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Discussion on "Scale and contact geometry effects on friction in thermal EHL: twin-disc versus ball-on-disc" by Liu, Zhang, Bader, Venner, Poll, Tribology International 154, 106694, 2021

By

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Abstract

Liu *et al.* recently published a study on friction in heavily loaded lubricated contacts. They compare measurements obtained from tests carried out on two tribometers, at constant entrainment speed, temperature and contact pressure. They observe deviations they attribute to the influence of scale and contact geometry effects.

We show, by means of experiments conducted under the same conditions and with the same lubricant, that three alternative effects, somehow dependent on each other, can explain their results: very different elastohydrodynamic conditions, the pervasive presence of significant thermal effects and the occurrence of a mixed lubrication regime in some tests.

This discussion provides an opportunity to reconsider how to conduct friction tests with in mind to focus on the lubricant response only. Proposals are presented in terms of operating conditions and lubricant selection, with the subsequent objective to better understand the mechanisms behind friction in highly loaded EHD contacts.

Keywords

EHL; Friction; Thermal Effects; Lambda parameter; Lubrication regimes

I. Introduction

Liu *et al.* [1] present a study that aims at highlighting the influence of the contact geometry on the resulting macroscopic friction in EHL. They observe significant deviations in the friction coefficients measured under nominally similar conditions of pressure, temperature and entrainment velocity but from two distinct contact geometries and test rigs. They conclude that a larger radius of curvature involves a higher lubricant heating and may bring about shear localization in the film.

Truly, the idea of decoupling the different physics that come into play in a lubricated contact and studying them separately is very appealing. This approach would make it possible to i) identify the different physics which govern the frictional response of a contact, ii) deduce the main parameters characterizing this behavior, iii) model this behavior to be able to predict, from **ex situ** parameters,

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the friction in a **real** contact. Yet, to specifically study scale and contact geometry effects would require a full understanding of the physical mechanisms that lead to friction in thermal, heavily loaded, EHD contacts. However, as far as we know, such a fundamental knowledge has not yet been established. This is widely due to the difficulty in isolating the physics involved in a frictional EHD contact (thermal heating, high-pressure rheology and thermodynamical state of the lubricant, regime of lubrication...). Thus, the corresponding concepts and simulation tools are not yet available. The ideal approach to model an EHD frictional contact would be to identify a set of physical variables characterizing the response of the lubricant relatively to the operating conditions it is experiencing. This would provide a set of dimensionless parameters that would take into account the relative effects of the various mechanisms at the origin of friction. However, for now, this is only a perspective: no one today can claim to know the exhaustive list of physical variables that govern friction.

The problem in itself is truly complex, certainly involving multi-physical aspects and perhaps multi-scale ones. This most likely can explain why, in the classical EHL literature, the authors have chosen to analyze and model EHD friction from experimental curves. However, the classical approach of friction in EHL cannot be supported by a thorough analysis. As an illustration, Bair has recently proved that information regarding the lubricant's behavior at average pressure cannot be derived from friction at average pressure [2]. He also reminded [2, 3] that effects such as the shear dependence of viscosity could occur in different parts of the contact (inlet, high-pressure zone) and make more complicate or even impossible the classical treatment of EHD friction from traction curves. Furthermore, it should be emphasized that the same complexity can apply to thermal effects, which are discussed in the subject paper [1].

The main purpose of this short communication is to highlight the difficulty of (i) interpreting friction results in relation to the operating conditions and (ii) obtaining results that are representative of the lubricant response only, under fully controlled conditions. In more detail, it aims at:

- providing and sharing original friction results complementary to those of Liu *et al.* [1],
- analyzing their operating conditions through their own data, some analytical models and dimensionless numbers, to show that dominant effects can be quantified and even anticipated,
- suggesting a more appropriate methodology with in mind to obtain more useful friction results in highly-loaded EHD contacts that reflect the lubricant response only.

II. Experimental approach and comparison

All the results presented here relate to squalane, which is the lubricant selected in [1]. It is a synthetic hydrocarbon of formula $C_{30}H_{62}$ purchased from Sigma Aldrich. Friction tests have been performed at 40°C (as in [1]) on a homemade ball-on-disc tribometer (Jerotrib) following the protocol described in [4]. Friction curves have been measured with steel balls ($R_x = 12.7$ mm) and sapphire or steel discs, by varying the normal load and the relative velocities, firstly at a unique entrainment speed,

u_e , of 0.8 m/s. This speed, which may seem low, has been chosen so as to generate a film thickness just sufficient to insure a full separation of the surfaces under the different normal loads applied at this temperature while minimizing the occurrence of undesirable effects in the contact inlet, such as excessive shear-heating or intense non-Newtonian effects. This strategy, described in more detail in [4], has already been applied to perform friction tests that characterize the response of the lubricant only to the imposed conditions of pressure, temperature and sliding speed.

As a reminder, the friction curves presented by Liu *et al.* [1] were measured in steel/steel contacts on two distinct test-rigs, with a Hertzian pressure of 1.95 GPa, a temperature of 40°C, and two entrainment speeds of 5 et 10 m/s.

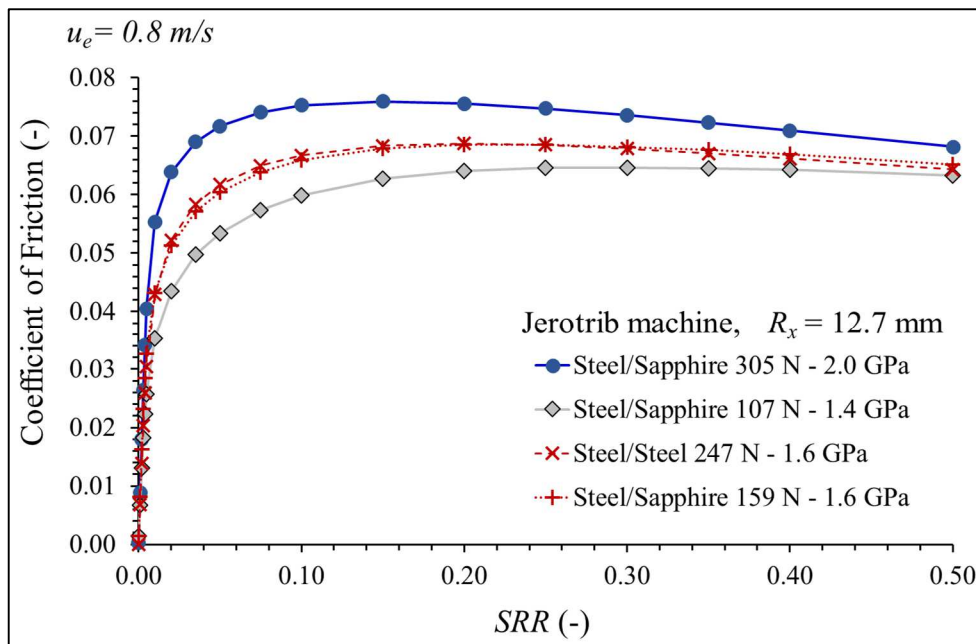


Figure 1. Friction measurements performed at LaMCoS-INSA Lyon, at $T = 40^\circ\text{C}$ and $u_e = 0.8 \text{ m/s}$ with different specimen materials, normal loads and Hertzian pressures.

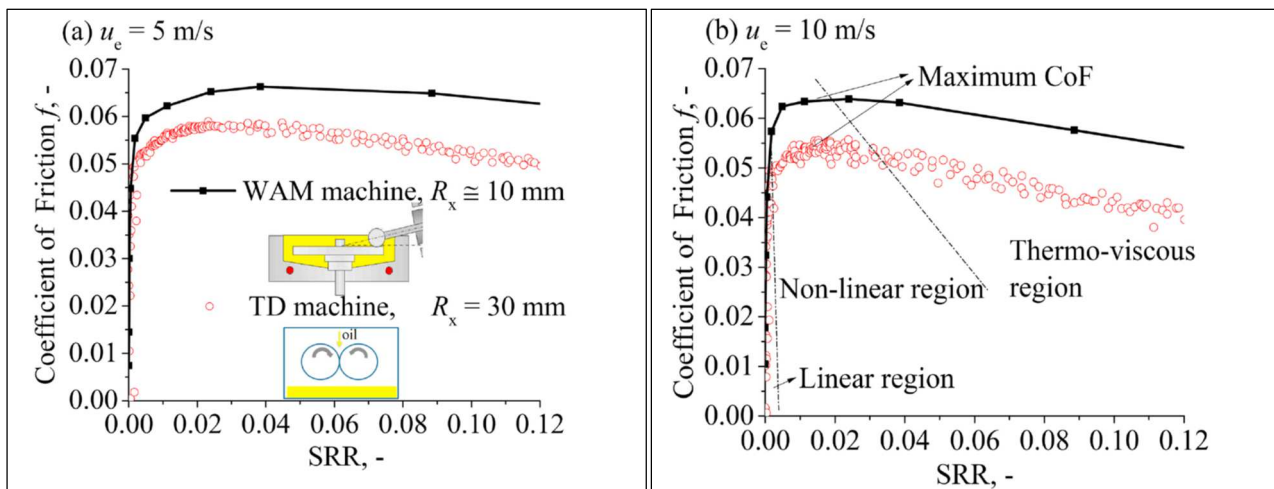


Figure 2. Traction curves measured with two test rigs at constant contact pressure (1.95 GPa) and supplied oil temperature (40°C), at $u_e = 5$ and 10 m/s , from Fig 2 (a) and Fig 2 (b) of reference [1], respectively.

In response to the Figure 2 of the publication by Liu *et al.* [1], three friction curves obtained for steel/sapphire conditions are plotted in Figure 1, where *SRR* means slide-to-roll ratio:

- One obtained at 305 N (circular blue dots) leading to a Hertzian pressure of 2 GPa. The operating conditions are therefore nominally the same as those of Liu *et al.* except the entrainment speed, which is much lower here (0.8 m/s).
 - One obtained at 159 N (red plus sign) leading to a Hertzian pressure of 1.6 GPa. The aim is to show the Hertzian pressure influence on friction, also with the next curve.
 - Finally one obtained at 107 N (grey diamonds) leading to a Hertzian pressure of 1.4 GPa. Surprisingly, the latter results in friction levels similar to those given by the WAM machine, as reported by Liu *et al.* in spite of a much lower Hertzian pressure (1.4 GPa against 1.95 GPa in [1]).
- In addition, the results of a steel/steel friction test carried out at 247 N (red x sign) are also plotted in Figure 1. The objective here is to show that, in our experiments, at constant temperature, entrainment speed and Hertzian pressure, the nature of the materials in contact has little or no influence on friction. Indeed, there is no significant difference between the two curves obtained at the same Hertzian pressure of 1.6 GPa, one with a steel/steel pair loaded at 247 N and the other with a steel/sapphire pair at 159 N. Reaching 2 GPa with steel/steel specimen would have required to overload the test-rig, which explains the use of a sapphire disk instead of a steel one. This point is more detailed in a dedicated figure in appendix, where we compare a set of friction curves obtained with different materials but also at different temperatures and entrainment speeds.

For comparison purpose, the results of Liu *et al.* [1], obtained on both a ball-on-disc (WAM) machine and a twin disc (TD) machine, are displayed in Figure 2. At constant contact pressure and temperature, it is clear that the maximum friction coefficients they report at 5 m/s - of 0.066-0.067 (WAM) and ~0.058 (TD machine) – are significantly exceeded by the one of 0.076 measured with our ball-on-disc tribometer at 0.8 m/s. This represents a difference of +14% and +31% relatively to the WAM and TD machine measurements at 5 m/s, these deviations being even larger when considering results at 10 m/s.

Another approach to show the substantial difference between the two sets of results is to find out to which pressure on our device the maximum friction values reported in Figure 2 correspond. The friction values of 0.066-0.067 for the WAM and 0.058 for the TD machine would correspond to results obtained for Hertzian pressures of respectively 1.4 and 1.2 GPa in our experiments, instead of 1.95 GPa in [1]. This large deviation can obviously not be attributed to a geometrical factor and comes very likely from the influence of undesirable phenomena.

Indeed, in the perspective to improve our understanding on the mechanisms occurring behind friction in EHD lubrication, these disagreements prove that the experiments reported by Liu *et al.* do not account for the rheological behavior of the lubricant alone. They take into account other effects that may not be limited to scale and contact geometry effects, as the title of the subject paper suggests. In the next section, we focus on the operating conditions of these distinct sets of results, in order to discriminate the possible origin of such discrepancies.

III. Real experimental conditions and undesirable effects

Liu *et al.* [1] evokes at least two effects that might primarily influence their experimental results: temperature rise and the occurrence of a lubrication regime that would no longer be full-film. These two phenomena are studied below using simple analytical models and in the light of original experimental friction results.

III.1. EHL operating conditions and thermal heating

On the left side of Table 1, we report several physical and dimensionless parameters that characterize, from the elastohydrodynamics point of view, the operating conditions applied to the three tribometers. At constant Hertzian pressure and lubricant supply temperature, the entrainment velocity is varied from 0.8 m/s, which is the value applied in our ball-on-disc test-rig, to 5 and 10 m/s which were used with the WAM and TD machine in [1].

For a given entrainment speed M , the load parameter, is much larger (by a factor of approx. 4) with the TD machine compared to those obtained with the other two test-rigs. Concurrently, the L value is lower (by about -40%) than that calculated for the two ball-on-disc test-rigs. Although carried out at the same Hertzian pressure, temperature and entrainment speed, the tests conducted on the TD and WAM machines are far from being equivalent from the elastohydrodynamics point of view. In conclusion, friction tests are markedly more severe (M is much larger) with the TD machine. One can even argue that they are very different: according to Table 1, the tests performed at 5 m/s on the TD machine ($M = 2062$, $L = 9$) might be comparable, in terms of elastohydrodynamics, to those carried out at 0.8 m/s on the WAM ($M = 2176$, $L = 7.4$).

u_e (m/s)	w (N)	Dimensionless parameters			Cheng	Chittenden <i>et al.</i>	Reduced h_c
		M	L	k	Φ_T	h_c (nm)	$\Phi_T \cdot h_c$ (nm)
WAM machine - steel/steel, $R_x = 10$ mm							
0.8	300	2176	7.4	1	0.99	88.3	87.1
5.0	300	547	11.8	1	0.91	309	281
10.0	300	326	14.0	1	0.82	494	406
TD machine - steel/steel, $R_x = 30$ mm							
0.8	4300	8210	5.7	1.4	0.99	139	137
5.0	4300	2062	9.0	1.4	0.91	486	442
10.0	4300	1228	10.7	1.4	0.82	778	638
Jerotrib test-rig - steel/sapphire, $R_x = 12.7$ mm							
0.8	305	1604	8.5	1	0.99	94.4	93.3
5.0	305	403	13.4	1	0.92	331	304
10.0	305	240	16.0	1	0.84	529	441

Table 1: Operating conditions and central film thicknesses for three entrainment velocities (in bold) used in the tests carried out on the WAM and the TD machines from [1], and on the Jerotrib test-rig used in this work. In all cases, $P_H \approx 2$ GPa, $T = 40^\circ\text{C}$, $SRR = 0$ and squalane is supposed to be a Newtonian fluid.

On a general level, friction is dissipation and thus thermal dissipation is inherent to any friction process. The question is therefore to estimate in what proportion thermal effects may influence friction, both in the subject paper experiments and during our own tests. We have shown in several studies [5, 6, 7, 8] that it is possible to estimate the appearance of non-negligible thermal effects when Φ_T , the film thickness thermal reduction factor defined by Cheng [9], becomes equal or less than 0.96. This corresponds to the first step in the occurrence of thermal effects in a lubricated contact: the lubricant is self-heating in the contact inlet zone, leading to a reduction in viscosity in this area and thus a reduction in film thickness.

In Table 1 we report Φ_T , the central film thickness predictions according the Chittenden *et al.* equation [10] and the reduced film thicknesses due to the temperature elevation according to Cheng. The Chittenden film thickness expression was chosen among many others based on the work of Wheeler *et al.* [11], who published a quantitative study on the capacity of several EHD equations to accurately predict film thickness, over significant ranges of operating parameters. The viscosity and the pressure viscosity coefficient of squalane at 40°C were calculated using the modified Yasutomi correlation and the definitions proposed by Bair and co-workers [12,13], which leads to values (15.0 mPa.s and 18.2 GPa⁻¹) very close to those reported by Liu *et al.* [1] (15.6 mPa.s and 18.0 GPa⁻¹). The operating conditions are those already given for the friction curves obtained at a Hertzian pressure of ~2 GPa and plotted in Figures 1 and 2.

Note that some assumptions can dramatically affect the film thicknesses reported in Table 1. Squalane is considered as a Newtonian fluid whereas it is shown in the literature that this is not the case. Furthermore, calculations are made assuming $SRR = 0$, thus disregarding the additional thermal effects that take place in the contact area during friction tests where $SRR \neq 0$. In addition, during those tests an accumulation mechanism (thermal over-rolling) occurs, due to the repeated passage of the surfaces in the contact zone. This phenomenon is very rarely balanced by the capacity of the device to absorb and transfer the energy produced within the contact between two passages. Each of the three above effects leads to a reduction of the film thickness and the values reported in Table 1 are therefore overestimations of the actual ones. They can, however, be considered relatively to each other.

Thermal effects due to shear heating in the inlet zone are mainly conditioned by the entrainment speed. Irrespective of the tribometer used for friction testing, Cheng's thermal reduction factor remains constant at constant u_e . Note that in the protocol we have developed in [4] and applied here, Φ_T remains close to 1 which means that the thermal effects due to inlet shear heating are negligible. In contrast, for the tests conducted at 5 and 10 m/s reported in [1], the Φ_T values indicate the presence of inlet heating. A simple estimation (at $SRR = 0$) based on the temperature dependence of the squalane viscosity and on the viscosity exponent in the Chittenden expression results in temperature rises above 4 and 8°C at 5 and 10 m/s, respectively. It is important to remind that additional thermal effects should be taken into account, such as heat dissipation in the contact

zone when $SRR \neq 0$ and thermal over-rolling, both of which contributing to higher temperature elevation.

An alternative method to assess the impact of operating conditions on the presence of thermal effects is to consider the power dissipated in the contact during friction tests, which is the product of the frictional force by the sliding speed. At constant SRR and friction coefficient, this power in the TD machine is more than 14 times larger than that generated by the other two devices, whatever the entrainment speed value. Overall, this is the consequence of i) using large-size specimen with large radii of curvature in the TD machine and ii) the strategy to carry out and compare tests under high entrainment speed and similar contact pressure.

As a conclusion, significant thermal effects, even at zero or very low SRR , affect film thickness and thus friction in the experiments of Liu *et al.* conducted at high entrainment speed and high normal load (especially for the TD machine). Almost all results fall within the thermo-viscous friction regime as defined by Habchi *et al.* [14], which contradicts what the authors have indicated in their Figure 2 (b), where this regime appears only above a certain SRR value.

As an example, we shown in Figure 3 a series of friction tests carried out at eleven entrainment speeds, specifically conducted to highlight the major influence of the entrainment speed. The operating conditions were as follows: steel/sapphire contacts, lubricant supply temperature of 40°C, normal load of 219 N (Hertzian pressure of 1.78 GPa) and entrainment speeds from 0.15 m/s up to 5 m/s. Three entrainment speeds are highlighted with large white symbols and continuous lines in Figure 3: 0.15, 0.8 and 5 m/s. They successively represent i) the lowest speed applied during these tests, ii) the entrainment speed applied to all the regular friction tests conducted at 40°C on Jerotrib (as those reported in Figure 1), and iii) the maximum speed applied to the latter which is also the lowest value chosen in [1]. The results obtained for the remaining intermediate velocities are plotted with black symbols and dotted or dashed lines. Here, the curves are all different from each other and cover a broad friction domain, because of the wide range of variation of u_e , with the lowest entrainment speed generating the highest friction and vice versa. In these experiments, due to the use of a single device, there is obviously no scale nor contact geometry effect that could matter. However, there is a noticeable similarity with the friction curves reported by Liu *et al.* For $u_e = 5$ m/s, while the Hertzian pressure is lower (1.78 against 1.95), the friction curve in Figure 3 reaches a maximum of 0.065, close to 0.066-0.067 found by Liu *et al.* for the WAM machine.

The friction coefficient exceeds 0.08 on Figure 3 at the lowest entrainment speeds. This is therefore much higher than the values reported by Liu *et al.* while the scale and contact geometry are not significantly different between our tribometer and the WAM machine. In their section 3.1 the authors claim "It is hard to measure an "isothermal" traction curve in experiments". In fact, the temperature control and regulation in tribometers is certainly a problem that has not been considered at the level of attention it requires. The design of Jerotrib includes the thermal regulation and insulation from the external environment of the two spindles and its ball bearings, of the two shafts and by conduction of the two specimens, in addition to the lubricant reservoir. The inlet temperature is monitored by

means of a Pt100 sensor, which is positioned approx. 10 mm upstream of the contact. As shown in [4], it is thus possible to carry out friction experiments under nominal isothermal conditions: here, the measured mean temperature of each test and its standard variations are minimum for $u_e = 0.65$ and 0.8 m/s and increase for both lower and higher velocities. For the latter, this trend is consistent with the progressive onset of a thermo-viscous regime, as suggested above.

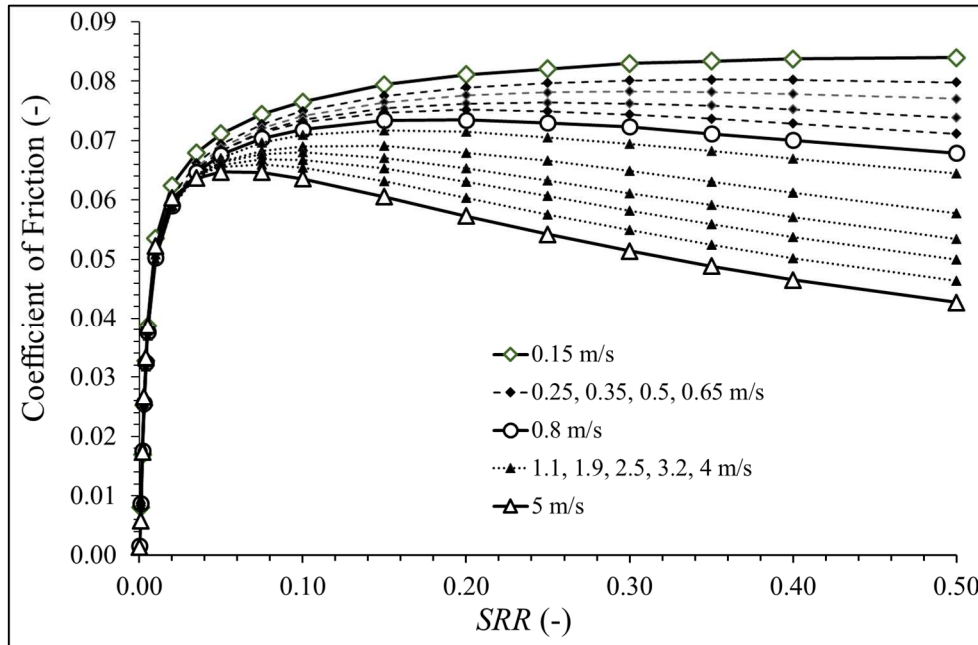


Figure 3. Friction measurements performed at various entrainment speed, from 0.15 m/s up to 5 m/s ($T = 40^\circ\text{C}$, $P_H = 1.78 \text{ GPa}$).

III.2. Lubrication regime

In their publication, Liu *et al.* mention in section 2.1: "All traction curves were measured at full film conditions and the lowest lambda ratio λ_{film} is 2.4. Note that λ_{film} is the ratio of the predicted oil central film thickness h_{cen} with smooth surfaces using Hamrock-Dowson equation [25] to the composite surface roughness R_q (Table 1) of the contacting pairs. The influence of surface roughness on the traction results is expected to be negligible". There are, in these sentences, several points that need to be clarified and corrected.

First and foremost, the criterion for assessing the lubrication regime, full-film or mixed, compares the minimum film thickness, not the central film thickness, to the composite RMS roughness of the surfaces. Most tribology textbooks confirm this definition: see for example the reference books [15]-[21], as well as a recent review of EHL by Greenwood [22], and even a recent publication [23] by the same authors as in the subject paper. Although this point does not fall within the scope of this short communication, it is absolutely coherent to compare h_m to the composite roughness of the surfaces. The probability of direct contact between the rubbing surfaces is maximal where they are closest to each other, and this occurs first where the minimum film thickness is located.

The second point concerns the choice of Hamrock Dowson's expressions (here ref. [24]) to compute the central and minimum film thicknesses in heavily loaded EHD contacts. Numerous works have

shown that these expressions were not reliable under such conditions. Experimentally, as early as 1981, Koye and Winer [25] expressed serious doubts about h_m , later confirmed and extended to h_c by Smeeth and Spikes [26]. Like the latter, but through a numerical approach, Venner [27] has very clearly shown that the slope of the curves h_m vs. u_e was different from that of Hamrock Dowson's model. The latter predicts weak variations (typically in the range 1.7 - 1.8) of the h_c/h_m ratio, even for highly loaded contacts. Several works ([7, 8, 11, 26, 27] among those already cited here) have shown that this ratio in fact varies in greater proportions. It can even reach or exceed 3 in the case of highly loaded contacts (at high M value), leading to much lower minimum film thickness in many cases.

u_e (m/s)	w (N)	Cheng	Moes Chevalier		Reduced Moes Chevalier	Roughness	Λ parameter
		Φ_T	h-ratio	h_m (nm)	h_m (nm)	R_q (nm)	h_m/R_q
WAM machine - steel/steel, $R_x = 10$ mm							
5.0	300	0.91	2.6	147	134	43	3.1
10.0	300	0.82	2.3	271	223	43	5.2
TD machine - steel/steel, $R_x = 30$ mm							
5.0	4300	0.91	4.5	231	210	224	0.9
10.0	4300	0.82	3.4	501	411	224	1.8
Jerotrib test-rig - steel/sapphire, $R_x = 12.7$ mm							
0.8	305	0.99	4.0	28.1	27.7	9.4	3.0
5.0	305	0.92	2.4	174	160	9.4	17.0

Table 2: Operating conditions and minimum film thicknesses in the tests carried out on the WAM and TD machine from [1], and on the Jerotrib test-rig in this work. In all cases, $P_H \approx 2$ GPa, $T = 40^\circ\text{C}$, $SRR = 0$ and squalane is supposed to be a Newtonian fluid.

Table 2 compares the reduced (by inlet thermal effect according to Cheng [9]) minimum film thicknesses with the composite roughness of the specimens used in the three test rigs. The minimum film thicknesses is computed using a combination of the Nijenbanning, Venner and Moes model [28] together with the Chevalier table [29], denoted Moes Chevalier here. It has proven to be the most accurate model to predict minimum film thickness, both in circular [7, 11] and elliptical [11] EHD contacts. As reported in Table 2, the Λ parameter is actually overestimated since it is determined according to two hypotheses: squalane behaves like a Newtonian fluid and $SRR = 0$, these two assumptions each leading to an overestimation of h_m .

For the WAM where important thermal effects are present at 5 and 10 m/s ($\Phi_T < 0.91$), we find $\Lambda = 3.1$ and 5.2, respectively, which suggests that the actual conditions are in - or close to - the full-film / mixed regime transition, because of the assumptions in the calculations. The situation is worse for the tests performed with the TD machine, especially because the combined surface roughness is more important. The Λ values reported in Table 2, which are also highly optimistic owing to the assumptions made, show that these tests were not carried out in the full-film regime.

In addition to thermal effects, this is another main cause of the discrepancy observed by Liu *et al.* [1] between the two series of friction results.

IV. Optimization of friction operating conditions

The achievement of friction measurements that only reflect the response of the lubricant to the operating conditions requires a clever choice of the experimental parameters. The previous section shows that a high entrainment speed generates large heat dissipation, and a low entrainment speed in comparison with the composite surface roughness can lead to a change in the lubrication regime. From the perspective of a better understanding of the mechanisms at the origin of EHD friction, the selection of the lubricant is also very important. These two points are developed and illustrated below.

IV.1. Some critical choices

Figure 4 presents a representation of the sensitivity to the entrainment speed of the friction curves plotted in Figure 3, all the remaining operating conditions being kept constant. The parameters under consideration in Figure 4 are the maximum shear stress (black dots), which is directly proportional to the maximum friction coefficient (times the constant ratio of the normal load to the contact surface), and the Λ values (white squares). Different conditions were repeated, in particular around 0.8 m/s, which is the reference value selected at 40°C in our friction tests with squalane.

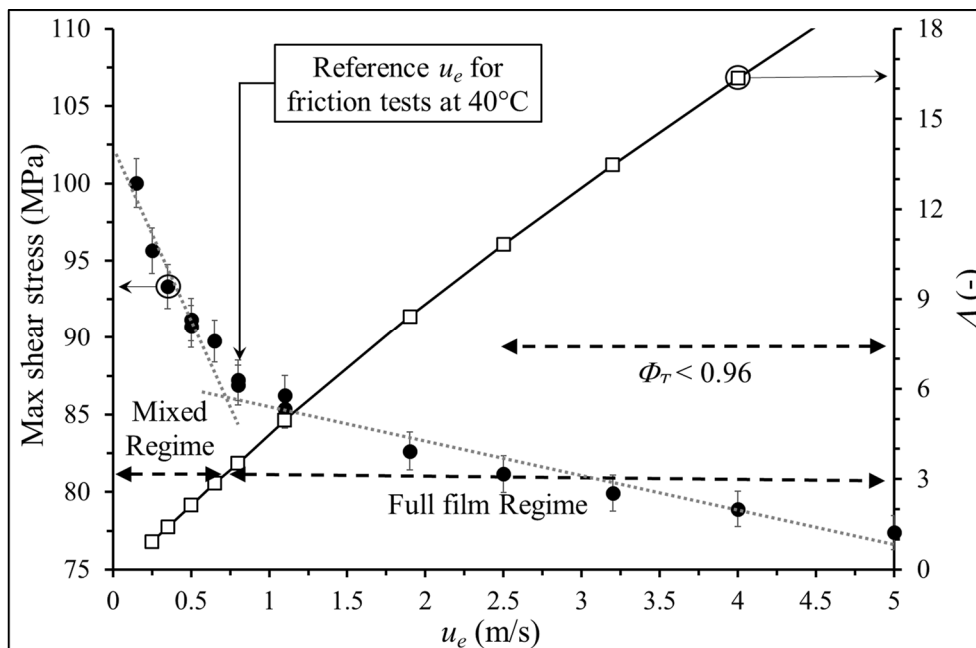


Figure 4. Maximum shear stress (left axis and black dots) and Λ (right axis and white squares) variations as a function of u_e , the entrainment speed, for the friction curves plotted in Figure 3. The dotted lines are just guides for the eye.

Figure 4 highlights two different tendencies. When u_e is equal to or greater than 1.1 m/s, the variation in maximum shear stress is different from the case where u_e is equal to or less than 0.65 m/s where the slope is higher. Moreover, the Λ calculation shows that the latter takes the value of 2.9 at 0.65

m/s and continues to decrease for lower entrainment speeds, whereas it is equal to 3.5 at 0.8 m/s, and continues to increase for higher u_e . The excellent agreement between the λ values and the variations of the maximum shear stress with the entrainment speed confirms the existence of a transition between the full-film regime ($\lambda > 3$) and the occurrence of a mixed regime ($\lambda < 3$), which appears at $u_e \approx 0.65$ m/s. This points out the difficulty to choose a suitable entrainment speed: u_e lower or equal to 0.65 m/s leads to a mixed regime contribution to friction, whereas much higher values (e.g. > 2 m/s) favors the onset of thermal effects. This narrow speed window has been here extended towards rather low u_e values because highly-polished specimen were used for friction tests. Because rough surfaces were used, this was clearly not the experimental strategy adopted in [1], which considerably reduces the possibility to discuss their results through a full-film approach.

The intention to validate a concept or experiments by a numerical model is commendable. However extreme attention must be paid to the assumptions and limitations of the simulation. As long as the heat dissipation within the contact remains low, and therefore has little effect on friction and is rather easily absorbed by the solids in contact, it is still possible to predict friction in a quasi-quantitative way. In addition to the development of a suitable numerical model, the rheological and thermo-physical properties of the lubricant, obtained independently of the tribological tests, are prerequisite [31].

In the case of highly loaded contacts, especially when they are subjected to high entrainment speeds, the conventional numerical models can no longer be adequate because the temperature boundary conditions change in time and in space, and are thus no longer known. The surface and the first layers of the specimens heat up and interact with their supports, which interact with their environment, etc. There is therefore a need for more appropriate numerical tools, capable to solve multiscale transient thermal EHD problems, from the contact to the test-rig scale.

IV.2. Lubricant selection

Squalane is a fairly well characterized lubricant. Since the work of Bair *et al.* [30], its non-Newtonian behavior is well described and adds some complexity in the understanding of friction mechanisms. In order to carry out experiments (why not simulations as well) that allow for progress in this direction, it is really necessary to use adequate lubricants, which present a Newtonian behavior up to very high shear stresses and whose rheological and thermo-physical properties are already known. An additional requirement for these fluids would be that their rheological properties are not very sensitive to temperature, or that friction be studied at a sufficiently high temperature to meet this objective. Thus, it would be possible to limit, to some extent, the consequences of thermal effects.

Ideally, the same approach should be implemented as that initiated by Bair in 2006 [32], in which he recommended the use of a number of fluids as reference lubricants to establish elastohydrodynamic lubrication as a quantitative domain, primarily from the point of view of film thickness generation.

As a potential candidate to initiate this list of reference fluids to study friction in highly loaded EHD contacts, benzyl benzoate meets all of the above criteria. It has been successfully tested under these

conditions and its rheological properties have already been published [4]. In addition, it is available with a high degree of purity, a requirement that is necessary to allow several teams to work under comparable conditions. Naturally, it would be desirable that other fluids could be proposed, if possible of different chemical nature. At the present stage, this remains an open question.

V. Conclusion and recommendations for future work

The many parameters that characterize a lubricated contact fully contribute to the difficulty to exploit friction curves. A relevant analysis requires a perfect knowledge and control of the operating conditions, including the lubricant and the surfaces of the solids. This is particularly essential when the goal is to link the rheological and physical properties of the lubricant to macroscopic friction.

In their paper, Liu *et al.* [1] intent to characterize the scale and contact geometry effects on friction. However, these effects are overwhelmingly dominated by thermal effects and biased by the difference in terms of lubrication regimes between the two devices they have used. Their operating conditions (entrainment speed, surface roughness) and their experimental strategy, based on keeping the contact pressure, temperature and entrainment speed constant, lead to very different conditions, from the elastohydrodynamics point of view.

For any experimental condition considered, it is important to ensure that the lubrication regime corresponds to a full-film regime. Unlike in [1], the film thickness to roughness ratio must be computed with the minimum film thickness. The estimation of film thickness in highly loaded EHD contacts must be performed using the most accurate and robust expressions available, not with the older or more popular ones.

This critical analysis provides the opportunity to suggest the ideal conditions for carrying out friction measurements to study the lubricant's response only, to the operating conditions found in highly loaded EHD contacts. A mixed friction contribution must be absolutely excluded, thermal effects have to be strongly minimized, and the lubricant should be selected according to its less complex behavior at high pressure/high shear stress. All these challenges should lead to friction tests being carried out:

- at the lowest entrainment speed ensuring a full-film isothermal lubrication regime,
- using the smallest possible specimens (at least their radii of curvature),
- with the smoothest possible surfaces,
- and with a lubricant exhibiting a rheological behavior as simple as possible and a low temperature dependence.

In the near future, proposals for adequate reference fluids to study EHD friction are expected i) to allow comparison and discussion between different groups and ii) to advance the understanding of friction in highly-loaded lubricated contacts.

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Appendix

The friction results obtained with different material pairs but at constant entrainment speed (0.3 or 0.8 m/s), contact pressure (1.2, 1.4 or 1.6 GPa) and temperature (20 or 40°C) are plotted in Figure A1. A steel disc (dashed lines) or a sapphire disc (continuous line) were used together with a steel ball. There is no significant difference between two curves obtained at constant operating conditions but from a pair of specimen of different materials. These new results confirm those previously published [4] for other type of lubricants and different operating conditions.

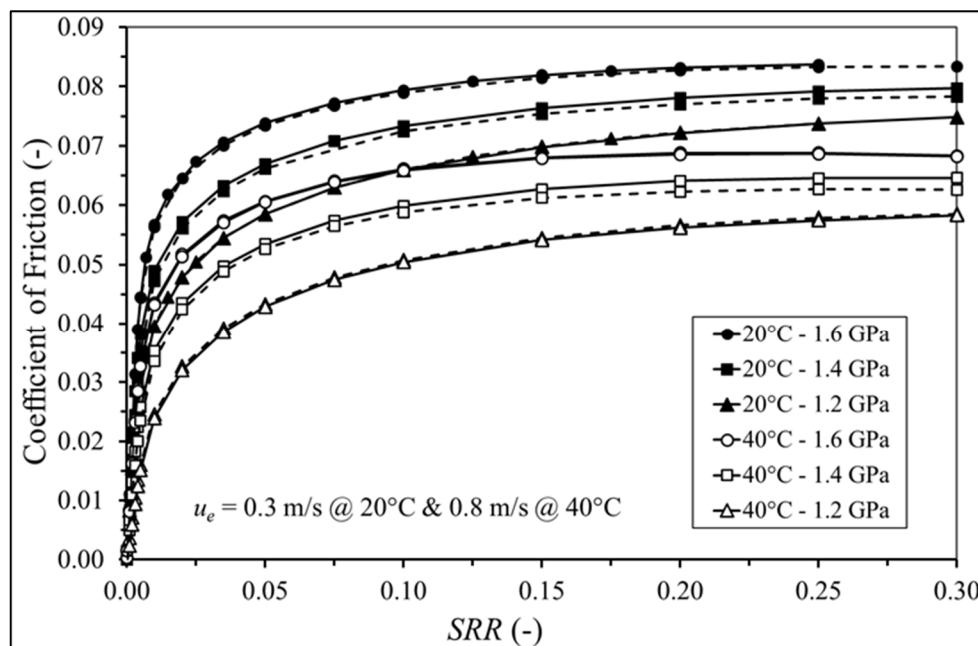


Figure A1. Friction measurements on squalane performed at two temperatures and entrainment speeds, and three different Hertzian pressures (1.2, 1.4 and 1.6 GPa). Tests were performed with a steel ball, and two different discs: a steel disc (dashed lines) or a sapphire disc (continuous lines).

Nomenclature

a	contact radius in the entrainment direction
b	contact radius in the transverse direction
E'	reduced modulus of Elasticity
G	Hamrock Dowson dimensionless material parameter = $\alpha \cdot E'$
h_c	central film thickness
h_m	minimum film thickness
h -ratio	h_c/h_m ratio
k	ellipticity ratio = a/b
L	Moes dimensionless material parameter = $G \cdot (2 \cdot U)^{0.25}$
M	Moes dimensionless load parameter = $W / (2 \cdot U)^{0.75}$
P	pressure
P_H	<i>Hertzian pressure</i>
R_x	reduced radius of curvature in entrainment direction
R_q	composite surfaces roughness
SRR	sSlide-to-roll ratio = $(U_{ball} - U_{disc}) / U_e$
T	temperature
u_e	entrainment speed = $(U_{ball} + U_{disc}) / 2$
U	Hamrock Dowson dimensionless speed parameter = $\mu_0 \cdot u_e / (E' \cdot R_x)$
W	Hamrock Dowson dimensionless load parameter = $w / (E' \cdot R_x^2)$
w	normal load
α	lubricant pressure viscosity coefficient
λ	minimum film thickness to roughness ratio = h_m / R_q
Φ_T	Cheng thermal film thickness reduction factor
μ_0	lubricant viscosity at the contact inlet

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