Friction

Comment

A comment on the review paper "Critical advances in superlubricity: From current challenges to sustainable development beyond laboratory" from a gear perspective

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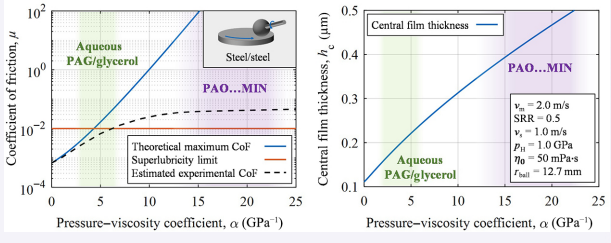
ABSTRACT: Tang et al. [1] present a thorough literature review on the advances in liquid superlubricity and provide a comprehensive overview of the challenges associated with translating model testing findings into practical applications. The review has led to the formulation of the current requirements for achieving liquid superlubricity, namely (I) a low surface roughness, (II) low viscosity, (III) appropriate contact pressure, and (IV) appropriate velocity. While the review is extensive, we think it lacks a critical examination of the potential application of the reviewed tribosystems in the lubrication of machine elements such as roller bearings and gears. The transfer of superlubricity from model scale to machine elements requires the consideration of boundary conditions of industrial applications. These include material properties, lubricant characteristics, and geometric and kinematic constraints. This comment discusses the work of Tang et al. [1] in the context of superlubricity from the perspective of gears, which are typically characterized by rolling–sliding motion and non-conformal contact geometry. We think that it is imperative to refine the conclusions to avoid misinterpretation of the challenges and potential of superlubricity. We demonstrate that superlubricity can be easily achieved in elastohydrodynamically lubricated steel contacts on engineering surfaces, not limited by the four requirements brought up by Tang et al. [1].



KEYWORDS: superlubricity; aqueous lubrication; water-based lubrication; polyalkylene glycols (PAGs); glycerol

1 Introduction

This is a comment on a recently published review on liquid superlubricity by Tang et al. [1]. The review is thorough, and the authors conclude that there are currently four requirements for achieving liquid superlubricity: low surface roughness (I), low viscosity (II), appropriate contact pressure (III), and appropriate velocity (IV). Requirements I and II have been discussed also in previously published review articles (e.g., Refs. [2, 3]), with similar conclusions. In addition, Tang et al. [1] have identified numerous studies using material configurations other than steel to achieve superlubricity, including ceramics (e.g., SiO_2 and Si_3N_4) [4–6] and oxides (Al_2O_3) [7] as well as coatings such as diamond-like carbon (DLC) coatings [8]. However, ceramic or coated machine elements are rarely used. They are found mainly in industrial applications with special environmental conditions, such as corrosive environments. Gears with high power transmission are typically made of heat-treated steel and are liquid-lubricated. The mean coefficient of gear friction for cylindrical steel gears using typical mineral and synthetic oils varies in the range of 0.02 to 0.08, depending on the operating conditions [9, 10]. Gears with negligible and low power transmission are often made from technical polymers [11]. In the case of liquid-lubricated polymer gears, the contact-mechanical friction can be lower than that of steel gears. At model testing, superlubricity has been found for oil-lubricated polymers, even with mineral oil [12]. We believe that the material configurations identified by Tang et al. [1] do not currently represent typical machine elements and that applying them would require enormous and disruptive innovation and development. Also, we believe that the four reported requirements for achieving liquid superlubricity are too narrow in the application of superlubricity to typical machine elements, such as gears. Therefore, this comment will discuss the work of Tang et al. [1] in the context of liquid superlubricity from a gear perspective. We will show some missing references considering aqueous lubricants, which have already demonstrated persistent liquid superlubricity considering engineering surfaces. Additionally, we will show the underlying mechanisms leading to liquid superlubricity and the boundary conditions for a transition from superlubricity to superefficiency. Thus, this comment aims to

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stimulate discussion on bridging the gap between model testing and transfer to industrial applications.

2 On the completeness of the literature review

In their review article, Tang et al. [1] provide a comprehensive list of over 160 references focusing on liquid superlubricity. Liquid superlubricity has become a highly active area of research, with many articles currently being published. For example, Scopus lists 314 articles for the keyword "Superlubricity" between 2021 and 2024, while it shows an increase of only 654 over the past twenty years. Some articles not cited in Ref. [1] have already demonstrated liquid superlubricity without meeting the requirements postulated by Tang et al. [1] (cf. Section 1). We think that this can lead to misinterpretations in terms of the challenges to transferring liquid superlubricity from the laboratory to machine elements. It appears that the literature review was conducted from a unilateral perspective. Two examples of superlubricity without meeting the requirements postulated by Tang et al. [1] are described within the following paragraph. In 2014, we published a study on aqueous glycerol and showed that superlubricity can be achieved at 1.35 GPa and with engineering steel surfaces [13], which shows that the requirements I to III postulated by Tang et al. [1] are not mandatory to achieve superlubricity. As can be seen in Fig. 1, coefficients of friction of around 0.01 can be achieved with aqueous glycerol with 10–20 wt% water, resulting in dynamic viscosities between 40–150 mPa·s. Thereby, fluid film lubrication was present when the water content in glycerol was 20 wt% or less.

In 2019, we considered different fully-formulated aqueous polyalkylene glycols (PAGs) of ISO VG 22 and ISO VG 46 at a FZG back-to-back gear efficiency test rig [9] under dip lubrication. Figure 2(a) shows exemplarily the measured mean coefficient of gear friction over the pitch line velocity for a contact pressure of approximately 1.7 GPa. Figure 2(b) shows a picture of the test gear's tooth flanks with an average arithmetic surface roughness of $R_a \approx 0.14$ µm. Figure 2(c) shows the kinematic viscosities ν of the considered aqueous PAGs (PAGWs): PAGW-09, PAGW-05A, and PAGW-05B. For all aqueous PAGs considered, superlubricity was found for a wide range of pitch line velocities corresponding to different severities of mixed lubrication. The results demonstrate that the postulated requirements by Tang et al. [1] are not necessary to achieve liquid superlubricity with engineering surfaces. Furthermore, the results underline that superlubricity is already possible in gears and thus beyond the model scale.

More discussion considering the underlying mechanisms is given in Sections 4.1 and 4.2.

3 Limitations of the literature review

The literature review of Tang et al. [1] focuses on current challenges in achieving liquid superlubricity on model scale and how to transfer the findings to real-world applications. However, most of the references in Ref. [1] concern conditions that are hardly transferable from the model to gear scale. Examples are the use of acids (Section 3.1) and the consideration of geometric and kinematic boundary conditions (Section 3.2), which are not representative of gears.

3.1 Acids

Typically, liquid lubricants used for gear lubrication consist of a base oil and additives, which determine the mean coefficient of gear friction [9, 10]. The concentration and type of additives will depend on the specific requirements of a gearbox. Commonly used additives include extreme pressure, antiwear, viscosity index improvers, friction modifiers, and corrosion inhibitors [14–16]. In the context of aqueous lubricants, Tang et al. [1] acknowledge the effectiveness of acids such as phosphoric acid [17–23], sulfuric acid [24], or hydrochloric acid [25], which have been demonstrated to facilitate liquid superlubricity on ceramic, oxide or glass surfaces. Of note is the study by Sun et al. [26], which examined 3-hydroxypropionic acid at a pH value of 1.5 with a contact partner being plain steel. These acids were found to induce accelerated corrosion in steel components and cause material incompatibility with elastomers used for seals [27–29]. Therefore, such acids are not suitable for use in steel gears and gearboxes based on the current state of technology.

3.2 Geometry and kinematics

The most important aspect in interpreting superlubricity

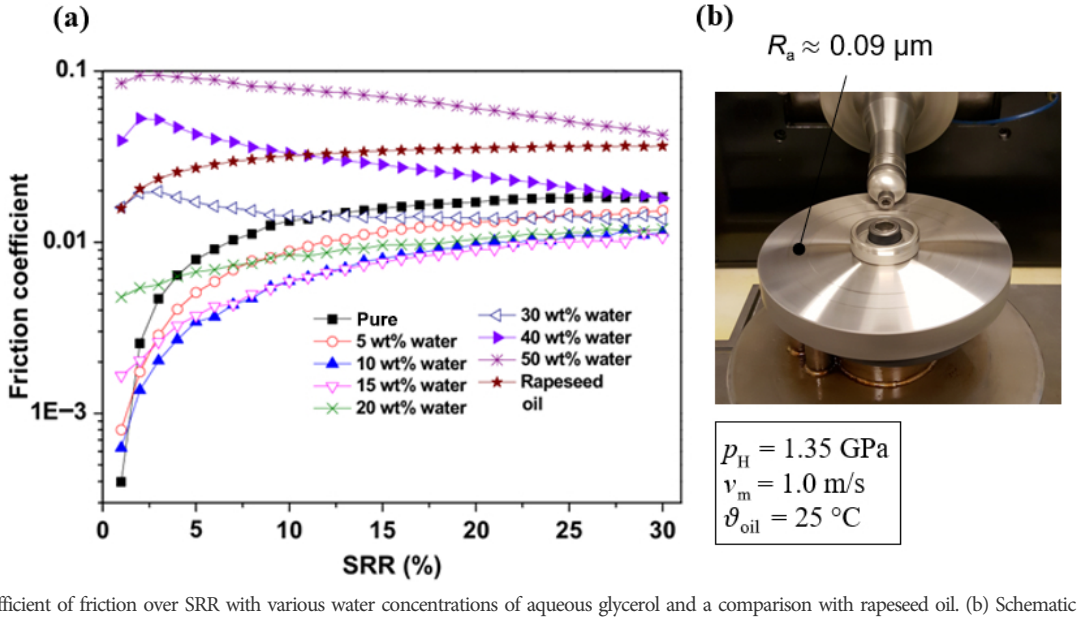
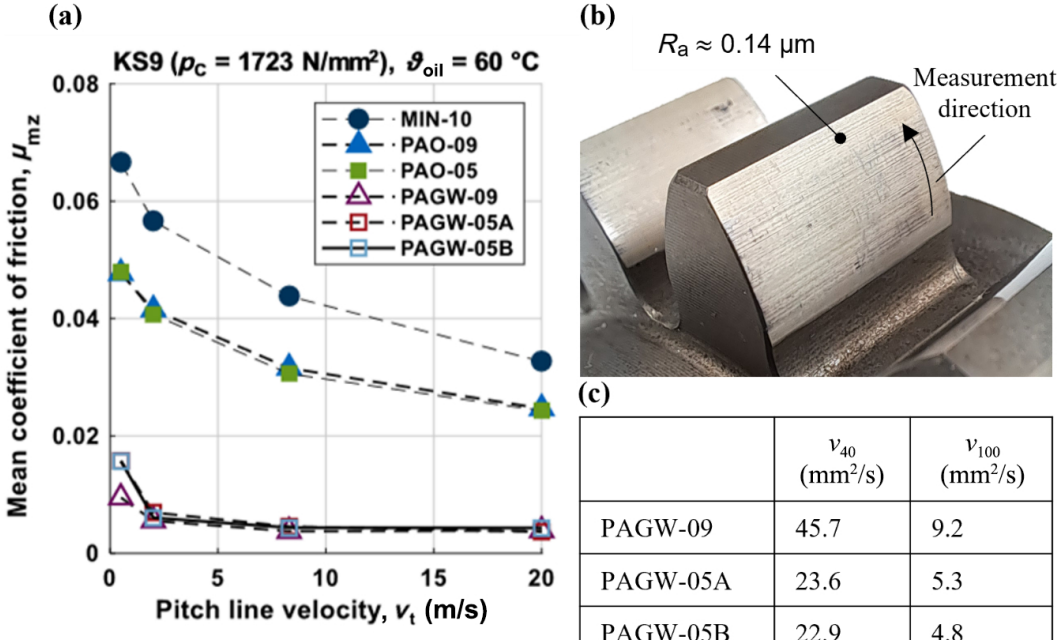


Fig. 1 (a) Coefficient of friction over SRR with various water concentrations of aqueous glycerol and a comparison with rapeseed oil. (b) Schematic picture of the considered ball-on-disk tribometer with ground disk. Reproduced with permission from Ref. [13] for (a), © Elsevier Ltd. 2013.

phenomena on a model scale is the contact geometry and kinematic conditions. Tang et al. [1] stated in their conclusion that in tribometers, severe and long running-in can be associated with wear and a change in contact geometry, and thus a reduction in contact pressure. However, in cylindrical gears, running-in is usually associated only with the wear of the surface asperities, rather than with changes to the tooth flank geometry [30]. The latter would result in a significant change to the load distribution in the loaded tooth contact and lead to transmission errors [31]. In tribometers, a wear-related increase in contact geometry under boundary or mixed lubrication can result in a reduction in contact pressure from the gigapascal to the megapascal range [4, 7, 17]. After wearing-in, a change from elastohydrodynamic (EHL) to hydrodynamic lubrication (HL) regime can occur. This means that elastic surface deformation and pressure-dependent increase in viscosity become negligible. Under HL, superlubricity is more feasible and is also present for mineral oils in journal bearings with pure sliding [32] or soft EHL rolling–sliding contacts with technical polymers [12, 33–35]. Furthermore, Tang et al. [1] rarely addressed the kinematic conditions in terms of the type of motion. Most of the studies cited refer to ball-on-disk configurations and reciprocating motion and/or pure sliding conditions, which do not account for gears with rolling–sliding conditions. The main difference is that pure sliding gives rise to much higher local temperatures, and the reciprocating motion causes the speed to vary from zero, and the contact can therefore go through standstills and repetitive start-ups, which is not representative of gear contacts. Figure 3(a) shows a schematic representation of the gear mesh position of a cylindrical spur gear, including the line load f_N, as well as the radii of curvature $R_{1/2}$ and tangential speeds



	ν_{40} (mm²/s)	ν_{100} (mm²/s)
PAGW-09	45.7	9.2
PAGW-05A	23.6	5.3
PAGW-05B	22.9	4.8

*Extrapolated from 80 °C

Fig. 2 (a) Measured mean coefficient of gear friction over the pitch line velocity for aqueous PAGs (PAGW-09, PAGW-05A, and PAGW05-B) compared to mineral oil and polyalphaolefin (MIN-10, PAO-09, and PAO-05), showing superlubricity in gear contacts. (b) Representative picture of test gear's tooth flank. (c) Kinematic viscosities of the aqueous PAGs. Reproduced with permission from Ref. [9] for (a), © The Author(s) 2019.

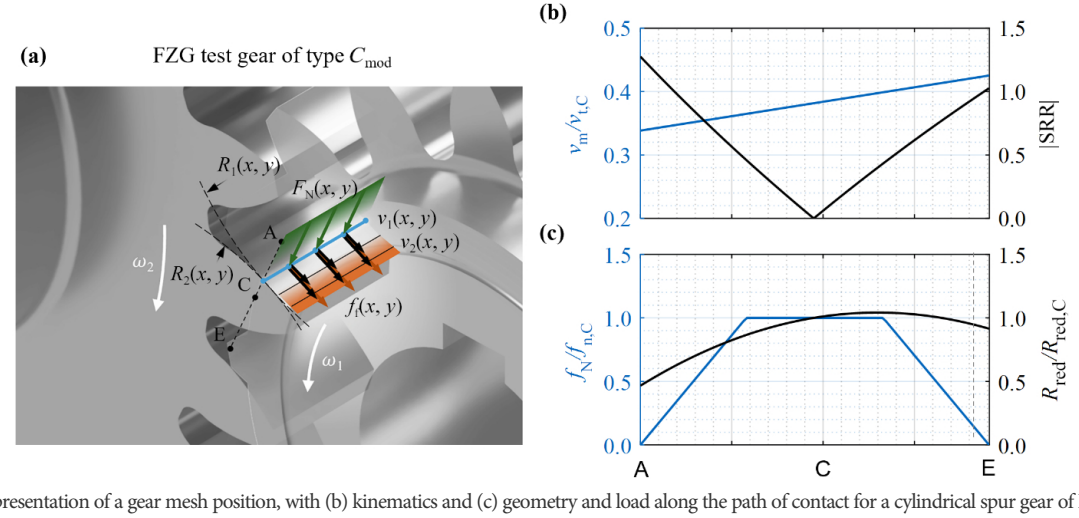


Fig. 3 (a) Representation of a gear mesh position, with (b) kinematics and (c) geometry and load along the path of contact for a cylindrical spur gear of FZG type C_{mod}. Reproduced with permission from Ref. [36] for (a), © ASME 2025.

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$v_{1/2}$ of the tooth flanks. Figures 3(b) and 3(c) show the normalized line load $f_n/f_{n,C}$, normalized normal radius of relative curvature $R_{red}/R_{red,C}$, normalized mean speed $v_m/v_{t,C}$, and slide-to-roll ratio |SRR| over the path of contact. The considered gear geometry is the same as that considered in Ref. [9] in Fig. 2.

Along the path of contact, the load, speed, and geometry are time-dependent. The mean speed increases continuously, while the amount of SRR increases with distance from the pitch point C. Hence, the gear mesh is a rolling–sliding contact with high sliding portions toward the beginning (A) and end (E) of contact. The use of liquid lubricants results in rolling–sliding EHL gear contacts.

It is noted that the original definition of superlubricity by Hirano and Shinjo [37] focused on sliding friction, while the coefficient of friction within rolling–sliding contacts refers to a rolling and sliding proportion. In rolling–sliding EHL gear contacts, the rolling friction is typically very low and sliding friction is dominant [9]. As there is no possibility to measure rolling and sliding gear friction separately, the sliding friction in the gear contact is even slightly lower than the mean coefficient of gear friction. In contrast to gears, the rolling friction at model testing can be ruled out by dynamic calibration as done in Refs. [38–41].

4 Discussion on mechanisms, requirements, and transferability

In their review, Tang et al. [1] provided a schematic representation of the underlying mechanisms of liquid superlubricity, with a focus on the type of aqueous lubricant. However, a primary mechanism resulting in superlubricity for aqueous lubricants under rolling–sliding EHL conditions, typically for gears, is not addressed (Section 4.1). We show that the requirements for achieving liquid superlubricity using engineering surfaces are not that challenging (Section 4.2), and that the transition from superlubricity in gears to superefficiency in gearboxes necessitates a detailed system analysis (Section 4.3).

4.1 Superlubricity with aqueous polyalkylene glycols and glycerol

A notable aspect of liquid superlubricity remains absent from the review of Tang et al. [1], namely, the consideration of aqueous lubricants investigated under rolling–sliding conditions. These lubricants exhibit persistent liquid superlubricity, negating the need for a running-in period, even at contact pressures in the gigapascal range. In addition to aqueous glycerol [13, 42, 43], aqueous PAGs have demonstrated superlubricity under rolling–sliding EHL conditions along with lubricant film formation capability [39–41, 44–47]. As shown in Fig. 2, superlubricity has already been demonstrated in cylindrical gears. The need for ultra-smooth surfaces or low viscosity is not mandatory, but the ability to form an EHL lubricant film between the contacting surfaces is of importance. Therefore, the requirements for achieving liquid superlubricity proposed by Tang et al. [1] in terms of (I) roughness, (II) viscosity, (III) contact pressure, and (IV) velocity are not independent from each other. Its interaction can be illustrated by the EHL film thickness and the lubrication regime, which is influenced by all of the above quantities. The EHL film thickness can be expressed based on a set of three dimensionless parameters for speed (U), material (G), and load (W), or even only two dimensionless Moes parameters M and L [48]. The lubrication regime can be estimated based on the ratio of EHL film thickness to a specific surface roughness parameter [49] and classified into boundary, mixed, and fluid film lubrication. Liquid superlubricity with aqueous PAGs and glycerol can be achieved without running-in period and any change in contact pressure and thus wear even for high contact pressures, wide range of viscosities and low velocities [13, 39]. The underlying mechanism is mainly attributed to the low pressure-viscosity coefficient α resulting in low contact viscosity and thus friction [41, 50]. It should be noted, that a low to moderate functional water content is sufficient to achieve liquid superlubricity [13, 39–41], and that no threshold water content is required [41]. Increasing the water content continuously decreases friction while film thickness simultaneously decreases. Thus, the higher the water content, the lower the surface roughness required for fluid film lubrication. Although the aim should be to achieve fluid film lubrication and avoid surface asperity contacts. It should be noted that superlubricity with aqueous glycerol and PAGs can even be present under calculated mixed lubrication conditions and does not necessarily require fluid film lubrication. However, deviations from typical analytical film thickness formulae were found for aqueous PAGs and glycerol [13, 38, 46] and the calculation accuracy of lubrication regimes needs to be improved [49].

4.2 Liquid superlubricity in rolling–sliding EHL contacts

As mentioned in Section 4.1, liquid superlubricity in EHL contacts with aqueous lubricants based on PAG or glycerol can mainly be attributed to their low pressure-viscosity coefficient α. The latter can be measured directly, e.g., by a high-pressure viscometer [50, 51] or estimated based on optical interferometry [52]. For aqueous glycerol, α is around 4–5 GPa^{-1} [13, 38, 50, 51], while for aqueous PAGs it is around 3–6 GPa^{-1} [39, 41, 45, 50]. This is much lower than for conventional mineral and synthetic oils like polyalphaolefin (PAO), which are typically in the range of 12–26 GPa^{-1} [45]. It should be remembered that the Newtonian viscosity qualitatively has a pressure dependence proportional to $e^{\alpha p}$, and the friction in EHL contacts is controlled by the viscosity at the EHL contact zone, where the pressure (and viscosity) is at its maximum. With α in the range of 3–6 GPa^{-1}, the viscosity at 1 GPa contact pressure only increases 20–400 times compared to α in the range of 12–26 GPa^{-1}, where the viscosity increases several orders of magnitude. Although the exponential dependence of viscosity on pressure presents a simple estimation and the real fluid behaviour may differ [53], it is easy to demonstrate how lubricants with a low value of α cause low friction. Assume that the film thickness is constant over the whole EHL contact and is governed by any film thickness expressions, for example, the one presented by Nijenbanning et al. [54]. Refinement of calculation can be achieved by considering thermal correction factors such as those by Gupta et al. [55]. Further, it is assumed that the pressure distribution follows the Hertz pressure. Then, it is easy to calculate an approximate value of the lubricant film thickness and the theoretical coefficient of friction as a function of the pressure–viscosity coefficient α. The graphs in Fig. 4 were obtained by assuming a maximum pressure of 1 GPa (normal force, F_N = 67 N), ideally smooth steel surfaces with ball-on-disk configuration and a ball radius of 12.7 mm, lubricant viscosity of 50 mPa·s, mean speed of 2.0 m/s, and SRR = 0.5 A detailed description of the calculation approach can be found in the Electronic Supplementary Material (ESM).

It is obvious that this simple estimation predicts superlubricity for α lower than approximately 5 GPa^{-1}, while the lubricant film thickness is only approximately 50% smaller than for conventional mineral and synthetic oils. However, the film thickness can still be enough to achieve fluid film lubrication on engineering surfaces. It

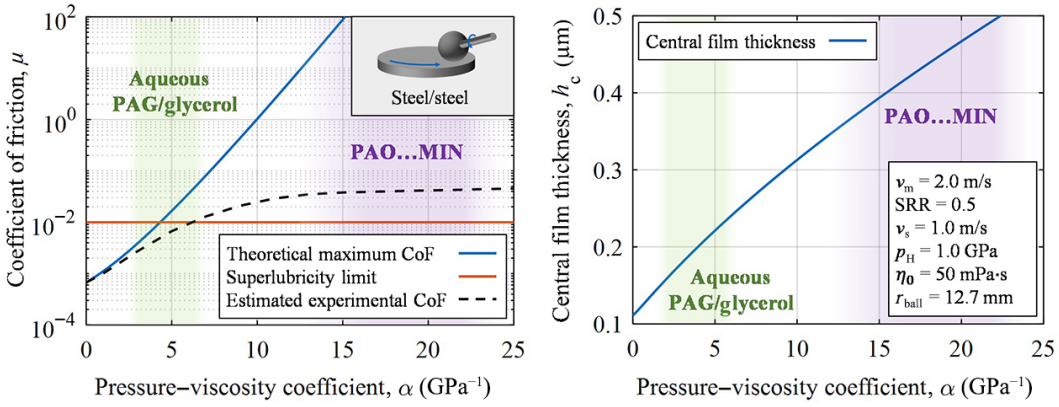

Fig. 4 Estimated coefficient of friction (left) and lubricant film thickness (right) over the pressure-viscosity coefficient in an EHL contact operating at a maximum contact pressure of 1 GPa.

is well-known that EHL friction is not only controlled by bulk viscosity but also, more importantly, by shear-thinning of the lubricants. Shear-thinning occurs even for the aqueous lubricants. By using ReaxFF molecular dynamics, we have been able to compute viscosity for glycerol in the range of 10^5–10^9 s^{-1} [56, 57]. It is shown that the viscosity starts to decline already at 10^5 s^{-1}. In Fig. 4, the shear rate is around 5×10^6 s^{-1}, where shear-thinning makes the viscosity drop one order of magnitude for glycerol [57]. This further reduces the coefficient of friction well below the superlubricity limit. However, shear-thinning also limits the friction for conventional mineral and synthetic oils and may lead to ultra-low friction, as theoretically shown by Habchi and Bair [58]. Furthermore, thermal effects lead to an increase in contact temperature and thus lower the viscosity, which also reduces friction, especially for high SRRs [59]. Consequently, the estimated experimental friction is drastically lower compared to the theoretical friction based on Newtonian fluid behaviour and isothermal conditions. Finally, it should be noted that pure glycerol and water-soluble PAGs can offer ultra-low friction [13, 41, 43], so the addition of water may not even be required to achieve liquid superlubricity: Water-soluble PAGs and glycerol have hygroscopic properties [41, 60], so a certain water content will be set depending on the ambient conditions.

4.3 Towards superefficiency

It is imperative to acknowledge the significant energy-saving potential associated with friction reduction through the implementation of superlubricity in practical applications. However, it is important to note that superlubricity in gearboxes only directly affects the load-dependent power losses by reducing interfacial friction in loaded contacts of machine elements. In addition to load-dependent power losses, load-independent power losses occur due to interaction of fluids with rotating machine elements, e.g., churning and squeezing. In the case of (aqueous) PAGs or glycerol, the higher density of approximately 30%–40% compared to mineral oil or polyalphaolefin [9, 40, 46, 47] can lead to higher load-independent power losses. The higher density results in higher dynamic viscosity, favoring EHL film formation and potentially leading to a comparable film thickness for aqueous PAGs compared to conventional lubricants of the same ISO VG [39]. However, at high speeds, the potential of superlubricity to drastically reduce load-dependent power losses may be overcompensated by higher load-independent power loss, as shown for aqueous PAGs by Yilmaz et al. [9] and Hofmann et al. [61]. Consequently, the path from superlubricity to superefficiency in gearboxes requires consideration of specific boundary conditions and targeted low-loss technologies [62]. In this manner, the very low load-dependent power losses and therefore heat generation of aqueous PAGs and glycerol may allow the use of injection or minimum quantity lubrication.

5 Conclusions

From our point of view, transitioning superlubricity from the laboratory-scale to machine elements does not necessarily require exotic materials and liquids, which are rarely used in industrial applications. We have shown that liquid superlubricity with aqueous PAGs and glycerol is already possible in rolling–sliding contacts and gears [9]. The primary mechanism leading to liquid superlubricity can be attributed to the low pressure-viscosity coefficient α, which continuously decreases with increasing water content. The optimal water content depends on the application and presents a trade-off between low friction, high film thickness, and heat transfer. A low surface roughness and high speed may allow the use of higher water contents and/or low lubricant viscosity. For specific applications and boundary conditions, the addition of water may not even be required to achieve liquid superlubricity.

We believe that further research should address the known challenges associated with aqueous PAGs and glycerol, the severity of which increases in proportion with the water content:

(1) Improvement of corrosion inhibitors: Cost-effective steel components can be sensitive to corrosion when lubricated with aqueous PAGs [51, 63].

(2) Increase of roller bearing lifetime: Typically used roller bearing steel AISI52100 shows drastically reduced service lifetime when lubricated with aqueous PAGs or glycerol and requires the use of costly materials [64].

(3) Ensure that the specified lubricant formulation is maintained over the lifetime: Water evaporation can significantly affect friction, film thickness, and the concentration of additives [40]. Therefore, methods to monitor and control the water content should be investigated.

(4) Understand the failure modes of aqueous PAGs and glycerol. It is seen in investigations on glycerol that wear rate can be higher than for oils even if the theoretical conditions (film parameter) are held constant [65, 66].

(5) Improve the determination quality of the lubrication regime to support the aqueous lubricant design based on practical requirements. It is seen that the traditional film parameter may not be useful. A more accurate way to create comparable conditions between aqueous and conventional lubricants would be to apply the Λ^* theory [49].

(6) Develop additives for aqueous lubricants, such as anti-wear

additives with similar effects as zinc dialkyldithiophosphate (ZDDP) has in oils.

Overcoming these challenges will facilitate the transfer of superlubricity with aqueous PAGs and glycerol to industrial applications. Further, the low power loss associated with superlubricity results in low component temperature and will allow the implementation of further low-loss technologies such as minimum quantity lubrication [67] to achieve superefficiency [62].

Author contributions

Stefan Hofmann: Writing—original draft preparation; Marcus Björling, Roland Larsson, and Thomas Lohner: Writing—review and editing: Thomas Lohner and Karsten Stahl: Supervision.

Availability of data and materials

All data used for this comment are included within the publication or published within the cited literature.

Declaration of competing interest

The authors have no competing interests to declare that are relevant to the content of this article. The author Roland Larsson is the Editorial Board Member of this journal.

Electronic Supplementary Material

Supplementary material is available in the online version of this article at https://doi.org/10.26599/FRICT.2025.9441208.

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