Tribofilm Formation of Simulated Gear Contact Along the Line of Action

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Abstract

In this paper, an experimental simulation method was used for evaluating the tribofilm formation in rolling/sliding contact at different points in the line of action. A ball-on-disc test method was employed by which the pressure and slide to roll ratio of gear contact could be simulated. In order to reach a general conclusion, four different oils and two surface roughness were involved in the experiments. The tribofilm evolution was captured using spacer layer interferometry method, and the correlation of tribofilm with the location at the line of action was studied. Results showed that there is a threshold pressure for the tribofilm formation around which the tribofilm growth rate is maximum. Above this threshold pressure, the tribofilm formation is not stable, and the wear is dominant. Below this threshold pressure, the tribofilm growth rate rises by increasing the pressure and the gear contact is safely protected by a stable tribofilm.

Graphic Abstract

Keywords Tribofilm · Environmentally acceptable lubricant · Gear · Lubrication

1 Introduction

In gears, the teeth are in rolling/sliding contact. When the gear is rotating, the contact condition changes at different locations at the teeth. At the locations near the pitch point, sliding is lower and the pressure is higher. While at the tip of the tooth, the slide to role ratio increases. This variation

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of the contact condition results in different frictional properties, and the appearance of different failure types at the specific points. Pitting is more common at the locations near the pitch point where the pressure is maximum, and the scuffing is found in the tooth tip where the sliding and temperature are the highest. Therefore, for understanding the performance and lifetime of the gears, it is critical to explore the tribological condition of each point on the gear line of action.

FZG test rig is widely used in gear oil testing such as studying frictional behavior of ester and mineral-based oil [1], the effect of base oil type and additives on the pitting in gears [2], and the energy efficiency of mineral base gear oils [3]. Twin disc test is also widely used to study the gears lubrication regarding the effect of surface roughness on film thickness and pressure distribution [4], scuffing initiation caused by the overload [5], pitting [6–8], and wear [9]. Kleemola and Lehtovaara used a twin disc machine to simulate the friction, temperature, and lubrication conditions of gear contact along the line of action [10]. It was indicated that twin-disc measurements can be used for simulating the change of lubrication conditions and the friction behavior trends in real gears [10]. Ball-on-disc test also has been used for simulating the gear contact. It is cheaper, and easy to work with. Björling et al. employed the ball-on-disc test rig for generating the friction maps and used these maps for estimating the friction in a gear set. The results showed that the ball-on-disc test can rank the oils' frictional properties as FZG test did [11]. Regarding the gear application, the ball-on-disc test rig has been widely used also for empirical simulation of the polymer gears [12], friction of the gear oils in different lubrication conditions [13–17], micro pitting [18], and scuffing [19–21]. Tribofilm plays an important role in failure prevention in gear contact, however, to this time, there has not been any paper investigating the tribofilm variation at the gear line of action.

The tribofilm formation has been studied by different techniques. Electrical contact resistance has been used for detecting the formation of anti-wear tribofilm [22], and AFM technique revealed the influence of contact pressure on the tribofilm growth [23]. Other techniques such as XPS and infrared spectroscopy have been also used for this purpose [24], however, spacer layer imaging method (SLIM) is gaining popularity among the other techniques. It is an in situ optical interference method that is quick and enables the accurate estimation of the tribofilm thickness [25]. This technique has been used for studying the influence of slide to roll ratio on the ZDDP tribofilm formation [26], the influence of lubricant on white etching cracking [27], characteristics of the phosphorous and non-phosphorous antiwear films [28], roughness and thickness of ZDDP films [29], and investigating the tribofilm formation during the scuffing test [19, 20]. However, this technique has not been used to

specifically study the tribofilm formation mimicking individual contacts during gear meshing.

In this study, a ball-on-disc machine was used to simulate the gear contact along the line of action. The objectives were to investigate the tribofilm thickness evolution and the factors that control the existence of this film in a simulated gear contact. For the simulation of a gear contact, an experimental method was used that tests different points along the line of action with their specific pressure and slide to roll ratio [10]. Picturing the tribofilm at each point, the ball-on-disc test rig was equipped with the SLIM. Four different industrial oils were used to capture the influence of the oil type. In addition, the specimens were manufactured with two different roughness to make sure the results can be generalized to the real components. By investigating the tribofilm evolution in a gear contact, the influence of geometry and surface quality of the gears can be better understood. This can contribute to the gear design, adjusting the working condition and the oil formulation for the gears.

2 Experiment Detail

2.1 Experimental Rig

The tribofilm measurement tests were carried out using a mini-traction machine that provided the rolling/sliding contact between a ball and a disc. Figure 1 shows a

Fig. 1 Schematic of MTM

schematic view of the test rig and ball/disc specimens. The friction force is measured by a load cell mounted between the ball shaft and the instrument body. The ball and disc speeds are controlled independently to achieve a wide range of lubricant entrainment speed and slide to roll ratio. Lubricant entrainment speed, sliding speed, and slide to roll ratio (SRR) are expressed in Eqs. (1), (2), (3):

$$U_e = \frac{U_d + U_b}{2} \quad (1)$$

$$U_s = U_d - U_b \quad (2)$$

$$\text{SRR} = \frac{U_s}{U_e}, \quad (3)$$

where U_d and U_b are respectively the disc and ball circumferential velocities in the contact point, U_e is the entrainment velocity and U_s is the sliding velocity.

During the tests, the pot and lubricant temperatures are monitored by two thermometers placed respectively in the pot wall and lubricant. Adjusting these two temperatures, a heater and a cooling circulating fluid are connected to the pot.

The tribofilm evolution is recorded by a technique called spacer layer imaging (SLIM). At different stages of the tests, the ball is loaded against a spacer layer of transparent silicon dioxide coated with a thin, semireflective layer of chromium. Using a white light source, a colored interference image of the contact is formed and recorded by the camera. The evolution of these interferometry images reveals the tribofilm formation in different stages and conditions [30]. The tribofilm thickness can be calculated according to the technique shown in Ref. [17].

Table 1 Ball and disc average roughness parameters

Specimen		S_a (nm)	S_q (nm)
Ball		8	10
Disc	Smooth	8	10
	Rough	125	173

Table 2 Properties of the used lubricant

	Kin. vis. @40 °C (mm ² /s)	Kin. vis. @100 °C (mm ² /s)	ρ @15 °C (kg/m ³)	VI	Comment
Oil A	148.2	19.1	970	146	Synthetic gear oil, EAL
Oil B	150	15	897	100	Mineral gear oil
Oil C	150	14.7	890	97	Mineral gear oil
Oil D	127.6	13.83	908	105	Mineral engine oil

2.2 Test Specimen

The ball had a diameter of 19.05 mm. The ball and disc specimens were both AISI 52100 steel with a hardness of 750–770 HV and elastic modulus of 207 GPa. All the balls had the same surface roughness, while two different roughness was considered for the disc to investigate the effect of surface roughness (Table 1). In Table 1, S_a is the average roughness height of area, and S_q is the Root-Mean-Square roughness height of area. The roughness values were measured using a Wyko NT1100 optical profilometer. For each test, a new ball and disc were used, and they were cleaned by immersion in toluene and isopropanol in an ultrasonic bath for 10 min.

2.3 Tested Lubricants

Four different oils were tested. Three of them were gear oils that belong to the 150 VG viscosity class. One of these three oils was an environmentally acceptable synthetic oil. Additionally, another mineral engine oil was selected which has a similar 40 °C kinematic viscosity, and it is practically used in ships for gear lubrication. All the oils except oil D that is an engine oil, comply with the DIN 51517 part 3 (CLP) standard. The oils specifications can be found in Table 2.

2.4 Experimental Procedure

Kleemola and Lehrovaara used a twin-disc device for the experimental simulation of gear contact along the line of action [10]. Their spur gear had a center distance of 91.5 mm, gear ratio of 1, normal module 4.5 mm, face width of 20 mm, contact ratio of 1.45, pressure angle of 20°, and profile shift of 0.176. With these specifications, the pressure and SRR of a contact point along the line of action can be estimated as Fig. 2.

In order to study the tribofilm evolution in a gear set, four different points on the line of action are tested (Fig. 2). The tests for each point include running for 120 min under the boundary lubrication, with the specific film thickness of 0.08 for the rough disc, and 0.46 for the smooth disc. The tribofilm images were recorded at different intervals of 15, 30, 45, 60 and 120 min. The tests are performed once with

Fig. 2 Maximum Hertzian pressure and SRR along the line of action for a specific gear [10]

the rough disc specimen (S_q 156 nm) and repeated with the smooth disc specimen (S_q 10 nm). The points on the line of action are marked in Fig. 2 as Points 1–4. Also, an additional point is tested which is numbered 5*. This point is not theoretically on the line of action, and it is tested for comparing the effect of pressure and SRR. By comparing this point with Point 4, the effect of SRR can be investigated, and by comparing it with Points 2 and 3, the effect of pressure can be explored. Table 3 shows the parameters of the tested points and the test conditions:

3 Results and Discussion

3.1 Effect of Pressure and SRR

At this stage of experiments, the rough disc specimen was used. For each oil, the tests of 2 h running were performed under the condition specified in Table 3. The tribofilm evolution of the oils A and B is shown in Figs. 3 and 4. In these images, four different points on the line of action are tested

(Point 1–4 in Figs. 2 and 3). Also, an additional point is tested which is numbered 5*. This point is not theoretically on the line of action, and it is tested for comparing the effect of pressure and SRR. By comparing this point with Point 4, the effect of SRR can be investigated, and by comparing it with Points 2 and 3, the effect of pressure can be explored.

In Figs. 3 and 4, for the case of Points 1&2, the tribofilm was very thin, and it was hard to be measured due to the wear. However, for Points 3, 4 and 5*, the tribofilm thickness was measured, and the thickness is given in Fig. 5. Points 4 and 5* have the same pressure but different SRR, however, these two points have almost the same tribofilm growth rate in Fig. 5. Thus, it can be said that the SRR has a minor effect on the tribofilm thickness at a constant sliding distance. The same result was found by Shimizu and Spikes for ZDDP additive, and it was concluded that for the same contact pressure, the tribofilm formation does not depend on the SRR [26].

In Figs. 3 and 4, by comparing the Point 5* versus 3 which have the same SRR, the influence of contact pressure can be revealed. By increasing the maximum hertzian pressure from 0.87 GPa (point 3) to 1.02 GPa (point 5*), the tribofilm thickness grows. This illustrates the effect of contact pressure on the tribofilm growth rate. Zhang and Spikes showed that the shear stress (friction coefficient $\times$ contact pressure) controls the rate of tribofilm formation of ZDDP [31], and Spikes claimed that this can be generalized for the other additives according to the Stress-augmented thermal activation theory [32]. Table 4 presents the shear stress and friction force for the case of oils A and B tested with the rough disc. The shear stress of oils C and D were very close to oil B. From Table 4, it is observed that oil A has the highest friction force and shear stress at all the points. Considering its highest tribofilm thickness, this is in agreement with the Stress-augmented thermal activation theory. On the other hand, from Table 4, the shear stress of the Point 5* is higher than that of Point 3 that again approves the Stress-augmented thermal activation theory. However, by comparing the Points 5* versus 2, the tribofilm thickness unexpectedly decreases by increasing the shear stress. Thus, very high shear stress

Table 3 Test conditions

	Point 1	Point 2	Point 3	Point 4	Point 5*
Maximum Hertzian pressure (GPa)	1.21	1.24	0.87	1.02	1.02
SRR %	6.1	42.5	42.5	110	42.5
Entrainment speed (mm/s)	150				
Temperature °C	100				
Duration (min)	120				
Tribofilm measurement intervals (min)	15, 30, 45, 60, 120				
Specimens	Rough disc (S_q 173 nm) and smooth ball (S_q 10 nm) Smooth disc (S_q 10 nm) and smooth ball (S_q 10 nm)				

Fig. 3 Tribofilm evolution of the oil A at different points in the line of action

results in the wear on the specimen's surface. It shows that there is an optimum pressure value in which the tribofilm thickness is at maximum. Gosvami et al. observed the same feature, and suggest that at very high pressures, the wear becomes dominant and prevents the tribofilm growth [23]. This high wear can be clearly observed in Figs. 3 and 4, at Point 2. This pressure can be named “tribofilm threshold pressure”, above which the wear prevents a stable tribofilm growth, and below which the tribofilm thickness can be estimated by the Stress-augmented thermal activation theory.

For the case of oils C and D (Figs. 6 and 7), the wear was dominant in all the points and no stable tribofilm could be measured. This means that there is a considerable amount of asperity penetration that removes the formed tribofilm. Thus, for these two oils, the pressure is too high, or in the other words, above the tribofilm threshold pressure.

3.2 Effect of Roughness

In order to investigate the influence of roughness, the smooth disc specimen was also used, and the experiments in Table 3 were repeated. For the case of oil A, no

considerable tribofilm was formed during 2 h (Fig. 8). The pressure was the same as the tests with the rough discs, and the shear stress of point 2 was around 65 MPa that is near to the amount of shear stress at point 3 in Table 4 for the rough disc. However, no considerable tribofilm or wear was observed. Since there was no wear, it means that the tribofilm removal rate has been very low. Therefore, the low growth rate is due to the small shear stress in the asperity contacts that was not high enough to drive the mechanochemical reaction of the additive molecules. Khaemba et al. used the same specimens and showed that despite the Hertzian pressure being the same, the average asperity pressure in the smooth specimen is around two times lower than in the rough specimens [33]. This emphasizes that the tribofilm threshold pressure is attributed to the pressure at the asperity level. Thus, for oil A, the asperity pressure in the smooth disc was far below the threshold pressure.

For the case of oil D (Figs. 9, 10), a different result was observed. Using the smooth disc (Fig. 9), a thick tribofilm was formed, while no tribofilm has been observed before for the rough disc (Fig. 7). For the case of rough surface test, the wear and tribofilm removal were so high in oil D.

Fig. 4 Tribofilm evolution of the oil B at different points in the line of action

Fig. 5 Tribofilm thickness vs sliding distance for the oils A and B, at different points (rough surface)

Table 4 The shear stress and friction force of the oils A and B at different points

	Shear stress (MPa)/friction force (N)				
	Point 1	Point 2	Point 3	Point 4	Point 5*
Oil A	81.47/6.53	100.64/8.42	68.92/2.89	77.79/4.47	83.60/4.80
Oil B	71.50/5.73	81.32/6.81	59.81/2.51	70.76/4.07	74.00/4.25

Fig. 6 Ball surface after 120 min test with oil C at different points in the line of action (rough surface) **Fig. 7** Ball surface after 120 min test with oil D at different points in the line of action (rough surface) **Fig. 8** Ball surface after 120 min test with oil A using a smooth disc

This means that for the oil D, with the rough disc, the asperity pressure was above the threshold pressure, and with the smooth disc, the points were below the threshold pressure, but near to it.

Therefore, the tribofilm threshold pressure is attributed to the pressure in asperity level. Despite having a similar Hertzian pressure and shear stress, the pressure in asperity level can significantly alter the tribofilm formation mechanism.

The specific film thickness for all the oils was calculated according to the method in Ref. [15]. The specific film thickness for all the tested oils was around 0.08 with the rough disc, and 0.45 with the smooth disc. The lubrication regime is in a boundary regime with very low specific film thickness

which is not common in the gears and it happens at very low pitch velocity conditions. Such a harsh condition was required for capturing the tribofilm formation within a short time. At higher pitch velocities i.e. higher specific film thicknesses, asperity pressure may decrease hence decreasing the tribofilm growth rate.

3.3 Tribofilm Formation Along the Line of Action

The final tribofilm thickness depends on the equilibrium between the rate of tribofilm growth and removal. Gosvami et al. show that the tribofilm growth rate in a single asperity contact is stress-dependent [23], and it fits a stress-activated Arrhenius model:

$$\Gamma_{\text{growth rate}} = \Gamma_0 \exp\left(-\frac{\Delta G_{\text{act}}}{\kappa_B T}\right), \quad (4)$$

where $\Gamma_{\text{growth rate}}$ is the tribofilm growth rate, and Γ_0 a pre-factor, κ_B Boltzmann constant, T absolute temperature and ΔG_{act} is the free activation energy of the rate-limiting reaction step. ΔG_{act} is assumed to be influenced by the stress according to:

$$\Delta G_{\text{act}} = \Delta U_{\text{act}} - \sigma \Delta V_{\text{act}} \quad (5)$$

Here ΔV_{act} is the activation volume and σ the driving stress. Gosvami et al. assumed this driving stress to be pressure [23], but Spikes believes that it is primarily the shear stress [32]. Based on this model, it can be said that the tribofilm growth rate depends on the pressure (or shear stress) in asperity level (σ), and the additives reactivity (ΔU_{act}).

This stress-activated model for tribofilm growth is widely accepted [31, 32, 34], however, the process of tribofilm removal is not yet well understood and there is no reliable equation that can model it. Jacobs and Carpick used the stress-activated model for modeling the rate of atom loss due to the wear [35], and Felts et al. used a similar model for oxygen removal from graphene [36]. Chen et al. used a linear wear model for the tribofilm removal [37]. In that model, the tribofilm removal rate changes with the height of the tribofilm. Azam et al. used a modified Archard wear model with a variable wear coefficient [38]. This is a logical assumption as the hardness of the tribofilm decreases by its height [39]. However, none of these theories are generally accepted and are not able to model the tribofilm removal under the wear condition. If there was a single accepted theory for the tribofilm removal, the amount of the tribofilm threshold pressure could be calculated by equating the tribofilm growth and removal rate at the moment where there is no tribofilm on the surfaces.

According to the above theories, if the asperity pressure is below the threshold pressure, the tribofilm starts to grow and reaches a specific thickness in which the tribofilm growth

Fig. 9 Ball surface after 120 min test with oil D using a smooth disc

Fig. 10 Tribofilm thickness vs sliding distance for the oils D (smooth surface), at different points

rate is equal to the tribofilm removal rate. Thus, the final tribofilm thickness depends on the equilibrium between the tribofilm growth rate and the tribofilm removal rate [31]. This can be seen in Points 3, 4, and 5* for oils A and B (Figs. 3 and 4). After some time, the tribofilm growth rate and removal rate become equal, and the tribofilm height tends to be constant (Fig. 5). On the other hand, if the asperity pressure is above the threshold pressure, the tribofilm removal rate is bigger than the tribofilm growth rate. Thus, no equilibrium is achieved, and there will be constant wear on the specimens. This is the case that is seen in points 1 & 2 in Figs. 3, 4, and Figs. 6, 7 for the case of oils C and D.

A value for the threshold pressure of oil A can be roughly estimated using the finding of Khaemba et al. [33]. With the

same rough disc specimen and smooth ball, Khaemba et al. calculated that the average asperity pressure is 3.9 GPa when the applied pressure is equal to 1 GPa. Such values give the proportion of “real contact area/nominal contact area” equal to 17%. For the case oil A and rough disc, the threshold pressure is between 1.02 and 1.24 GPa. Taking the average of 1.13 GPa for such pressure and considering 17% for the proportion of “real contact area/nominal contact area”, the average asperity pressure is 4.4 GPa. This number is very rough estimated, however, it is comparable to the pressure found by Gosvami et al. [23] for the condition in which wear becomes dominant.

In conclusion, the tribofilm thickness is highly influenced by a specific threshold pressure above which the wear is dominant. Below this threshold pressure, the tribofilm growth rate increases with the pressure. It is very important to note that this pressure is at the asperity level. The tribofilm growth dependence on the pressure, and the threshold pressure can be schematically illustrated as in Fig. 11.

Now, the tribofilm growth rate on this simulated gear contact can be explained. Here the “rate” means nanometer tribofilm per meter sliding distance (Fig. 5). It was discussed that the SRR does not have a considerable influence on tribofilm growth, and it has been verified by another study [26]. Thus, the variation of the tribofilm growth rate along the line of action depends mainly on the pressure.

Considering a specific point on the line of action, the tribofilm growth rate (nm/m) depends on its relative pressure to the tribofilm threshold pressure (Fig. 12). Accordingly, there are three possible scenarios for a gear set:

1. The pressure on the points in the line of action are all below the threshold pressure (case of Fig. 9)

Fig. 11 Schematic of the tribofilm growth rate vs asperity contact pressure

Fig. 12 The tribofilm threshold pressure and the pressure distribution along the line of action for a specific gear

2. The pressure of some points is above, and some points below the threshold pressure (case of Fig. 3)
3. The pressure on the points in the line of action are all above the threshold pressure (case of Fig. 7)

Figure 11 shows these three scenarios. For any one of these cases, the points which are closer to the threshold pressure have the highest tribofilm growth rate (nm/m). Above the threshold pressure, wear is dominant and prevents a stable tribofilm formation. Below the threshold pressure, a stable tribofilm can be formed, but the growth rate depends on how far it is from the threshold pressure. The optimum condition is when all the points are below the tribofilm threshold pressure because the surface can be protected by the formation of a stable tribofilm. According to the discussions, the location of the threshold pressure depends on asperity pressure and the reactivity of the tribofilm, and it can be changed by the roughness, the base oil, or the reactivity of the additives.

In conclusion, it is hard to say that the tribofilm growth rate is higher in the pitch point or the tip of this simulated gear tooth. It is important to pay attention to the unit of tribofilm rate in this context which is nm of tribofilm per meter of sliding distance. This means that in gears, the actual formation time of tribofilm should decrease when moved from pitch point towards tip of the tooth due to linear increase of sliding speed (assuming constant pressure). The higher sliding speeds at the tip zone of gear tooth causes also higher temperature at this zone [10], which was kept within a specific limit in this study and needs further studies. On the other hand, the gear loading may vary largely in different applications, and the Hertzian pressure levels used in this study represent typical values. When the ground gears flanks are new, probably the gear tooth is running above the threshold pressure. However, after the running-in period, the asperity peaks are smoothened, and the conditions are less harsh and more similar to what was tested in this study. Super finished gear surfaces may be near to the roughness levels presented in smooth surfaces in this study. Noticeably to mention that the SLIM technique is hard to be employed for very rough surfaces, and another tribofilm measurement technique should be used to mimic rough (hobbed) surface gear application. However, the results in this paper present insight into the tribofilm formation in gear contact and can be used in future studies.

4 Conclusion

The objective of this work was to investigate the tribofilm film thickness evolution in a simulated gear contact. Besides, it was important to explore the factors that control the existence of tribofilm at different tribological conditions. A ball and disc test equipment provided sliding/rolling contact. Several tests were performed with the specific pressure and slide to roll ratio to mimic the conditions at different points at the line of action. For each test, a separate set of tribofilm images was recorded by the SLIM technique. Four different industrial oils were tested, and the specimens were manufactured in two surface roughness.

The results showed that:

- The tribofilm formation does not depend on the slide to roll ratios in the range of this simulated gear which has low entrainment speed and low temperature rise at different locations along the line of action
- There is a tribofilm threshold pressure around which the tribofilm growth rate is maximum.
- The tribofilm threshold pressure is very sensitive to the surface roughness; thus, it is attributed to the pressure in asperity level. Despite having a similar Hertzian

pressure and shear stress, the pressure in asperity level can significantly alter the tribofilm formation mechanism.

- Above this threshold pressure, the tribofilm formation is not stable, and the wear is dominant. Below this threshold pressure, the tribofilm growth rate rises by increasing the pressure.
- Considering a specific point on the line of action, the tribofilm growth rate mainly depends on its relative pressure compared to the tribofilm threshold pressure. The points which are closer to the threshold pressure have the highest growth rate. The points with higher pressure are prone to damage.

Finding the exact asperity threshold pressure is hard because there is not an accepted theory or model for the tribofilm removal mechanism. However, using this experimental method, the threshold pressure can be found for any combination of gear design, material, and oil.

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Author Contributions RB: Conceptualization, Methodology, Software, Validation, Formal analysis, Investigation, Resources, Data Curation, Writing—Original Draft, Writing—Review & Editing, Visualization. AL: Conceptualization, Writing—Review & Editing, Supervision, Project administration.

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Data Availability The authors confirm that the research in this article is original and that all the data given in the article are real and authentic.

Declarations

Conflict of interest The authors declare no potential conflicts of interest with respect to the research, authorship and/or publication of this article, and we have fully respected the research ethics principles.

Ethical Approval The authors confirm that neither the entire article nor any of its parts have been previously published and is not under consideration elsewhere. Being accepted, this article will not be published elsewhere in the same form, in English or in any other language, without the written consent of the Publisher. Furthermore, the authors declare no potential conflicts of interest with respect to the research, authorship and/or publication of this article, and we have fully respected the research ethics principles.

Consent to Participate All authors (Reza Bayat and Arto Lehtovaara) have contributed to preparing the submitted manuscript entitled “Tribofilm formation of simulated gear contact along the line of action”. No other person has significantly contributed to its preparation.

Consent for Publication This article has the full consent of all authors, and all authors have approved the manuscript and its submission and publication on the journal of Tribology Letters.

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