction is high. If one topography is rough, the coefficient of friction decreases sharply. The fluid with then forms individual, slightly defined fluid lines and the 90th percentile of the fluid height increases. With two macroscopically rough topographies, the fluid lines are clearly defined and the coefficient of friction decreases due to a smaller shear area of the fluid.

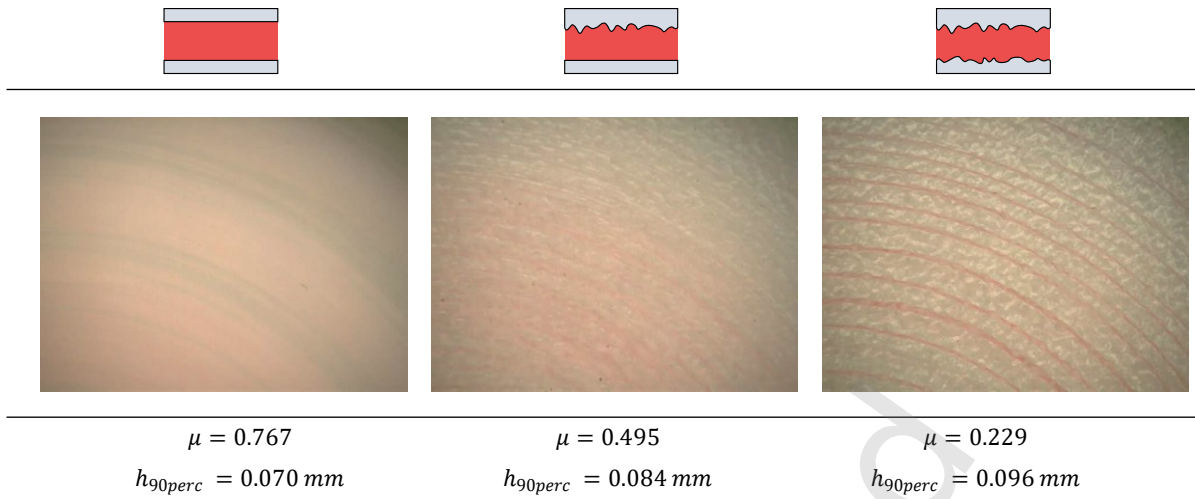


Fig. 25 Snapshots of the fluid distribution, mean coefficient of friction and 90th percentile of fluid height for different topographies at 15 rpm, $F_n = 27 \text{ N}$ and 640 μl with pure glycerin

4 Discussion: How the formation of fluid lines influence friction

In the regime of mixed lubrication, fluid lines have formed in the friction gap, when using glycerin as a lubricant. The relationship between the formation of these fluid lines and the frictional force is discussed in the following (**Fig. 26**). For that, the friction mechanisms are identified based on the results presented above. A minimal mathematical model for calculating the coefficient of friction is derived from these hypotheses and then compared with the measurements. The comparison is used to assess whether the measurement data and interpretations are consistent.

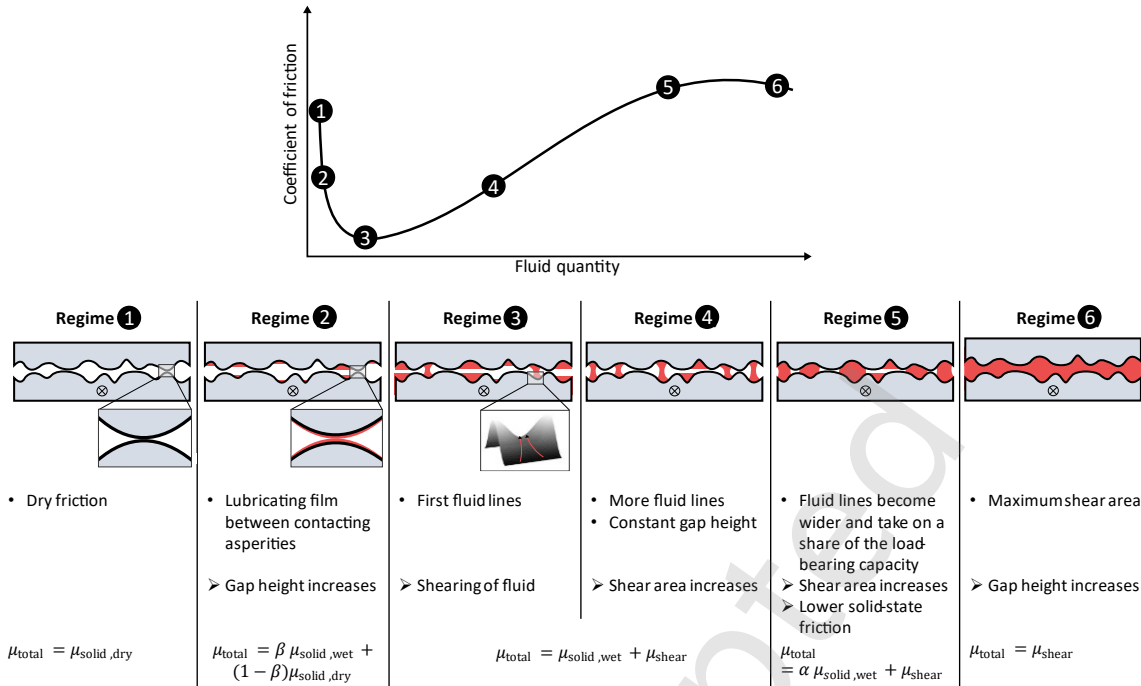


Fig. 26 Influence of the fluid distribution on the coefficient of friction in different friction regimes

Starting without lubricant, the surfaces are initially dry and the coefficient of friction results solely from the solid contact (**Regime ①**):

$$\mu_{\text{total}} = \mu_{\text{solid,dry}} \cdot \quad (4.1)$$

The addition of very small amounts of fluid initially reduces the coefficient of friction considerably (see **Fig. 17**). Only a few per cent of the gap is filled with fluid (**Regime ②**). Up to this point, no fluid lines have formed but the gap height has already increased (**Fig. 19**). This indicates that the fluid forms a lubricating film between the individual asperities that come into contact with each other and thus pushes the topographies apart. Until no fluid lines are formed, the coefficient of friction is independent of the speed (**Fig. 17**). When all contacts are wet the coefficient of friction can be described by:

$$\mu_{\text{total}} = \mu_{\text{solid,wet}} \cdot \quad (4.2)$$

where $\mu_{\text{solid,wet}} < \mu_{\text{solid,dry}}$. While only a few contacts are wet at the beginning, it can be assumed that the share of wet contacts increases as the amount of fluid increases (**Regime ②**):

$$\mu_{\text{total}} = \beta \mu_{\text{solid,wet}} + (1 - \beta) \mu_{\text{solid,dry}}, \quad (4.3)$$

where β is the share of wet contacts. In the range of the minimum of the Stribeck-like curve, the first fluid lines begin to form (**Regime ③**, **Fig. 18**), which are sheared between the topographies (**Fig. 21**). They contribute a further portion to the frictional force due to the shearing of the fluid (**Regime ④**):

$$\mu_{\text{total}} = \mu_{\text{solid,wet}} + \mu_{\text{shear}}. \quad (4.4)$$

By assuming Couette flow and Newtonian fluid, the friction coefficient μ_{shear} due to shearing of the fluid can be derived by:

$$\mu_{\text{shear}} = \xi \frac{1}{F_n} \int_0^{2\pi} \int_0^R \eta \dot{\gamma} r dr d\phi = \xi \frac{1}{F_n} \int_0^{2\pi} \int_0^R \eta \frac{2\pi f r^2}{h_{\text{fluid}}} dr d\phi = \xi \frac{1}{F_n} \frac{4\pi^2 f \eta}{3 h_{\text{fluid}}} R^3, \quad (4.5)$$

where ξ is the share of shear area to disc area, η the fluid viscosity, h_{fluid} the fluid height, f the disc frequency and R the disc radius. According to the results the fluid lines do not significantly influence the gap height at 15 rpm (**Fig. 19**). Consequently the coefficient of friction increases with the formation of fluid lines (**Fig. 17**). The analysis of the flow has shown that the shear area of the fluid paths increases linearly as the amount of fluid increases (**Fig. 22**, **Regime ④**). With the same amount of fluid, it is nevertheless advantageous if fluid paths are formed, as these have a comparatively low shear area and shear rate (**Fig. 24**). Only if the gap is nearly fully filled with fluid does more fluid cause an increase in the gap height (**Regime ⑤**). Then the shear rate decreases with more fluid while the shear area rising more slowly, which results first in a more slowly increasing and then in a decreasing friction force. This effect was also observed in the results (**Fig. 17**) and can be mathematically described with the coefficient $\alpha \in [0,1]$:

$$\mu_{\text{total}} = \alpha \mu_{\text{solid,wet}} + \mu_{\text{shear}}. \quad (4.6)$$

If the contacts are completely separated from each other, the frictional force results solely from the shear of the fluid (**Regime ⑥**):

$$\alpha = 0 \rightarrow \mu_{\text{total}} = \mu_{\text{shear}}. \quad (4.7)$$

Comparison of model and measurement

The equations derived in the discussion can be summarised in a single equation for all friction regimes:

$$\mu_{\text{total}} = \alpha(\beta \mu_{\text{solid,wet}} + (1 - \beta) \mu_{\text{solid,dry}}) + \mu_{\text{shear}}, \quad (4.8)$$

$$\text{where } \mu_{\text{shear}} = \xi \frac{1}{F_n} \frac{4}{3} \frac{\pi^2 f \eta}{h_{\text{fluid}}} R^3.$$

In the following, the consistency of this equation with the measured friction coefficients (**Fig. 17**) will be investigated. In **Regime ①** and **③**, the coefficients of friction are read directly from the measured friction curve and the coefficient β is linearly interpolated between these regimes.

$$\beta = \begin{cases} \frac{V}{V_{\beta}} & \text{if } V < V_{\beta} \\ 1 & \text{if } V > V_{\beta} \end{cases}. \quad (4.9)$$

Between friction **Regime ③** and **⑥**, the friction caused by the shear of the fluid μ_{shear} is added to the unchanging solid friction component. Whether this assumption also corresponds quantitatively with the measured values will be examined here. To calculate μ_{shear} , in addition to the specified process variables (f, F_n, η, R) the fluid height h_{fluid} and the share of the shear surface ξ have to be determined. In equation (4.5) a constant fluid height across the location was assumed. However, the height of the fluid paths varies in the actual experiment, although with a low standard deviation. Therefore, an equivalent inverse fluid height is calculated using the mean fluid height and the standard deviation:

$$\frac{1}{h_{\text{fluid}}} \approx \int_0^{h_{\text{max}}} \frac{1}{h_{\sigma} \sqrt{2\pi}} e^{-\frac{1}{2} \left(\frac{h-h_{\mu}}{h_{\sigma}} \right)^2} \cdot \frac{1}{h} dh. \quad (4.10)$$

The equivalent fluid heights are calculated for three measured fluid distributions (**Fig. 18**) and are interpolated using a second-degree polynomial over the fluid quantity. To calculate μ_{shear} , the share of the shear area ξ has to be defined. According to **Fig. 22**, ξ increases linearly with the fluid volume from the minimum in the friction curve until the proportion is 1 when the gap is fully filled.

Friction

Accordingly, a linear function of ξ between the minimum of the friction curve $V_{\xi start}$ and the maximum $V_{\xi end}$ is assumed:

$$\xi = \begin{cases} 0 & \text{if } V < V_{\xi start} \\ \frac{-1}{1 - \frac{V_{\xi end}}{V_{\xi start}}} V + \frac{1}{1 - \frac{V_{\xi end}}{V_{\xi start}}} & \text{if } V_{\xi start} < V < V_{\xi end} \\ 1 & \text{if } V > V_{\xi end} \end{cases} \quad (4.11)$$

From the maximum of the friction curve, with more fluid a decrease in solid-state contact is assumed due to an increase in the gap height. For simplicity, it is assumed that this decline is linear:

$$\alpha = \begin{cases} 1 & \text{if } V < V_{\alpha start} \\ \frac{-1}{V_{\alpha end} - V_{\alpha start}} V + 1 - \frac{-1}{V_{\alpha end} - V_{\alpha start}} & \text{if } V_{\alpha start} < V < V_{\alpha end} \\ 0 & \text{if } V > V_{\alpha end} \end{cases} \quad (4.12)$$

The selected parameters for the model are summarized in **Table 2**.

Table 2 Parameters for equation (4.8)

Parameters	Value	Derived from
F_n	27 N	Fig. 14
f	[6,10,15,25,40] rpm	Fig. 14
η	1.5 Pa s	Literature
R	0.084 m	Measured
$\mu_{solid, wet}$	0.175	Fig. 15
$\mu_{solid, dry}$	0.6	Fig. 15
$\bar{h}(140 \mu l, 640 \mu l, 4540 \mu l)$	[0.064, 0.094, 0.204] mm	Fig. 16
$h_\sigma(140 \mu l, 640 \mu l, 4540 \mu l)$	[0.0097, 0.0133, 0.0227] mm	Fig. 16
V_β	240 μl	Fig. 15, Fig. 17
$V_{\xi start}$	240 μl	Fig. 20
$V_{\xi end}$	4000 μl	Fig. 15, Fig. 16
$V_{\alpha start}$	4000 μl	Fig. 15, Fig. 17
$V_{\alpha end}$	6000 μl	Assumption

The calculated coefficient of friction with this basic model compared to the measured coefficient of friction is shown in **Fig. 27**. In the regime of dry friction and at the minimum of the Stribeck-like curve, the coefficients of friction for the model were extracted directly from the results, which is why they also correspond directly. However, even in the area of full filling, the model corresponds well with the

measured data. The calculated shear force at the measured fluid heights therefore corresponds to the measured friction coefficients. In the regime of mixed lubrication, the model only matches the measured values at low speeds. For high speeds, the coefficient of friction is overestimated. So the assumption that the fluid paths have no influence on the gap height not applies for high speeds. For high speeds, the fluid probably takes a shear of the normal load, pushes the topographies apart and reduces the coefficient of friction. The assumed correlation between fluid distribution and coefficient of friction can thus be confirmed for low speeds, while at high speeds hydrodynamic effects must be taken into account.

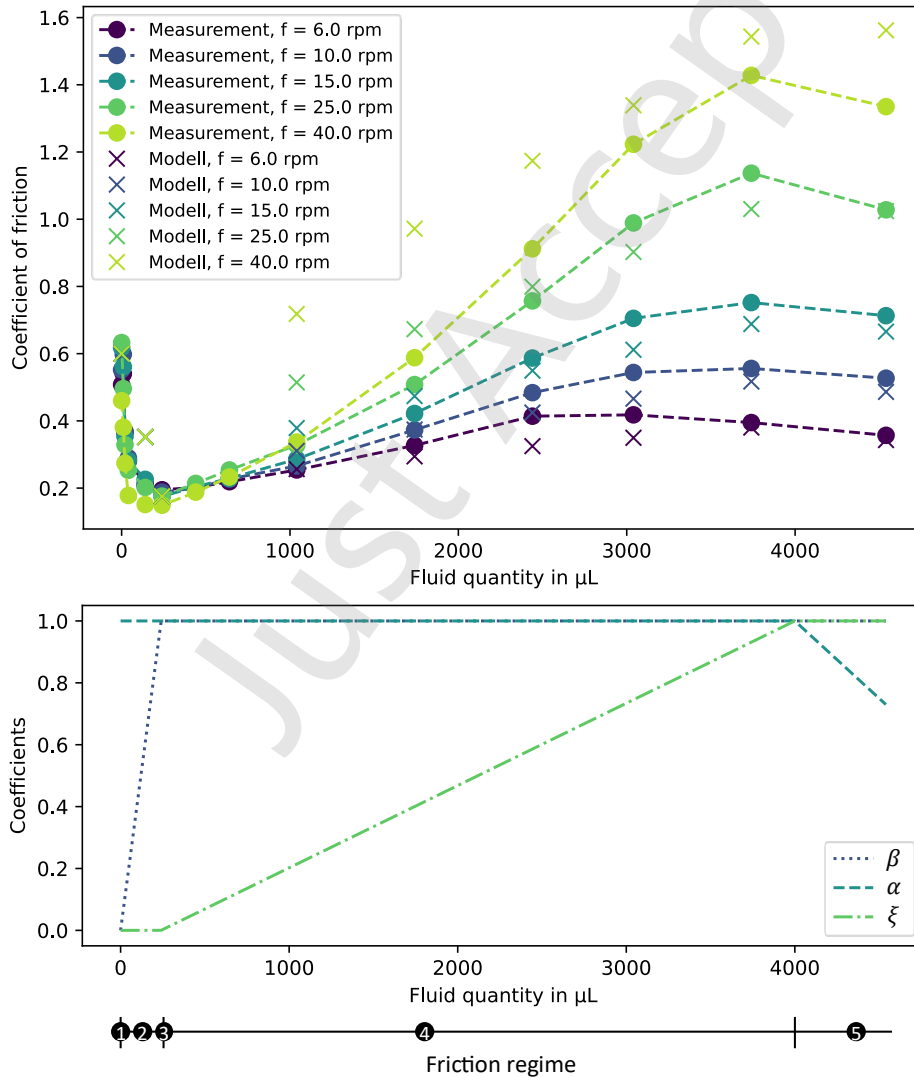


Fig. 27 Measured and model derived coefficient of friction for pure glycerin with two ornamental glass discs and the coefficients α , β and ξ for the model.

5 Conclusions

Until now, little has been known about fluid distribution in starved tribological contacts. This study presents an initial experimental approach to this issue. Using scaled surface roughnesses, it has been possible for the first time to establish a correlation between fluid transport, gap height, process parameters and the coefficient of friction. With coloured lubricant and transparent topographies, the fluid height was calculated based on the colour intensity of the fluid. The shear area of the fluid was quantified using Particle Tracking Velocimetry (PTV). By varying the amount of fluid, a Stribeck-like dependence of the Coefficient of friction was observed and the amount of fluid had a significant influence on the fluid distribution. In the area of mixed friction, the formation of fluid lines in the friction gap has been observed, which is favoured by the following parameters:

- High viscosity
- High velocity
- Surface roughness

The distance between the fluid lines is of the same order of magnitude as the average distance between the asperities of the topographies. The fluid lines are sheared between the topographies, whose shear area increases linearly after the minimum in the Stribeck-like curve. The height of the fluid lines increases degressively with the amount of fluid, but does not contribute significantly to an increase in the gap height for low speeds.

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Author contributions

Timo Fräger: Writing – original draft, Validation, Visualization, Software, Methodology, Investigation, Formal analysis, Data curation, Conceptualization

Kassandra Kampen: Writing – review and editing, Visualization, Supervision, Methodology, Investigation, Conceptualization

Michael Müller: Writing – review and editing, Supervision, Resources, Project administration, Funding acquisition, Conceptualization

Availability of data and materials

Are available from the corresponding author upon reasonable request.

Competing interests

The authors have no competing interests to declare that are relevant to the content of this article.

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Timo Fräger. He studied mechanical engineering at Technische Universität Braunschweig, Germany and received his master’s degree in 2025. Afterwards he joined the university’s institute of acoustics and dynamics as a research assistant. His research focuses on fundamental friction studies and the modelling of fluidic systems.



Kassandra Kampen. She obtained her bachelor's degree in mechanical engineering from the Technische Universität Braunschweig in 2020 and her master's degree in 2023. She then began her

Ph.D. at the Institute of acoustics and dynamics 2023. Her research focuses on experimental studies of tribology and lubricated friction.



Michael Müller. He received his Ph.D. degree in 2007, and his Habilitation degree in 2016, both from Technische Universität Braunschweig, Germany. From 2011 to 2022 he lead the research group on lubricated contacts at the Institute for Dynamics and Vibrations. In 2020, he was appointed Associate Professor. Since 2023, he is Head of the group „Friction and Nonlinear Dynamics“ at the Institute for Acoustics and Dynamics, Technische Universität Braunschweig. His research focuses on fundamental dynamical problems (including cellbiological issues) and tribological basics and applications, with gap flows in general, such as in wet grinding processes, bearings or compression flows of adhesives.