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RESEARCH

Frictional Response of Lubricant in EHL Contact under Transient Bi-directional Shear Loading

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A B S T R A C T

Consideration of performance, efficiency and energy consumption is nowadays an inherent part of the design of every modern machine. These factors are mainly determined by mechanisms taking place within the lubricated contacts. Unfortunately, the physical origins of these mechanisms have been investigated exclusively for steady conditions that do not occur in actual contacts of machines affected by vibrations. This study presents novel experimental data describing friction in the elastohydrodynamically lubricated contact exposed to the main steady sliding motion along with lateral-sliding micro-oscillations. Friction forces were measured simultaneously in two perpendicular directions of point contact. It is shown that the lubricant response in the main direction of motion suffers from shear-thinning and thermal effects whereas its lateral response is isothermal Newtonian. Moreover, the lateral friction affects the friction in the main direction, but not vice versa, when the majority of shear flow is maintained in this main direction. These finding are attributed to the perturbation of structural arrangement of lubricant. The results also suggest that a response of mineral oils to shearing is anisotropic. A limiting shear stress is discussed since the total friction was not able to exceed a certain value.

1. INTRODUCTION

Since many of the elastohydrodynamically lubricated (EHL) contacts work under transient conditions or are subjected to vibrations, there is presently a need to consider more realistic operating conditions in theoretical/numerical and experimental studies dealing with the lubrication mechanisms. These counterformal contacts can be found in the applications of rolling element bearings, gears, cams and followers, constant velocity joints, traction drives, continuously variable transmissions, ball screws, etc. The presence of sliding motions and subsequently the unsteady shear loading of lubricant is typical for such EHL conjunctions. Unfortunately, even today, the overwhelming majority of studies continue to focus on steady state conditions despite the fact that, during the last few years, there has been a sufficient progress in the predictions of both the EHL film thickness and the friction/traction. In certain limits, the steady state solutions are sensible approximations. On the other hand, the investigation of transient conditions can be a way to build on these frequently used and often nearly exhausted approaches and to introduce a new and more realistic perspective on the behaviour of EHL interfaces. This should be done mainly via an experimental approach, as this is the manner how to ultimately prove or refute phenomena and hypotheses, while the theoretical/numerical solutions should serve primarily to explain and generalize these findings.

The efficiency, energy consumption, and performance of machines are directly associated with their contact frictional responses. The changes in EHL friction depending on a sliding speed were shown in 1960s via disc machine measurements [1] where the first traction curves were obtained. The limit of shear stress in lubricant (limiting shear stress τ_{lim}) was speculated by Smith [2] and confirmed by Plint [3] and other followers. Then, several entire decades have been devoted to investigation and clarification of the effects impacting friction, viscosity and other rheological parameters of lubricants, such as the effects of surface roughness, thermo-physical properties, temperature, pressure, density, shear stress, shear strain, strain rate, molecular structure, etc. The knowledge of these effects has been used to establish more general and physics-based models of rheological flow curve (rheological models) for EHL predictions.

It is well known from the previous studies that the four regimes of traction curves can be obtained and distinguished regarding to the contact conditions: linear viscous, non-linear viscous, plateau, and thermal. An excellent quantitative description of these friction regimes for mineral oil can be found by the interested reader in the numerical results [4] based on measured rheological data [5].

The experimental studies referring to the non-steady state conditions began to emerge mostly during the 1990s. However, a majority of these studies deal with the film thickness rather than friction. Some exceptions are listed below. The relation of the transient sliding speed to friction was studied by Hess and Soom [6] on a disc machine, where the speed of the disc was varied at constant acceleration and deceleration for different frequencies of oscillations. These authors pointed out that the size of obtained friction loops increases with the increase in the frequency or viscosity of oil with respect to speed. Moreover, the constant time delay was observed between the minimum speed and the maximum friction. This time delay was independent of speed and was influenced by load and viscosity. The transient frictional response described above was attributed to combined entraining and squeeze film action. This experimental study has been a basis for several theoretical friction models [7-10]. Unfortunately, a large part of these results was obtained for the boundary and mixed regimes of lubrication, rather than for the EHL regime.

Nishikawa and Kaneta [11] showed that the effect of EHL squeeze action can reduce the friction and surface damage during the short-stroke sliding reciprocation as the oil is entrapped by the cyclic impact loads, when the contact suffers from starvation [12] without these impact loadings; especially at short stroke lengths. The effect of impact speed on the film shape and thickness of the entrapped lubricant has recently been described in [13]. For sliding oscillations, the oil did not behave like a viscous fluid as friction was almost the same regardless of the differences in sliding speed [14]. A reduction in friction due to the transversely oriented bump, which probably introduced a small amount of oil during its entry into the point contact, was reported in [15] for the similar conditions.

A significant effect of heating by shearing of the lubricant film was demonstrated via the ball-on-disc simulator during a variation in the slip ratio [16]. The resulting calculations were focused on the design of traction drives. Later, series of experiments on the ball-on-disc apparatus were carried out by Bassani and Ciulli et al. [17]. First, an evaluation methodology was developed and the previously published results by Hess and Soom [6] were confirmed. After that, the effect of both the thermal properties of the contact bodies and the slide-to-roll ratio (sliding speed divided by mean speed of contact surfaces; SRR in short) on the friction was investigated [18]. It was shown that the temperature-viscosity wedge phenomenon (caused by variation of viscosity and temperature through the film thickness [19]) together with the squeeze action can significantly affect the friction according to SRR and the differences in the thermal properties. Also, the squeeze effect was more pronounced at high frequencies and was possibly reduced by the value of SRR. Accordingly, friction decreased faster when the body with a lower thermal conductivity (glass disc) was running faster than the body with a higher thermal conductivity (steel ball) [20,21].

A free sliding oscillating motion was used in [22] to establish a friction law in relation to the operating conditions and rheology of pressurized lubricants. A friction force was not measured directly at the contact interface, but it was estimated from the energy dissipation. The overall dissipation was attributed to viscous damping at the area around the EHL contact, as the pin was passing through the layer of lubricant during reciprocation, and to the friction within the EHL contact. It was admitted that the friction characteristic should be significantly affected by a periodic variation of the lubricant film distribution within the contact and a lubricant piezoviscosity should be taken into account under such conditions. A corresponding numerical model [23] has been developed for the frictional responses obtained on the relaxation tribometer used in [22]. A good agreement between the experimental and numerical data was achieved. Moreover, it was pointed out that thanks to this oscillatory approach it is possible to measure very low values of friction with the "unsurpassed" accuracy of the coefficient of friction of ±0.0002. Friction was recently measured simultaneously with the film thickness of the lubricant at transient kinematics from the boundary up to the full-film lubrication regimes [24]. The total friction was given by the contributions of sub-contact areas of different film thicknesses. Also, the formation of friction loops and the squeeze action were observed for high frequencies and long stroke lengths [25].

From the foregoing outline of unsteady friction studies, it is evident that these studies bring new insights into the behaviour of EHL interfaces. On the other hand, the conditions used in the above-mentioned experiments, albeit transient, are still very distant from the real ones. Often, low pressures are used, and the transient movement is always performed in only one direction of contact. The influence of vibrations that are present in every EHL contact during any machine operation is thus completely omitted. It is still far from clear what the frictional response of high-pressurized lubricant in the actual EHL contacts is, and where the omnidirectional vibrations occur. Being aware of various limitations and deviations from the actual conditions, this work can be a starting point for future investigations of such EHL contacts. For this reason, the present study should not be seen as a completed analysis, but rather as its introductory part. In this study, a sliding EHL contact is subjected to lateral-sliding micro-oscillations in order to examine the frictional response simultaneously in both the main and lateral directions of contact and to evaluate these increments with respect to the total friction of the contact and to the friction under corresponding steady state conditions. The lubricant rheology is considered.

2. MATERIALS AND METHODS

2.1 Experimental apparatus

The experimental apparatus used in this study for friction measurements is a modified ball-on-disc apparatus. The original ball-on-disc apparatus was designed primarily for direct observation and measurement of film thickness in the EHL contacts via the thin film colorimetric interferometry technique (TFCI) [26,27]. The accuracy of film thickness measurement by TFCI is higher than 1 nm. In the related study [28], the apparatus was used to reveal the influence of lateral vibrations on film thickness.

Figure 1a illustrates a schematic view of the modified mechanical part of the apparatus, where a relative motion between the rolling element (ball) and the flat transparent window allows a formation of the EHL film separating the contact surfaces. The point contact was loaded through the window employing a lever mechanism with a dead weight (see [28] for more details). The ball of 25.4 mm diameter was made of bearing steel (100Cr6) and polished to the average surface roughness Ra of less than 7 nm. A transparent 3 mm thick window was coated with 10 nm thick semi-reflective layer of chromium with the aim to apply the TFCI technique for measurement of film thickness and determination of the normal load. The contact surface of the chromium layer is considered to be optically smooth. Normal loads F_N and contact pressures p_h were quantified by the static contacts from the comparison of the deformed shapes of contact bodies obtained by TFCI and calculated from the Hertz theory.

The window was fixed in the compliant mechanism with flexure hinges and a rotational motion of the ball was controlled by a close-loop servo drive. It means that a pure sliding motion was applied in the main direction of contact x (see Fig. 1b). At the same time, a cam mechanism moved harmonically the ball in the lateral direction of contact y, which resulted in transient bi-directional shearing of lubricant. A lateral position of the ball was measured with a non-contact displacement transducer with dynamic resolution of 20 nm and the sampling frequency of 10 kHz. The same sampling frequency was used for measuring of friction forces F_{Tx} and F_{Ty} acting in the main x and the lateral y direction of contact, respectively. All quantitative data were simultaneously recorded at the time intervals of one second. Friction forces were transferred from the contact through the flexible structure and the connecting rods to the precise load sensors. This design enables the friction forces to be measured simultaneously in both directions of the lubricated contact.

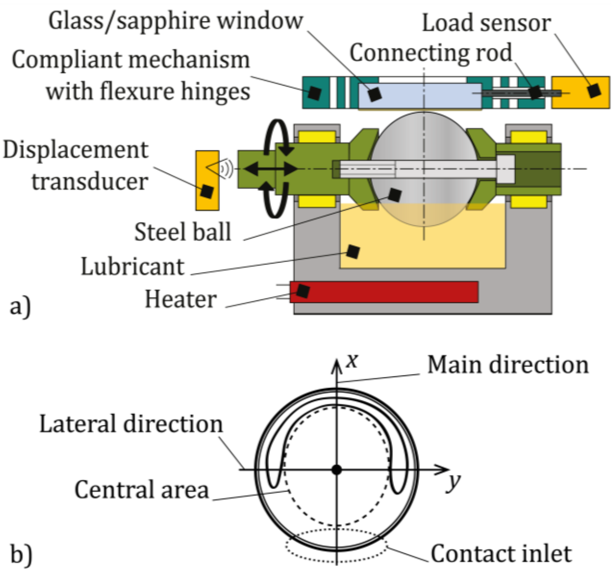

Fig. 1. (a) Schematic representation of experimental apparatus and (b) EHL point contact.

The forces are subsequently used to calculate the coefficient of friction (CoF) according to the following formulas:

$$\mu_x = F_{Tx}/F_N\,,\ \mu_y = F_{Ty}/F_N \tag{1}$$

Similar expressions can be used for the estimation of mean shear stress $\bar{\tau}$ acting over the contact area at mean contact pressure $\bar{p}$:

$$\overline{\tau_x} = \mu_x\bar{p},\ \overline{\tau_y} = \mu_y\bar{p} \tag{2}$$

A measurable range of CoF in both directions is from $\mu = 0.005$ to $\mu = 0.2$ with uncertainty of CoF under 0.0005. The photograph of the above described system is shown in Fig. 2.

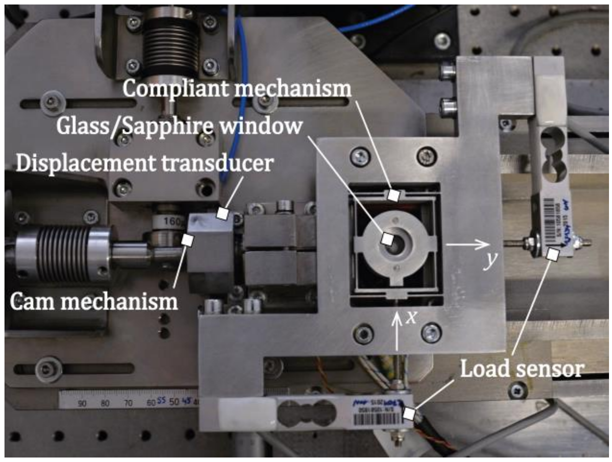

Fig. 2. Mechanical part of experimental apparatus with indicated directions *x* and *y*.

2.2 Experimental conditions

Combinations of two loads F_N and two material configurations were employed to achieve various maximum contact pressures p_h and Hertzian contact diameters d. A contact was created between the 100Cr6 steel ball and the N-BK7 glass or sapphire window. The resulting values of p_h and d are listed in Table 1. Please note that almost the same contact diameter (area) is achieved for the highest and the lowest pressure.

Table 1. Hertz contact conditions.

Material of window	F_N (N)	p_h (GPa)	d (μm)
N-BK7 glass	35	0.53	357
N-BK7 glass	89	0.72	487
Al_2O_3 (sapphire)	89	1.32	358

In the course of measurements, the sliding speed in the main direction of contact u_x was kept fixed at 0.1 m/s. This relatively low speed was applied to reduce the impacts of inlet shear heating, bulk heating and other associated thermal effects on the EHL contact, which are significant at high sliding speeds. A lateral sliding motion was performed for different combinations of frequencies and stroke lengths. The frequency f ranged from 30 to 100 Hz and the stroke length s was in the range from 30 to 220 µm. If these stroke lengths are compared with the contact diameters given in Table 1, then it is clear that the stroke ratio S/D is lower than one. It means that only micro-oscillations were used in this study. The S/D ratio varies from 0.06 to 0.62. A total of 96 measurements were performed under individual contact / kinematic conditions. These conditions allow to apply short-stroke oscillations with S/D < 1, but at the same time they prevent the starvation observed in [11,14,15] under sliding reciprocation.

All experiments were carried out at a lubricant temperature of 40 °C ±0.5 °C. The temperature of lubricant was measured close to the contact inlet with a thermocouple and maintained by a single loop temperature controller. The lubricant was heated through an oil cup via cartridge heaters. Considering low speeds used in the experiments, a bright stock (BS) mineral oil was chosen as the lubricant. This oil is usually employed in low-speed applications such as marine engine oils, cylinder lubes, and gear oils, where a high viscosity is desired. The ambient viscosity and the pressure-viscosity coefficient of BS at 40 °C are 0.329 Pa·s and 19.03 GPa, respectively. The same BS oil has been frequently used by Kaneta and Nishikawa in their studies on dynamic motions [11,14,15].

2.3 Determination of variables

Since the frictional response of lubricant is assessed in relation to the kinematic variables such as position or speed, it should be mentioned how the values of these variables were obtained. Figue 3 shows a relative displacement between the surfaces of the ball and the window in the lateral direction of contact s_y. This displacement is defined as the distance of instantaneous position of the ball s_{by} relative to the instantaneous position of the window s_{wy}; accordingly:

$$s_y = s_{by} - s_{wy} \tag{3}$$

The lateral position of the ball s_{by} is driven by the cam mechanism and is measured with a displacement transducer, as mentioned above. Due to the friction forces acting on the compliant mechanism, to which the window is attached, the window changes its relative position s_{wy}. The instantaneous position s_{wy} is determined considering the force F_{Ty} and the stiffness of the measuring system k, which includes a flexible structure, connecting rods, and force sensors, as follows:

$$s_{wy} = F_{Ty}/k \tag{4}$$

The stiffness k of 0.165 N/µm was estimated during the system design using the finite element analysis (FEA) and subsequently confirmed on the experimental apparatus by direct observations of changes in the position of the window. In the final consequence, the change in the relative position s_y over time is not only determined by the excitation mechanism of vibrations, but it is also dependent on the frictional response of the lubricant. The central (mean) point, around which the resulting relative motion oscillates, was chosen as the reference point for the position s_y.

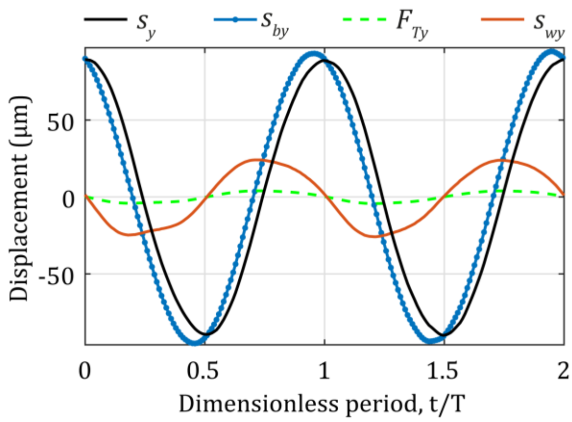

Fig. 3. Relative displacement between contact surfaces.

The lateral sliding speed u_y is then estimated based on the numerical derivation of s_y with respect to the sampling time. The results of the derivation are further filtered via the fast Fourier transform (FFT) and its inverse (IFFT) algorithms to eliminate numerical errors. The data in the frequency domain, the frequency of which is close to the number of data points, are removed from u_y.

3. RESULTS AND DISCUSION

3.1 Uni-directional shear loading

The BS lubricant was first subjected to shearing only in the main direction of contact. The sliding speeds from 0.01 m/s to 0.5 m/s were used together with different contact pressures and material configurations. The central film thickness h_c was measured to ensure that EHL conditions were attained. The film thickness ranged from 25 up to 375 nm for given sliding speeds and contact pressures, as it is illustrated in Fig. 4. The EHL regime was confirmed by means of the minimum value of the lambda ratio λ exceeding 3.5 (full-film lubrication corresponds to $\lambda > 3$) according to the composite surface roughness of the contacting surfaces $R_{1,2}$:

$$\lambda = h_c / \sqrt{R_1^2 + R_2^2} \tag{5}$$

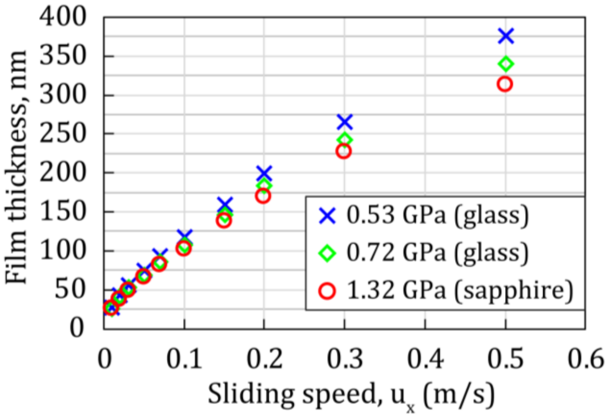

Fig. 4. Central film thickness of BS oil under pure sliding conditions.

Corresponding CoF in the main direction μ_x is illustrated in Fig. 5. CoF in the lateral direction of contact μ_y was at the level of signal noise under these conditions. Therefore, CoF μ_x can be considered as the overall frictional response μ. These values of CoF are used below for the comparison with CoF obtained under bi-directional shearing of lubricant. From Fig. 5 it can be seen that CoF decreases with the increase in the sliding speed for all pressures. However, an opposite trend could be expected with respect to a general trend of Stribeck-Hersey curve, as the full-film lubrication is considered where friction (traction) increases with the increase in the relative speed of contact surfaces due to viscous shearing of lubricant. These results indicate the influences of shear thinning and/or in-contact shear heating of lubricant. The obtained traction curves in Fig. 5 then belong to the thermal regime of EHL traction where the thermal effects on lubricant take place.

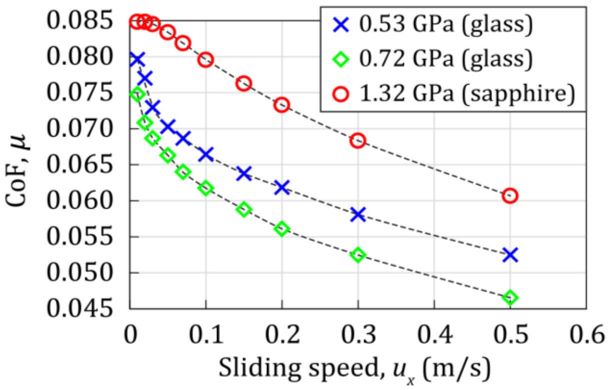

Fig. 5. Frictional response of BS to pure sliding in the main direction of contact.

Just like in the case of speed, friction should be higher for higher contact pressures due to the increase in lubricant viscosity. This did not occur at material configuration of steel ball - glass window probably due to more pronounced thermal effects at higher pressure of 0.72 GPa than at 0.53 GPa. To be true, the effects of heating and shear thinning of BS oil on the reduction of its viscosity and film thickness would have to be greater than the effect of pressure on the increase in BS viscosity.

When the sapphire window is used instead of glass, a significantly higher friction is obtained. This is due to a considerable higher viscosity of lubricant resulting from the contact pressure. It should be noted that the thermal properties of sapphire are closer to the properties of steel than in the case of glass properties. Heat can thus accumulate more at the surface of glass than at the surface of sapphire window, thereby raising the temperature and affecting the viscosity of the lubricant.

3.2 Bi-directional shear loading – friction in the lateral direction of contact

Consequently, the sliding EHL contact at $u_x = 0.1$ m/s was exposed to the lateral-sliding oscillations of different stroke lengths and frequencies introducing the bi-directional shearing of BS lubricant. Fig. 6a shows the mechanical excitation of lubricant as a shear strain to which BS is experienced for four different stroke lengths at frequency of 60 Hz and pressure of 1.32 GPa. Accordingly, Fig. 6b represents CoF in the lateral direction of contact as the stress response to the excitation under such conditions.

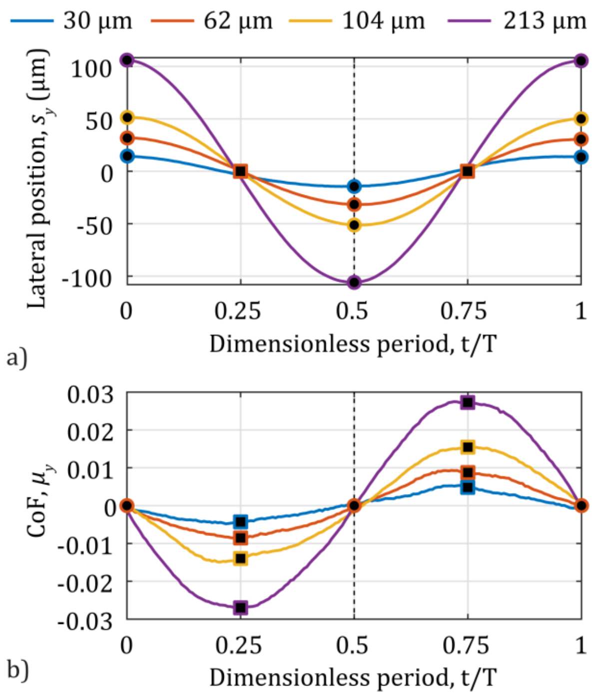

Fig. 6. a) Changes in lateral position, and b) CoF in lateral direction of contact during the cycle of oscillatory motions at f = 60 Hz, and p_h = 1.32 GPa for different stroke lengths.

If the corresponding excitations and responses of BS are compared at the ends of strokes (highlighted by circles in Fig. 6), then it can be seen that the friction is close to zero for different maximum positions at the ends of strokes. Similarly, various values of CoF were measured for the zero positions at the half-stokes (symbols of squares). This means that there is no significant dependence between CoF (stress) and the lateral position (strain). Therefore, a substantial viscoelastic behaviour of BS lubricant cannot be expected under these circumstances. The charts for other pressures, frequencies and stroke lengths are not listed in this article for better clarity, since these results were found to be very similar to those presented above.

The level of viscoelasticity can be quantified from the experimental data by evaluation of the phase shift φ between the excitation and the response of lubricant. The algorithm of Hilbert transform was applied to the data sequence in Matlab in order to obtain the phase information on the measured signals. The resulting phase shift φ for measurements under different combinations of kinematic and contact conditions is shown in Fig. 7. The error bars represent the standard deviation of the phase shift determination between the signals. The maximum lateral speed $u_{y,max}$ achieved over the stroke is used on the horizontal axis of the graph to distinguish the individual kinematic conditions.

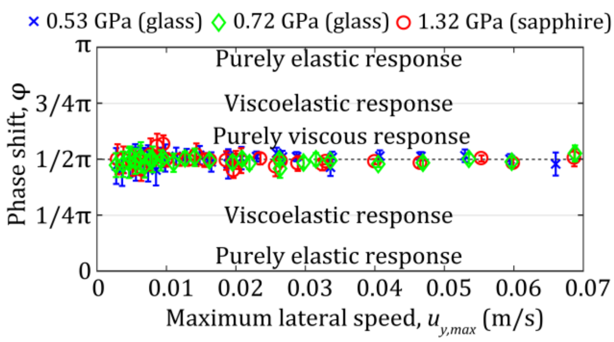

Fig. 7. Phase shift between the excitation and the response of BS lubricant to oscillatory shear stress in the lateral direction of contact.

Based on the phase shifts in Fig. 6 and 7, it is evident that the frictional response of BS is out of phase by $\varphi \approx \pi/2$ from the strain for all tested conditions. These results point out to a nearly purely viscous behaviour without the influence of elasticity in the lateral direction of contact even at the highest pressure, when a purely elastic response would occur at $\varphi = 0$ or $\varphi = \pi$.

When the cyclic variations of CoF and the lateral position are plotted against each other with respect to their phase shift, the Lissajous plots are obtained (see Fig. 8). Since the phase shift between these two variables is close to $\pi/2$, the curves (the Lissajous patterns) are in the shape of concentric ellipses for various stroke lengths and frequencies. Three successive cycles are shown for each condition. The Lissajous patterns are quite stable and they do not differ significantly for the individual oscillatory cycles over time. It is rather obvious that the frictional response in the lateral direction is then time-independent of the rheological point of view and without the consequence of thermal effects developed during the measurements. Moreover, the Lissajous patterns are symmetrical according to both axes, where the horizontal axis corresponds to the ends of strokes and the vertical axis to the maximum speeds over the cycle. Therefore, finally, the applied frequency or stroke length are not important; the only important parameter (in the context of these conditions for the frictional response in this direction of contact) is the resulting lateral speed.

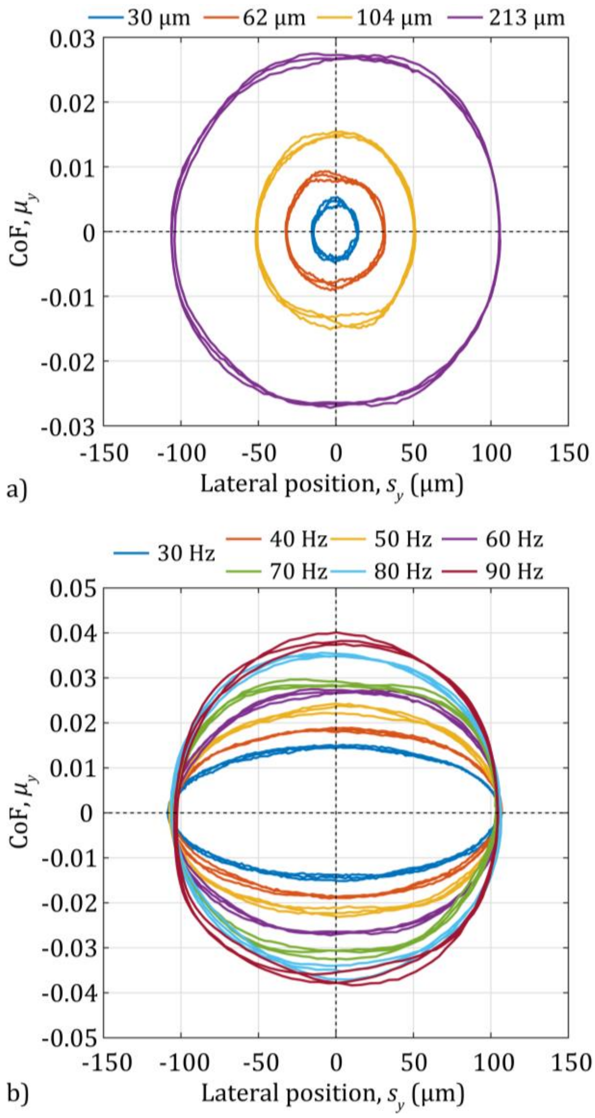

Fig. 8. Lissajous plots showing viscous behaviour of BS at p_h = 1.32 GPa; a) for different stroke lengths and f = 60 Hz; b) for s=213 μm and different frequencies.

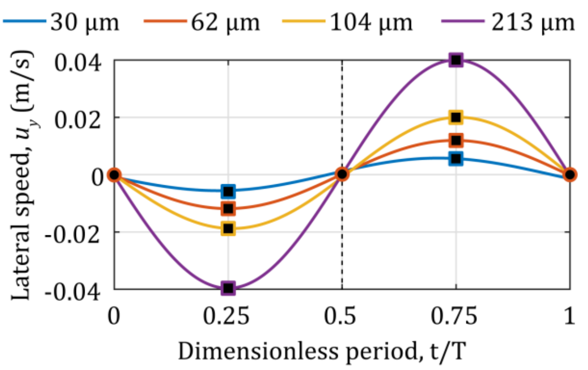

Fig. 9. Courses of lateral sliding speed during the cycle of oscillatory motions at f = 60 Hz and p_h=1.32 GPa for different stroke lengths.

Subsequently, the lateral speed u_y can be considered as the excitation representing the rate of shear strain (shear strain rate) of BS instead of the position s_y. Let us recall that the lateral sliding speed u_y, the course of which during the oscillation cycle is shown in Fig. 9, is given by the numerical derivation of position s_y. Due to derivation, the phase shift of $\pi/2$ between s_y and u_y is obtained. Then, speed u_y and CoF μ_y are in the phase, as is apparent from the comparison of Fig. 6b and Fig. 9.

To evaluate the influence of speed u_y on CoF μ_y, the absolute values of their extrema (maximum and minimum) reached over the individual stroke periods (see the square symbols in Fig. 6b and Fig. 9) were taken into account from the record lasting 1 second. The mean values of these data points of u_y and μ_y were assigned to each other with respect to the experimental conditions. The result is illustrated in Fig. 10, where the linear frictional response of BS can be seen in dependence on the sliding speed.

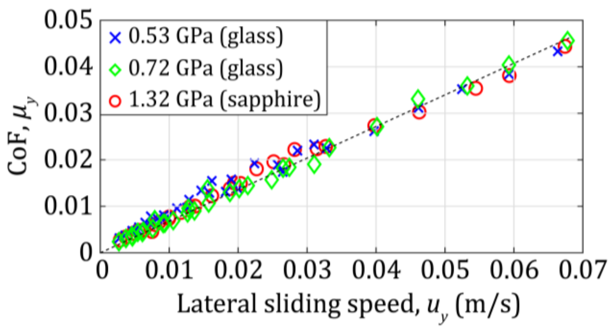

Fig. 10. Frictional response of BS lubricant in the lateral direction of contact under transient bi-directional shearing.

It is noteworthy that this frictional response is the very same for all contact pressures and even for all material configurations used in the experiments. A linear response is clearly not caused by elastic creep of contact bodies explained in [29], which may be mistakenly substituted for a viscous response at low values of SRR. For the current data, the shear stress is linearly proportional to the contact pressure; this proportionality is given by the value of CoF as in formula (2). In the light of these findings, the frictional response in the lateral direction of contact is very likely isothermal and Newtonian viscous, which is highly desirable for investigation of the rheological properties of lubricants under high pressure (mainly pressure-temperature-viscosity relationship usually analysed via high pressure viscometers). Both the isothermal and Newtonian behaviour is very difficult to achieve for actual EHL contacts, especially for those exposed to high pressure and sliding conditions.

3.3 Comparison of uni- and bi- directional shear loading responses and total friction

The two different cases of frictional responses are compared to bring insight into the behaviour of CoF in the lateral direction of contact. The first case is the friction under pure sliding without vibrations, which is shown in Fig. 5, and the second one is the lateral friction under vibrations, as in the case of Fig. 10. As was explained above, if pure sliding is applied only in one direction of contact, the frictional response of lubricant naturally depends on the contact pressure and is reduced with the increase in the sliding speed because of shear thinning and shear heating. On the contrary, none of these effects was observed for μ_y under vibrations, where the friction uniformly increased with the sliding speed.

An explanation for these substantial differences is as follows: when the lubricant is shear loaded only in the main direction of contact and this shearing is of sufficient magnitude and takes a sufficient time, then the structural arrangement of lubricant is changed with respect to the direction of shearing (flow). Since the BS lubricant is a mixture of different hydrocarbon chains with structures of various complexity, the shear stress about 1 to 5 MPa is needed to perturb or disrupt (align, stretch, deform, disintegrate) its chains and structures. The shear stress $\bar{\tau}$ applied to the BS was significantly higher approximately of 20 up to 70 MPa at $u_x = 0.1$ m/s for different pressures (see formula (2)), when the shear strain rate was about $9{\cdot}10^5$ s^{-1} at the measured film thickness of 110 nm ±10 nm (assuming that the velocity of lubricant flow is distributed linearly through the film thickness). Consequently, the non-Newtonian behaviour (shear-thinning) as the response to the applied stress, along with the thermal and pressure effects on the lubricant viscosity, was obtained in the main direction of contact.

However, when the BS lubricant affected by shearing in the main direction is additionally exposed to short-term shearing oscillations in the lateral direction, then it seems that only the isothermal Newtonian response can be observed in the lateral direction, because the other effects are included in the response in the main direction. This linear response is probably given only by the perturbation of current structural arrangement of lubricant in the perpendicular direction to its main flow. The structure may be re-arranged after its unloading during oscillation cycles at the ends of stroke; therefore, this phenomenon can be repeated in subsequent cycles over and over again, as is evident from Fig. 8.

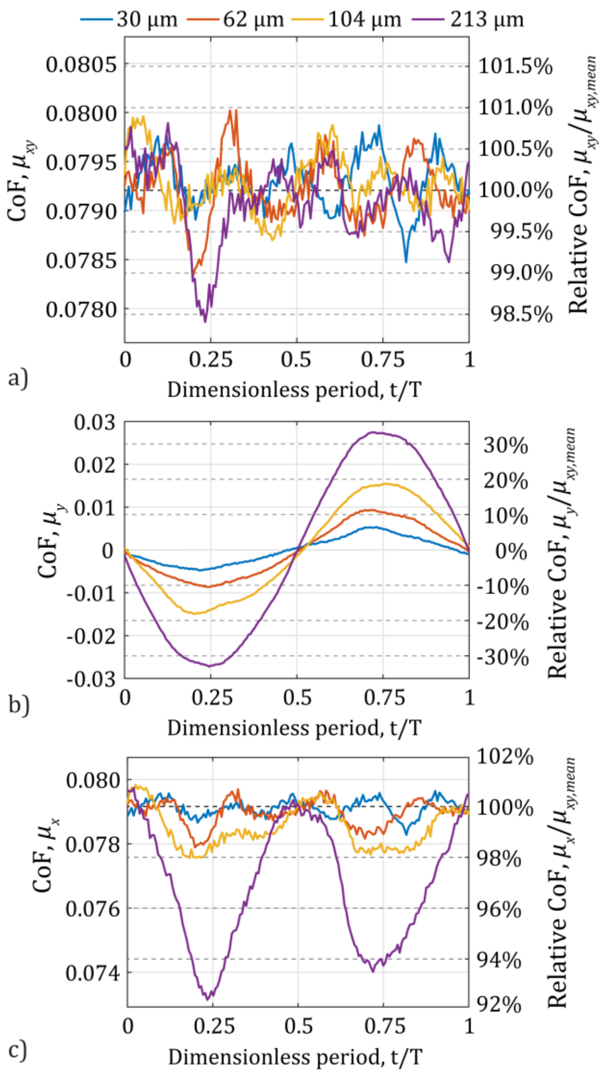

Fig. 11. Frictional response of BS lubricant in the direction of overall sliding speed (a), and in the lateral (b) and the main direction (c) of contact under transient bi-directional shearing at f = 60 Hz and p_h=1.32 GPa for different stroke lengths.

It would be interesting to compare these experimental results with the results of molecular dynamic simulations [30] involving similar conditions. For the sake of clarity, this can be imagined as a tension-loaded rod, which is additionally cyclically loaded by a radial force in the middle of its length. If this assumption is correct, then the friction (stress) in the lateral direction μ_y should affect the friction in the main direction of contact μ_x under vibrations, but not vice versa at the same main sliding speed.

It should be noted that there is an interrelation between the steady and oscillatory shearing flow of lubricant known as the Cox-Merz rule [31]. This rule is mainly applied to the data from vibrational viscometers (often employing cut quartz crystal or shear impedance spectrometer) to characterize the rheological parameters and complex responses of lubricants (low-shear viscosity, shear thinning, time-dependence, linear or non-linear viscoelastic behaviour, etc.). However, it turned out that this rule is applicable only in a very limited range of operating conditions with respect to EHL [32].

Figure 11a shows the total CoF μ_{xy} expressed in the direction of instantaneous sliding velocity over the dimensionless period t/T of motion. It is noteworthy that the mean values of total CoF $\mu_{xy,mean}$ for different stoke lengths are very similar ($\mu_{xy,mean} = 0.0793$) regardless of different lateral sliding speeds. The exhibited disturbance of total friction is given, in particular, by the signal noise which determines the uncertainty of CoF measurements of 0.0005. This disturbance of friction corresponds to approximately ±0.5 % of the mean value of total friction $\mu_{xy,mean}$ under these conditions, as it is illustrated by the second vertical axis of relative CoF. The fluctuations in CoF higher than 0.5 % of $\mu_{xy,mean}$ were observed near the quarter of the oscillation cycle for the two conditions of stroke lengths. Since these fluctuations of 1.5 % of $\mu_{xy,mean}$ did not recur during the reciprocation of motion, their origin is attributed to the dynamic response of the measuring system (not to the response of lubricant) to excited vibrations. Moreover, the occurrence of these fluctuations was rather random, not being identical from cycle to cycle.

The total CoF μ_{xy} is the result of vector sum of the friction components μ_y and μ_x showed in Fig. 11b and Fig. 11c, respectively. It can be seen that the friction in the lateral direction μ_y reaches almost ±30 % of $\mu_{xy,mean}$ for certain conditions, whereas the friction in the main direction μ_x varies in the range of at most 8 % of $\mu_{xy,mean}$ under the same conditions. It is important to note that when both the positive or the negative friction force F_{Ty} is obtained in the lateral direction, then the friction force in the main direction F_{Tx} is reduced. In other words, the friction in the main direction μ_x is influenced by the friction in the lateral direction μ_y, but not vice versa, because there is no other reason for such fluctuations in μ_x than the changes in μ_y, when the main shearing of lubricant is maintained by the constant value of sliding speed u_x. Above that, the friction in lateral direction μ_y was not affected by μ_x, since μ_x varied with contact pressure; however, this change in μ_x had no impact on μ_y, as can be seen in Fig. 10. This observation justifies the above discussed reasons for the differences in the uni- and the bi- directional shear loading responses of the BS lubricant regarding the perturbation of its structural arrangement and the pressure and thermal effects. Moreover, these results suggest that a response of mineral oils to shearing should not be treated as isotropic.

Moreover, achieving of limiting shear stress τ_{lim} is often reported in experiments for similar conditions. There are two indicators of this phenomenon. First, the vector sum of CoFs μ_x and μ_y is not able to significantly exceed a certain value of friction (stress) when this value is represented by $\mu_{xy,mean}$, or by the value of CoF μ under the corresponding steady state conditions (according to contact conditions and speeds). Secondly, the increase in the vector sum of sliding velocities u_x and u_y is not reflected in the change of total CoF μ_{xy} in the direction of this overall velocity. Then also the assumption that the velocity of lubricant flow is distributed linearly through the film thickness (Couette flow) is unlikely to be correct [33,34]. Nevertheless, reaching of τ_{lim} cannot be attributed only to the applied vibrations, but it is a consequence of a combination of contact conditions (pressure, temperature, shear stress) and a complex lubricant rheology.

In the final consequence, the mean values of total CoF $\mu_{xy,mean}$ achieved under vibrations (bi-directional shearing) are very close to the values of CoF μ obtained under steady state conditions at $u_x = 0.1$ m/s (unidirectional shearing) with respect to the contact pressures, as is illustrated in Fig. 12.

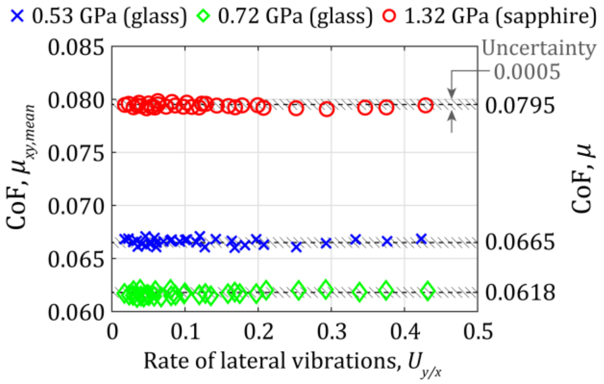

Fig. 12. Comparison of the mean values of total CoF under vibrations with CoF under steady state conditions at u_x = 0.1 m/s.

The rate of lateral vibrations $U_{y/x}$ on the horizontal axis is represented by the ratio of speeds acting in the lateral and main directions. More specifically, it is the ratio of the mean of absolute values of these speeds achieved during one cycle of motion:

$$U_{y/x} = |u_y|_{mean} / |u_x|_{mean} \tag{6}$$

Steady state conditions occur for $U_{y/x} = 0$, whereas a pure sliding reciprocation is assumed at $U_{y/x} = \infty$. Since $U_{y/x}$ was lower than 0.5 for all experimental conditions used in this study, the contact was exposed only to mild micro-oscillations. It can be assumed that the symptoms of the non-Newtonian response of lubricant on its shear loading and the thermal or pressure effects in the specific directions of contact are influenced by the ratio of sliding speeds $U_{y/x}$. If the speed in the lateral direction of contact exceeds the speed in its main direction over most of the cycle (situation when $U_{y/x}$>1), and thus the direction of the majority of shear flow is changed, then the effects of shear-thinning, heating and pressure on the frictional response of lubricant would probably also be observable in the lateral direction of contact. In addition, the frictional response is predetermined by the behaviour of film thickness in the central area of contact under such conditions. The entrapment of thick film of lubricant due to the squeeze action, the film dimple due to the thermal wedge action, the film thickness fluctuations as the result of dynamic motions, the reduction in the film thickness due to starvation, and the film failure may occur under severe vibrations and affect substantially (positively or negatively) friction. None of these phenomena, except for slight fluctuations in the thickness due to the changes in the entrainment speed and because of a surface roughness, was observed in the film thickness under conditions of the current study; therefore, they did not affect the resulting frictional response. In further studies, it would be advisable to experimentally investigate likewise the frictional response in the both directions of contact under kinematic conditions for $U_{y/x} > 1$ and analyse the behaviour of lubricant through numerical simulations.

4. CONCLUSION

In the present experimental study, the frictional response of lubricant in the point EHL contact was investigated under transient conditions of bi-directional shear loading. A modified ball-on-disc apparatus was employed, where the contact was exposed to a steady sliding motion in the main direction of contact simultaneously with the sliding oscillations in the lateral direction of contact. Friction forces were measured using the compliant mechanism with flexure hinges by the precise load sensors in both directions of contact at the same time. Different materials of contact body, contact loads, and frequencies and stroke lengths of lateral motion were applied during the experiments, where the stroke lengths were shorter than the contact diameter.

In regard to the results of the friction measured in the lateral direction of contact, a nearly purely viscous behaviour of BS lubricant was obtained. The influence of lubricant elasticity was negligible even at the highest contact pressure of 1.3 GPa and for different stroke lengths. The course of friction was stable with respect to the oscillatory cycles and regardless of stroke lengths and frequencies of lateral motion. The lateral sliding speed was detected as the only crucial parameter affecting the friction in the lateral direction. It is remarkable that no impacts of shear-thinning, shear-heating, and likewise no influence of contact pressure on the frictional response of lubricant were found in the lateral direction in contrast with the frictional response in the main direction of contact. The absence of these effects is explained via the comparison of frictions under uni- and bi- directional shearing of lubricant. It is deduced that the source of the frictional response in the lateral direction is the perturbation of structural arrangement of lubricant in the perpendicular direction to its main flow. The results suggest that a response of mineral oils to shearing should not be treated as isotropic. This may be useful for investigating the rheological properties of lubricants under severe in-contact conditions of EHL conjunctions.

For conditions where the main shear flow of lubricant was maintained in the main direction of contact, it was shown that the friction in the lateral direction affects the frictional response in the main direction, but not vice versa. The total values of the frictional responses under vibrations were similar to each other in spite of different lateral sliding speeds and considering the contact pressures. These values were also comparable with the values of friction obtained under steady state conditions at the same main sliding speed. The last-mentioned results indicate the occurrence of the limiting shear stress phenomenon, which was discussed for these specific conditions. Finally, the investigation of frictional responses for EHL contact under more severe vibrational conditions is proposed.

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Nomenclature

d	diameter of Hertzian contact area (m)
f	frequency (Hz)
F_{N}	normal load (N)
F_{T}	friction force (N)
h_c	central film thickness (m)
k	stiffness of the measuring system (N/m)
$\bar{p}$	mean contact pressure (Pa)
p_h	maximum Hertzian contact pressure (Pa)
$R_{1,2}$	roughness of contact surface (m)
s	stroke length (m)
s_{by}	lateral position of the ball (m)
s_{wy}	lateral position of the window (m)
s_y	relative displacement between contact surfaces (m)
t	time (s)
T	period of motion (s)
u	sliding speed (m/s)
$U_{y/x}$	rate of lateral vibrations; speeds ratio
λ	lambda ratio
μ	coefficient of friction; CoF
$\bar{\tau}$	mean shear stress (Pa)
τ_{lim}	limiting shear stress (Pa)
φ	phase shift

Subscripts:

max	maximum value over a given period
$mean$	mean value over a given period
x	in the main direction of contact
xy	vector sum of variable components in the directions x and y
y	in the lateral direction of contact