

On grease redistribution, inlet replenishment, starvation, and cage effects in rolling contacts: Insights from LED-induced fluorescence

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Abstract:

Lubricant supply at the inlet of rolling contacts is critical in elastohydrodynamic lubrication (EHL), as it governs starvation and film thickness. In grease-lubricated

bearings, replenishment is controlled by grease properties and bearing components. However, due to the lack of direct visualization, the starvation process and its mechanisms remain unclear. Especially the influence of the cage in lubricant transport and redistribution and its role in regulating inlet grease supply are still not fully understood. In this study, the LED-induced fluorescence (LED-IF) technique was developed to analyze grease distribution at the inlet in a ball-on-disc setup, enabling direct observation of starvation and its mechanisms. The residual grease layer height H_a was quantified and correlated with the dimensionless meniscus length m to construct a lubrication supply map (H_a - m), which was further used to analyze the effect of cage clearance on grease redistribution and replenishment. Results show that grease is mainly supplied via a stringing mechanism. Over time, grease becomes fragmented, preventing the reformation of strings, thus limiting replenishment and causing starvation. EHL film thickness depends not only on the residual grease layer height but also on its spatial distribution, whereby the H_a - m distributions show reasonable consistency with film thickness results. Cage clearance substantially affects the residual grease layer height by altering grease redistribution. An appropriate clearance enhances grease entrainment, while the optimal value depends on operating conditions and grease properties. Excessively large and small clearances reduce replenishment. Therefore, lubrication is governed by the combined effects of entrainment speed, grease type, and residual grease layer height.

Key words:

Rolling Bearing; LED-IF; Elastohydrodynamic Lubrication; EHL; Starvation; Redistribution

1. Introduction

Rolling bearing performance and service life are strongly influenced by lubrication conditions. Under elastohydrodynamic lubrication (EHL), the formation of a sufficiently thick lubricant film is essential to separate contacting surfaces, thus, reducing frictional torque as well as surface-related wear and failure mechanisms. Among the influencing factors, the amount of lubricant available at the EHL inlet directly determines the degree of starvation and, therefore, controls the film thickness that can be generated within the contact region [1-3].

In grease-lubricated rolling bearings, the lubricant supply process differs substantially from that in oil lubrication. Greases exhibit a yield stress and tend to remain displaced once squeezed out of the raceway by rolling elements, making replenishment to the contact region more difficult. The return flow of the lubricating base oil is mainly driven by the surface tension forces acting at the free surface and around the rolling elements, while being resisted by lubricant viscosity [4, 5]. At high entrainment speeds, the replenishment of lubricant towards the raceway center may become insufficient, leading to starvation and a substantial reduction in film thickness [6]. Previous studies have shown that the actual film thickness in rolling bearings lubricated with grease is often smaller than the film thickness predicted for the base oil alone, indicating that the contacts are essentially operating under starved lubrication conditions [7]. Starvation has been identified as an important cause of lubrication failure in many bearing systems and is particularly likely under conditions with low temperature, high rotational speed, or large aspect ratio contacts [8].

Early investigations of starvation were mainly conducted using simplified point-contact configurations, such as ball-on-disc setups, focusing on film thickness and replenishment mechanisms. As such, Chiu [4] showed that starvation increases with rolling speed and viscosity due to insufficient inlet replenishment and highlighted the key role of surface tension in film recovery. Cann et al. [9, 10] demonstrated that film thickness under starved conditions is governed by the amount of oil available for reflow and the lubricant's ability to replenish the contact zone. Furthermore, a quantitative analysis of lubricant loss in grease-lubricated EHL contacts established a simplified relationship between film decay and the number of overrolling cycles, while identifying the key replenishment mechanisms. These studies indicate that film

thickness under starvation is governed by the balance between lubricant loss and replenishment, decreasing with overrolling cycles until a steady state is reached. In fundamental ball-on-disc experiments, surface tension and capillary forces provide the main driving mechanisms, whereas lubricant viscosity and entrainment speed tend to suppress the replenishment process [11, 12]. However, under starved conditions, the replenishment of lubricant into the contact zone is limited as it cannot be sufficiently sustained solely by the passive backflow of grease from the side bands along the raceway. If the lubricant replenishment mechanism observed in ball-on-disc models is used to predict lubrication behavior in entire bearings, it is mostly falling short as practical bearings feature additional structural and operational factors that contribute to the replenishment of grease into the rolling track beyond the basic backflow mechanism. Specifically, the motion of rolling elements (including rolling and spinning), vibration, and cage-induced lubricant redistribution play a crucial role in transporting lubricant into the contact zone and maintaining film formation [12-15].

The cage is a key component in rolling bearings, primarily used to guide and separate rolling elements. However, increasing evidence suggests that the cage and variations in cage clearance and cage geometry also play an important role in lubricant transport and redistribution inside the bearing [16-18]. Damiens et al. [17] reported that reducing cage clearance may scrape lubricant from the inlet region under oil lubrication, whereas it may enhance film thickness through grease redistribution under grease lubrication. Cann and Lubrecht [18] further pointed out that cage design may either improve or deteriorate lubricant redistribution, thereby determining the performance limits of grease-lubricated bearings. In addition, grease within the cage pocket may undergo rolling and shearing processes, which can promote base oil bleeding and alter the local lubricant supply to the contact region [15, 16].

Despite the recognized importance of the cage for lubricant transport, experimental studies directly linking cage-induced lubricant redistribution with inlet replenishment of the contact remain limited, particularly under grease lubrication conditions. Most studies have focused on cage motion [19] and lubrication behavior of the cage pocket to ball contact region [20] whereas Liu et al. [21] experimentally demonstrated that cage replenishment significantly improves lubrication under certain amplitudes in oscillating bearings. These investigations have shown

that cage replenishment is an important mechanism for maintaining lubricant supply along the raceway. Other influencing factors, such as ball spin [18], may also contribute to lubricant replenishment; however, these effects were not the primary focus of the present study. Furthermore, optical visualization techniques and multiphase computational fluid dynamics (CFD) methods have increasingly been applied to investigate lubricant flow and distribution inside bearings and cage regions [21-27]. For example, experimental observations have demonstrated that cage motion strongly affects lubricant migration within ball bearings, suggesting that cage-controlled lubricant transport can fundamentally alter inlet supply conditions [19]. However, most of these studies have been conducted under oil lubrication conditions. For grease-lubricated bearings, the problem becomes more complex due to the presence of a thickener network structure, and pronounced non-Newtonian and viscoelastic behavior as well as history-dependent characteristics of grease. As a result, grease supply, replenishment, and starvation processes at the contact inlet are more difficult to characterize. In addition to capillary driven return flow, grease transport is affected by channeling, churning, grease migration, and oil bleeding from the thickener network. Furthermore, the opaque nature of grease makes direct optical visualization challenging. Consequently, the mechanisms by which cages influence grease redistribution and inlet supply remain insufficiently understood, and the quantitative relationship between inlet grease layers and contact film thickness has not yet been established, especially for non-uniform lubricant distribution. Therefore, systematic studies on cage induced grease redistribution and its influence on inlet grease supply are required to improve the understanding of grease replenishment and to support bearing design optimization [15].

Optical interferometry and fluorescence visualization techniques have become powerful tools for investigating lubricant distribution in rolling and bearing contacts. Jin et al. [28] combined optical interferometry with fluorescence labeling to directly observe grease film formation in sliding-rolling concentrated contacts, revealing clear differences between grease films and base oil films. Tohyama et al. [29] used multi-wavelength interferometry and fluorescence imaging to simultaneously measure oil film thickness and distribution in deep groove ball bearings operating at high speeds. Chen et al. [23] introduced transparent bearing structures to visualize

lubricant flow inside ball bearings and further proposed a pixel-dependent fluorescence quantification method to measure lubricant distribution near rolling contacts [30]. Liang et al. [19] developed a visualization model bearing with a glass outer ring and used fluorescence tracing to investigate lubricant redistribution inside bearings. Their results showed that re-lubrication is driven by a combination of capillary flow and mechanical redistribution. Aamer et al. [31] combined flow visualization experiments with multiphase CFD modeling to analyze lubricant flow behavior in the cage region of ball bearings.

Although these studies have greatly improved the understanding of lubricant transport in oil-lubricated bearings, direct visualization of grease redistribution at the contact inlet remains scarce. Most existing research on grease lubrication has focused on film formation in contacts or indirectly inferred grease supply capability. Moreover, Liu [32] showed that cage clearance, together with the thickness and spatial distribution of the residual grease layer on the rolling elements determines the lubricant supply at the contact inlet and, thus, governs the resulting film thickness. However, this understanding is largely based on indirect evidence, as the spatial distribution of grease in the inlet region has rarely been directly observed. In particular, key features, such as the residual grease layer at the inlet and the geometry of the lubricant reservoir, remain underexplored. As a result, the quantitative relationship between inlet grease layer characteristics on the surface contact pair and reservoir size of contact region has not yet been established, which limits the accurate evaluation of starvation severity and replenishment capability.

In this study, an LED-induced fluorescence (LED-IF) technique is employed to visualize the spatial distribution of grease in the inlet region during a ball-on-disc experiment, enabling direct observation of grease migration and redistribution. To address the limitation that existing studies are largely confined to qualitative descriptions and lack quantitative characterization of grease behavior in the inlet region, the present work achieves a quantitative measurement of the residual grease layer height at the inlet and establishes its correlation with the inlet meniscus length in the contact region. On this basis, starvation due to limited grease supply at the inlet in ball-on-disc contact is investigated, focusing on the role of the cage in grease supply and redistribution. By combining residual grease layer height, inlet meniscus length, and film

thickness, the redistribution and replenishment capability under different cage clearances are evaluated. It also provides insights into grease replenishment and guidance for cage design and lubrication evaluation where direct film thickness measurement is not feasible.

2. Materials and methods

To investigate the spatial distribution of residual grease at the contact inlet and its relationship with the inlet meniscus length, the lubricant film thickness was measured using a custom-made optical lubrication film test-rig depicted in Fig. 1. The AISI 52100 steel ball was supported by two V-shaped bearings and was, thus, allowed to freely rotate while a loading unit pressed it against a BK7 glass disc. The working surface of the disc was coated with two layers: A bottom Cr layer of approximately 20 nm with 20% reflectivity to achieve interference fringes with high contrast as well as a top SiO₂ layer of 200 nm thickness. The experimental parameters are listed in Tab. 1. Before each experiment, a nominal grease volume of 2 mL was applied using the same syringe and distributed as uniformly as practicable along the complete circumference of the cleaned disc track. The same application procedure was used throughout the study to standardize the initial grease charge and placement. Two greases were employed within the scope of this study, namely PAO-Li-2-50 and PAO-Li-1-50 (DGMK 810, FUCHS LUBRITECH GmbH, Germany). Both featured a polyalphaolefin (PAO) base oil with 50 mm²/s kinematic viscosity at 40 °C as well as a Li-based thickener with additive package RC9505. PAO-Li-2-50 was of NLGI grade of 2 (cone penetration 265–295 × 10⁻¹ mm), whereas PAO-Li-1-50 was of NLGI class 1 (cone penetration 310–340 × 10⁻¹ mm) according to DIN 51818. According to DIN 51817, the oil bleeding rates of PAO-Li-2-50 and PAO-Li-1-50 were calculated based on the mass percentages of the separated oil and were found to be 1.103% and 5.608%, respectively. The worked cone penetration and additional physical and rheological properties of the two greases are summarized in Tab. 2, the entrainment velocity was varied between 16 and 512 mm/s in discrete steps and different cage designs were employed to study the influence of cage clearance and grease redistribution at the contact inlet, see Fig. 1. The tested clearances were equal on both sides of the cage. To facilitate visualization, the cage front end included a through-hole structure which resulted in an incomplete scraping effect at the front edge of the cage, whereas the rear cage pocket, aligned with the entrainment direction

ensured that the cage pocket generated a clear scraping effect on the grease. Each test was terminated after the film thickness had remained at a near-zero level for a sufficient period to confirm sustained starvation, i.e., after a sufficient number of revolutions N , thereby minimizing the risk of wear-related changes to the optical measurement surface. After each test, the glass-disc contact region was visually inspected. No pronounced wear tracks were observed; only occasional minor local scratches were present. Where necessary, the contact position was adjusted slightly in the radial direction before the subsequent experiment to avoid these local surface imperfections, while remaining within essentially the same nominal radial region of the disc. As is common practice in optical EHL testing, the coated glass disc was reused for multiple experiments provided that the optical surface remained suitable for measurement.

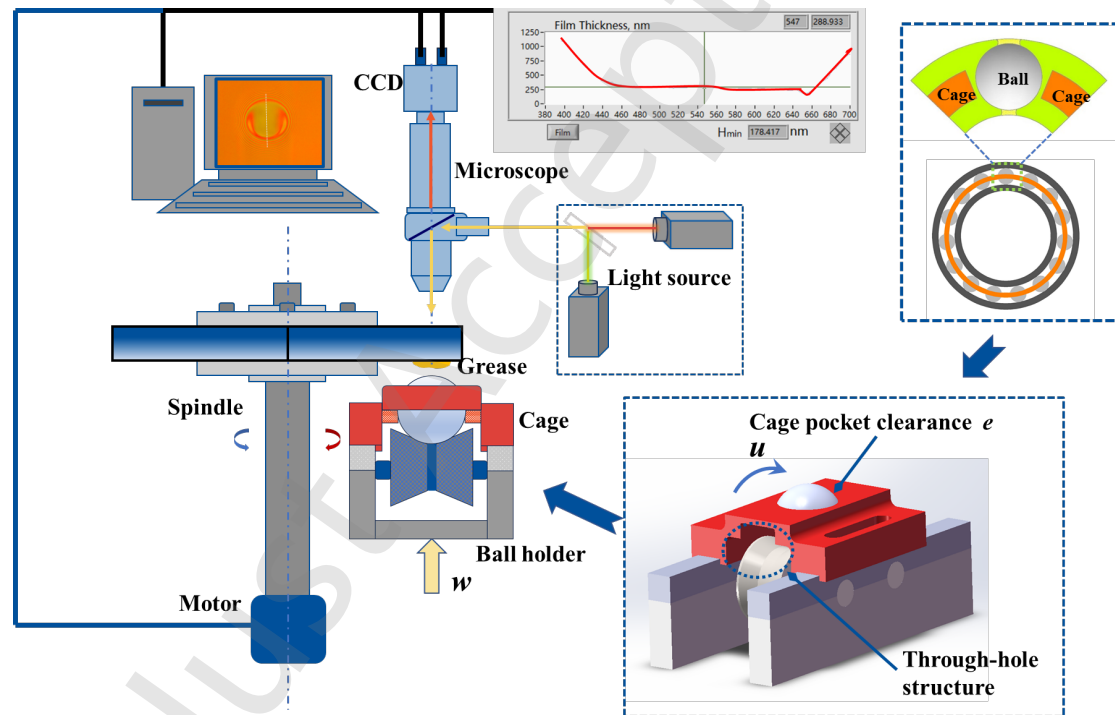


Fig. 1. Schematic of the adapted optical lubrication film test setup according to [33].

Tab. 1. Summary of experimental parameters.

	Parameter	Values
Fixed	Disc diameter D_d	150 mm
	Disc thickness h_d	15 mm
	Disc surface roughness, $R_{a,d}$	4 nm
	Ball diameter, D_b	25.4 mm
	Ball surface roughness, $R_{a,b}$	14 nm
	Normal load, w	30 N
	Nominal Hertzian pressure p_H	518 MPa
	Hertzian contact half width a_H	164 μm

	Ambient temperature T	22 ± 2 °C
	Relative humidity RH	25 ± 5 %
	Grease volume V	2 mL
	Greases	PAO-Li-1-50, PAO-Li-2-50
Varied	Cage clearances e	0.10, 0.20, 0.25, 0.35, 0.50 mm
	Entrainment velocities u	16, 96, 256, 384, 448, 512 mm/s

Tab. 2. Main physical and rheological properties of the tested greases.

Parameter	PAO-Li-2-50	PAO-Li-1-50	Source/Test methods
Base oil type	PAO	PAO	Supplier report
Thickener type	Li-based	Li-based	Supplier report
Additive package	RC9505	RC9505	Supplier report
NLGI grade	2	1	DIN 51818
Worked cone penetration	$265\text{--}295 \times 10^{-1}$ mm	$310\text{--}340 \times 10^{-1}$ mm	DIN ISO 2137
Kinematic viscosity of separated oil phase at 40 °C	50 cSt	50 cSt	Supplier report
Kinematic viscosity of separated oil phase at 100 °C	7.8 cSt	7.8 cSt	Rheometer, DIN 51810-1
Density of the base oil at 40°C	0.87 g/cm ³	0.87 g/cm ³	Supplier report
Oil bleeding rate	1.103%	5.608%	DIN 51817
Oscillatory amplitude-sweep response, G' , G'' , δ , yield strain	See Fig. S3(a)	See Fig. S3(b)	Rheometer, DIN 51810-2
Oscillatory amplitude-sweep response of dyed grease, G' , G'' , δ , yield strain	See Fig. S4(a)	See Fig. S4(b)	Rheometer, DIN 51810-2
Apparent viscosity-shear rate behavior of grease at RT	See Fig. S5	See Fig. S5	Rheometer, DIN 51810-1
Apparent viscosity-shear rate behavior of grease at RT of dyed grease	See Fig. S5	See Fig. S5	Rheometer, DIN 51810-1

Note: RT denotes room temperature.

A thin EHL film formed between the ball and the disc during operation, which was precisely measured using a dichromatic interference intensity modulation (DIIM) technique as described in [34] and the central film thickness h was evaluated every 100 revolutions of the glass disc. Furthermore, as illustrated in Fig. 2, the grease distribution was observed using the LED-IF technique, with fluorescence images captured by an industrial camera (acA1440-73gc, Basler,

Germany) under illumination from a commercially available LED coaxial light source (8 mm, blue light, adjustable intensity; YEUBXDJ, China), from both the front view via the through-hole and the top view through the glass disc. Owing to the optical configuration, the top- and front-view fluorescence images could not be acquired simultaneously and were therefore obtained in separate runs under the same nominal conditions. The two views provided complementary observations and were not treated as statistical replicates. For each nominal condition, two runs were performed. Owing to incomplete synchronized image acquisition in one run, only the run containing the complete film thickness and fluorescence dataset was used for quantitative analysis. The dataset was selected solely on the basis of data completeness and not on the observed lubrication behavior. The grease layer height, H , was determined from the fluorescence intensity at each lateral position. The local average grease layer height, H_a , was then calculated as the average value of H within a lateral region of 350 μm around the contact. The grease observed at this location originates from the previous cycle, i.e., it first exited the contact region, passed through the rear cage pocket, and then reappeared at the observation position. The top view was used to measure the inlet length L of the lubricant reservoir in the contact region. This was defined as the distance from the contact center to the oil-air meniscus along the rolling direction (see Fig. 2). The meniscus position was identified from the background-corrected fluorescence images with the assistance of the corresponding intensity profiles. A universal fixed raw-intensity threshold was not used; instead, the local background level was used as a reference to distinguish the grease-covered region from the grease-free region. To facilitate comparison, the inlet length was normalized by the Hertzian contact half axis width, yielding the dimensionless inlet length[1]

$$m = \frac{L}{a_H}. \quad (1)$$

Within the scope of this study, the fluorescent dye Coumarin 6 (98%, Sigma-Aldrich, USA) with maximum absorption (excitation) wavelength of 444 nm, fluorescence emission peak at ca. 505 nm and a corresponding Stokes shift of about 61 nm was employed to stain the greases. A 500 nm long-pass optical filter suppressed scattered excitation light near 450 nm while allowing the fluorescence emission signal to pass through. A blue LED provided excitation at 450 nm, and the detected emission wavelength was 550 nm. Coumarin 6 was added to the

greases at 0.02 wt% as a fluorescent tracer. No solvent or additional carrier oil was used during dye incorporation, in order to avoid altering the grease formulation. The Coumarin 6 powder was first thoroughly mixed with a small amount of grease to prepare a concentrated dye-grease premix. The remaining grease was then added gradually in several portions, followed by repeated scraping, folding, and kneading until a visually homogeneous dyed grease was obtained. The dyed grease was stored in the dark for 12–24 h before testing. To evaluate the possible influence of the dye on the grease rheology, both the apparent viscosity–shear rate behavior and the oscillatory amplitude-sweep response of the dyed and undyed greases were compared. No measurable difference was observed within the experimental scatter, indicating that the addition of 0.02 wt% Coumarin 6 did not significantly affect the overall flow behavior or viscoelastic response under the present test conditions. Possible effects on oil bleeding behavior and thickener microstructure were not independently quantified.

The fluorescence images obtained from the front view were further processed to characterize the spatial distribution of grease on the ball surface at the contact inlet. Therefore, the grease layer height along the central cross-section of the y-axis was extracted from the fluorescence intensity distribution as shown in Fig. 2.

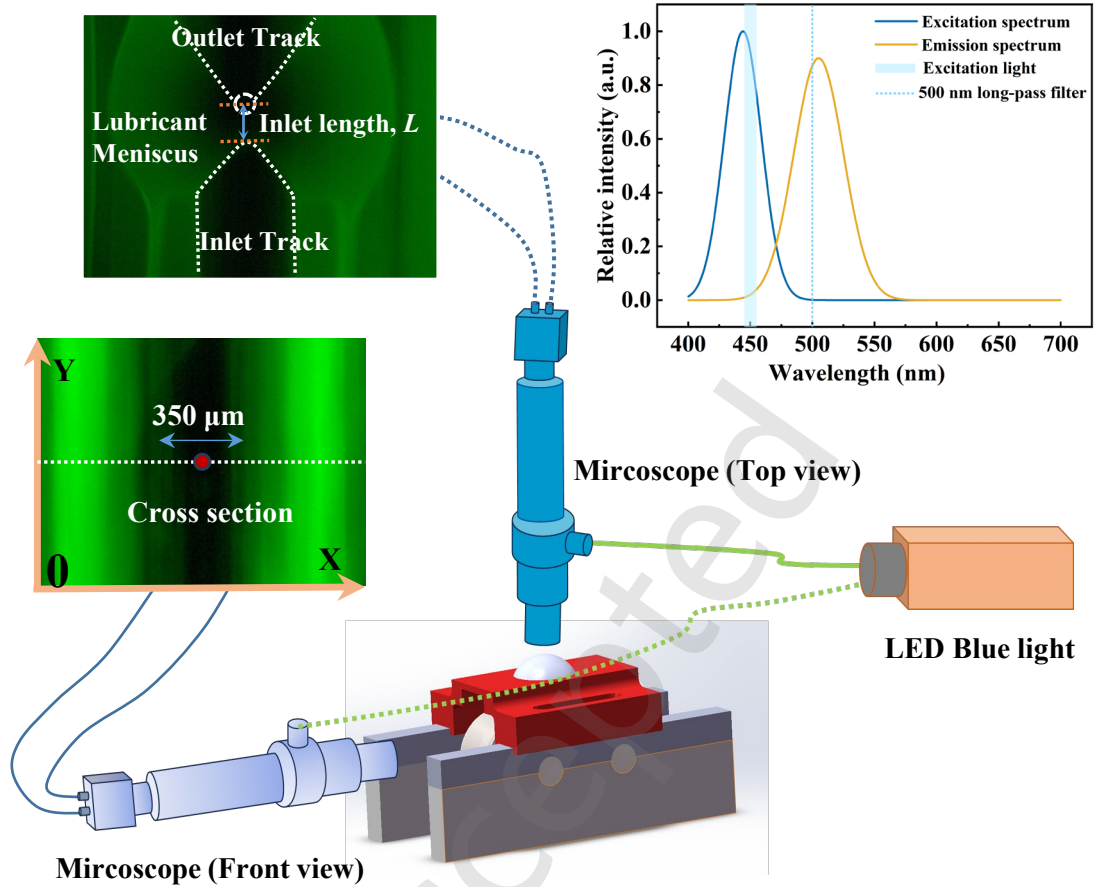


Fig. 2. Schematic diagram of the fluorescence setup.

To enable calibration-based estimation of the residual grease layer height on the steel ball surface at the contact inlet, the fluorescence intensity I was initially calibrated using the same fluorescence observation system. To this end and to improve optical transmission and minimize reflection, a 53 mm diameter anti-reflection glass plate with a transmittance of up to 98% was placed in contact with a circular retaining ring, thereby enhancing the signal-to-noise ratio of the fluorescence measurement, see Fig. 3a and b. To obtain a wide calibration range, four steel balls with different diameters (7.9, 11.9, 15.9, and 25.4 mm) were used, extending the calibration range of grease layer thickness and providing a broad range of geometrically known grease heights of the calibration. Thereby, each ball was paired with a matching retaining ring and the clearance between the ball and the retaining ring was uniformly filled with the fluorescent dye containing grease. The calibration was carried out individually for PAO-Li-2-50 and PAO-Li-1-50 using the same illumination, camera, exposure, optical-filter, and image-processing settings as those employed in the rolling experiments. The sample was illuminated vertically using the excitation light source, and fluorescence images were recorded from the top

view. The images were then sampled uniformly at intervals of 3° in the circumferential direction and 0.1 mm in the radial direction. Based on the known ball diameter, the grease layer height

$$H_i = R - \sqrt{R^2 - d_i^2} \quad (2)$$

at each sampling point was calculated. At each known calibration height, multiple fluorescence-intensity values were obtained from spatial sampling of the calibration images. These values represent spatial samples rather than independent repeat experiments. The distributions are therefore described using the median and interquartile range (Q1–Q3), together with the mean and spatial standard deviation in the Supplementary Material. Conventional confidence intervals based on the number of pixels were not used because they would incorrectly treat spatial samples as statistically independent experimental repeats. Instead, calibration variability was assessed using the observed spatial distributions, residual statistics, and height-dependent prediction uncertainty. The following empirical saturating relationships were used to describe the overall dependence of the recorded intensity on the geometrically known grease height:

$$I = 159.1765 \cdot (1 - e^{-7.20269 \cdot H}) + 100.968 \quad (3)$$

for PAO-Li-2-50, and

$$I = 132.7968 \cdot (1 - e^{-4.5311 \cdot H}) + 122.1574 \quad (4)$$

for PAO-Li-1-50. Here, I is the recorded 8-bit fluorescence intensity in arbitrary units, and H is expressed in millimetres in Eqs. (3) and (4). Equation (3) was evaluated over $0 \leq H \leq 0.5428$ mm, whereas Eq. (4) was evaluated over $0 \leq H \leq 0.5082$ mm.

When evaluated against the individual spatial intensity samples within these ranges, Eq. (3) gave $R^2 = 0.835$ and $RMSE = 23.67$ arb.u. for PAO-Li-2-50, while Eq. (4) gave $R^2 = 0.709$ and $RMSE = 28.33$ arb.u. for PAO-Li-1-50. Evaluation against the mean intensity at each known height gave $R^2 = 0.904$ and $RMSE = 17.52$ arb.u. for PAO-Li-2-50, and $R^2 = 0.829$ and $RMSE = 21.04$ arb.u. for PAO-Li-1-50, as summarized in Table S1. The corresponding low-height calibration behavior and residual distributions are provided in Figs. S1 and S2, respectively. Since the residual scatter is height-dependent, the calibration-related uncertainty was not treated as a single uniform confidence interval for all H values. Instead, its influence on the subsequent

H_a - m analysis was assessed using representative systematic perturbations of ± 20 and ± 40 μm , and the resulting sensitivity ranges are summarized in Table S2.

At large grease heights, the fluorescence signal approached the upper range of the 8-bit camera response, and the high-intensity plateau region was therefore treated with caution. To ensure consistent relative comparisons, the same Coumarin 6 concentration, LED intensity, exposure time, optical configuration, and region-of-interest definition were used for all calibration and rolling experiments. Background contributions were corrected using grease-free regions or reference images acquired under the same optical conditions. No additional flat-field correction was applied. The possible influence of grease oil bleeding on the LED-IF calibration is further discussed in Supplementary Note S1.

Therefore, the established I - H relationship should be regarded as an empirical, system-specific calibration that enables estimation of the local grease layer height within the investigated response ranges. Its uncertainty is height-dependent, and absolute H values should be interpreted with particular caution in the low-sensitivity and signal-saturation regions. Under otherwise identical optical conditions, the calibration remains suitable for comparing the evolution and spatial redistribution of grease among the investigated operating conditions.

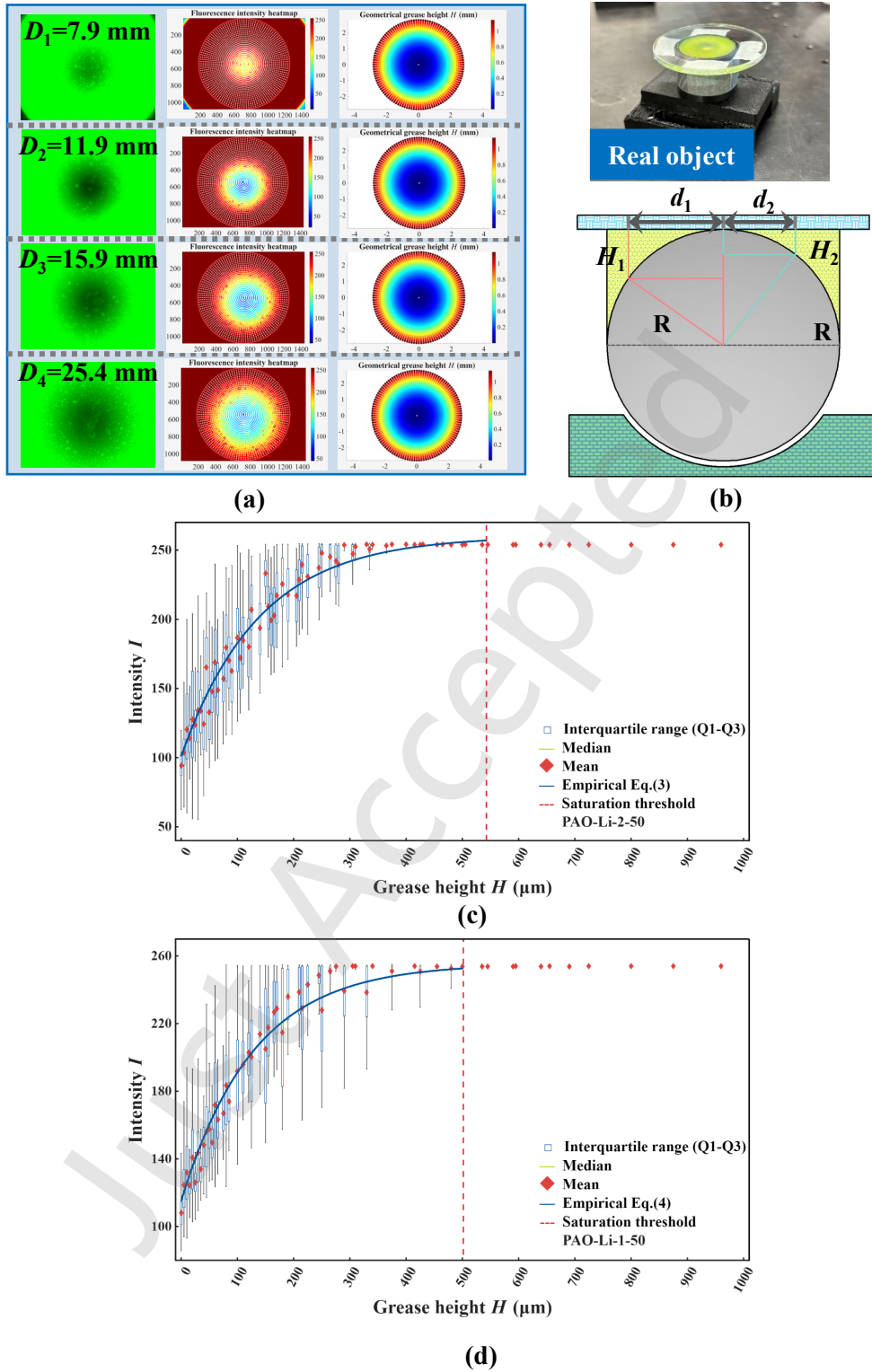


Fig. 3. (a) Sampling points for the top-view images of steel balls with four different diameters as well as corresponding fluorescence intensity and grease height values for PAO-Li-2-50, (b) geometric schematic for calculating the grease film height, and fitted relationship between fluorescence intensity and grease height from the fluorescence calibration procedure for (c) PAO-Li-2-50 and (d) PAO-Li-1-50.

3. Results and Discussion

In the following, the results are first presented for the reference case in section 3.1 without a cage in order to clarify the fundamental mechanisms of grease redistribution, inlet replenishment, and starvation. Subsequently, in section 3.2, the influence of cages with different clearances is analyzed and interpreted using film thickness measurements together with fluorescence-based observations to derive an H_a - m lubrication supply framework.

3.1 Grease distribution at the inlet and within the contact without a cage

As a first reference case, the evolution of the central EHL film thickness with the number of revolutions at different entrainment speeds for grease PAO-Li-2-50 and without cage is shown in Fig. 4. Thereby, severe starvation, i.e. film thickness dropping down to zero, was reached within approximately 400 revolutions for all tested speeds. In the case of lowest speed of 16 mm/s, this already happened after 100 revolutions, while for the moderate speeds of 96 and 256 mm/s starvation set in after 300 revolutions. This indicated that grease replenishment to the contact region is not governed solely by surface tension and capillary forces. Instead, additional factors must exist in the ball-on-disc contact system that influence this behavior.

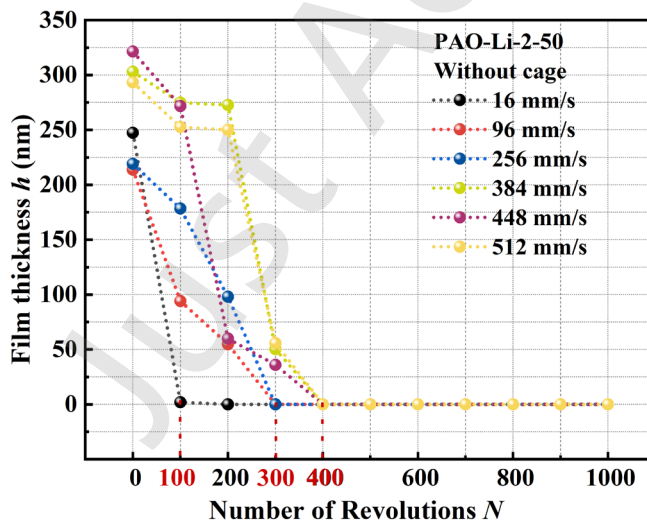


Fig. 4. Central film thickness versus number of revolutions for PAO-Li-2-50 without cage.

To further investigate this process, the grease fluorescence distribution from the top and front view at entrainment speeds of 256 and 512 mm/s are presented in Figure 5a and b, respectively. It can be observed that the grease distribution near the contact region (top view) and along the raceway (front view) was closely correlated with the previously presented film thickness. As

the amount of grease decreased in both views and the distance between the lateral ridges and the contact region increased, the film thickness dropped sharply. At the early stage of operation ($N = 100$), the raceway in the contact region was fully covered with grease. Although excessive grease was squeezed out of the contact by the ball-on-disc interaction, this process was not completed within few overrolling events due to the adhesive and viscoelastic nature of the grease. In addition, grease adhering to the ball and disc surfaces formed a series of lateral ridges near the contact region, which continuously interacted with each other around the smallest gap between the glass disc and ball. These grease bands as shown in the front view images at $N = 400$ and $N = 600$ exhibited a periodic approach–retraction motion with corresponding grease bands on the glass disc around the contact area. This resembled the behavior observed in tackiness tests [35], but the gap between the two surfaces remained constant in the present study. This phenomenon is typically referred to as stringing [32], and the corresponding grease bands are termed stringing stripes.

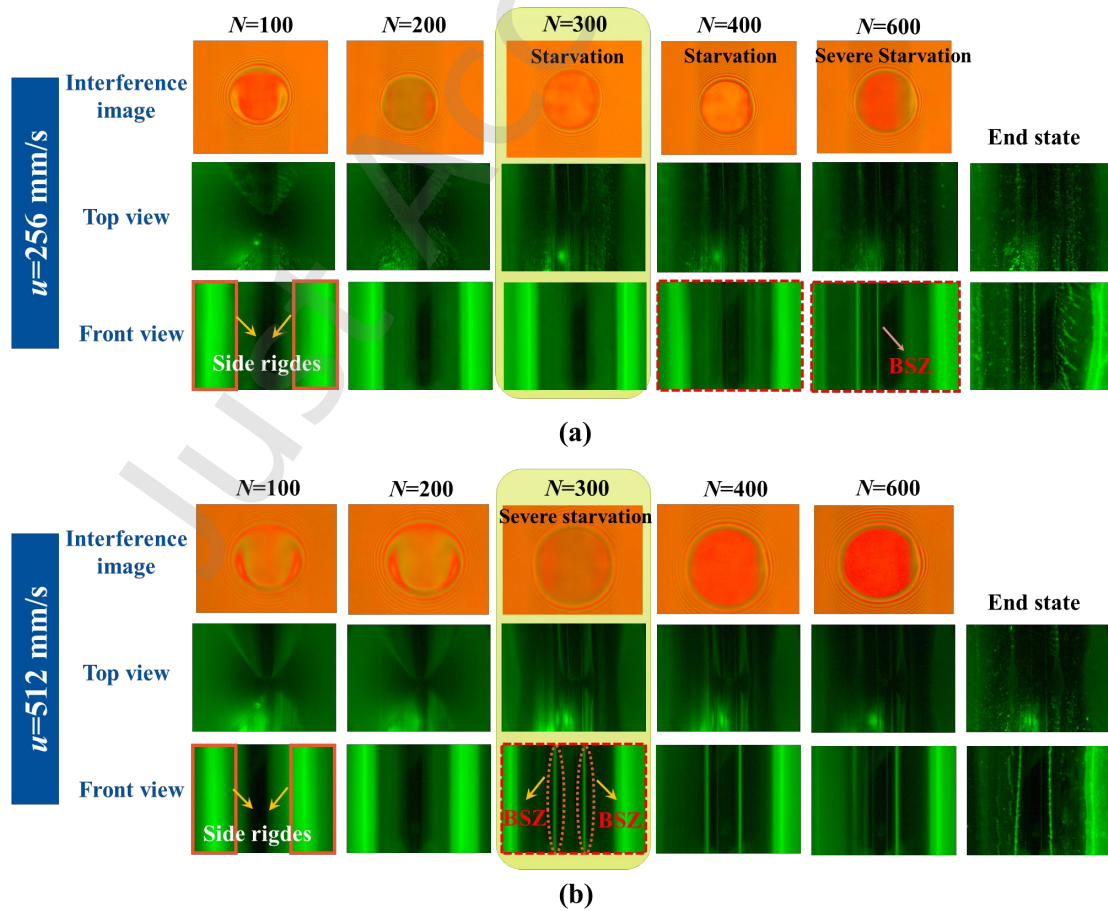


Fig. 5. Interference images as well as top and front view fluorescence images after various

numbers of revolutions for PAO-Li-2-50 without cage at entrainment velocities (a) $u = 256$ mm/s and (b) $u = 512$ mm/s.

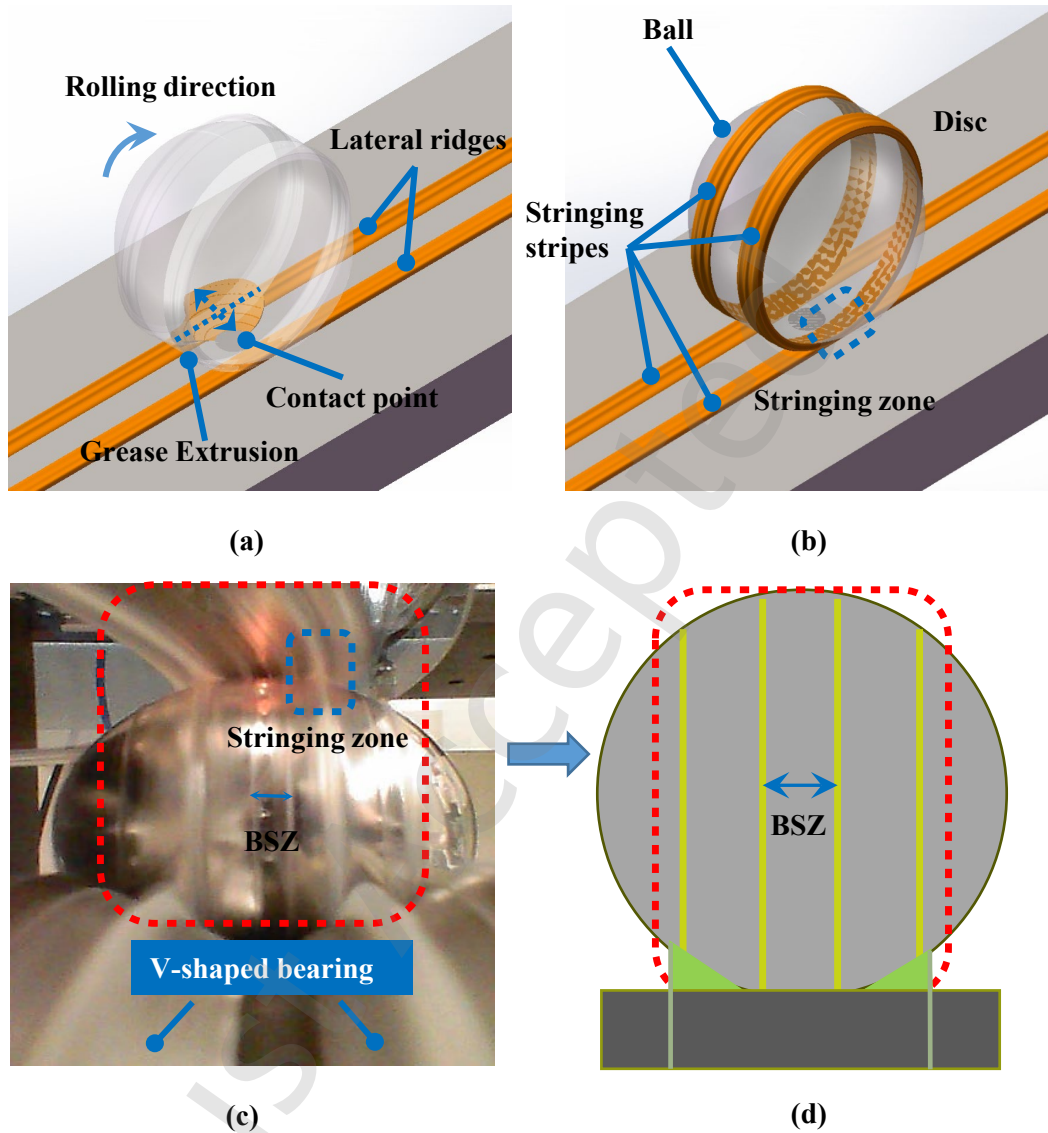


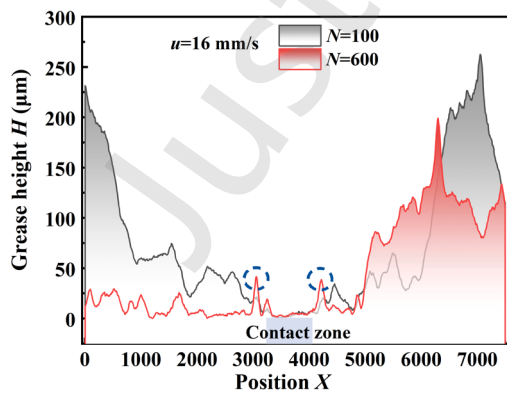
Fig. 6. Schematic illustration of the lateral ridges formation process: (a) Excess grease is squeezed out, forming lateral ridges; (b) Formation of stringing process; (c) Photograph of the stringing zone; (d) Schematic diagram of the BSZ location. Schematic drawings are not to scale.

A schematic illustration of the stringing phenomena is provided in Fig. 6a and b, whereby the stringing zone captured from the front view using a conventional optical camera is depicted in Fig. 6c. Before $N = 300$, a relatively large amount of grease adhered to the ball and disc surfaces. The separated string-like structures observed in the front view are attributed to the coupled effects of surface adhesion, grease cohesion and viscoelasticity, shear/extensional deformation, and local redistribution near the contact. Therefore, string formation, rupture, and local

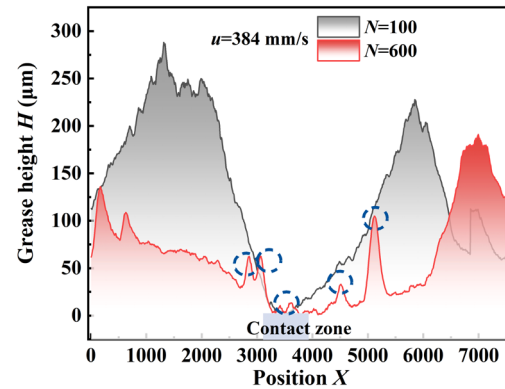
coalescence should be regarded as successive or coexisting stages of grease redistribution rather than mutually exclusive mechanisms. During the stringing process, coalescence of grease led to a temporary increase in local volume, allowing part of the grease to re-enter the contact region and contribute to lubricant supply at the inlet (Fig. 6a). However, the contribution of this process to film formation gradually weakened with increasing number of revolutions, and the measured film thickness therefore continued to decrease, as shown in Fig. 4. This can be attributed to the inherent outward spreading forces associated with contact dynamics, excessive grease was continuously expelled from the contact region, driving the outward migration of the lateral ridges. As the number of revolutions increased, the lateral ridges progressively moved away from the contact, accompanied by a gradual reduction in their volume due to grease loss. Consequently, the volume of grease involved in the stringing process became nearly constant, and no further replenishment to the contact region occurred. The system eventually reached an equilibrium state, as shown in the top view of Fig. 5b at $N = 300$, marking the onset of severe starvation where the film thickness in the contact was close to zero. At this stage, a small amount of grease remained adhered to the ball and disc surfaces, forming the so-called boundary stringing zone (BSZ) [32], whose location is further illustrated schematically in Fig. 6d. Meanwhile, the lubricant reservoir observed in the top view ruptured. Therefore, the formation of the BSZ in the front view provides a useful indicator for identifying the onset of severe starvation.

The spatial distribution of grease at the contact inlet from the initial operating stage $N = 100$ to the final severe starvation state $N = 600$ with different entrainment speeds is shown in Fig. 7. Thereby, X denotes the lateral position in the front-view image, i.e., the horizontal coordinate from left to right across the observed region of the ball surface (dimensionless, used only to indicate relative position). For the considered load case, the nominal Hertzian contact width corresponded to approximately the region around $X = 3750 \pm 300$. At the initial stage, grease was squeezed out of the contact region and formed relatively high lateral ridges on both sides, with heights reaching approximately $300 \mu\text{m}$. As the number of revolutions increased, the stripes in the stringing region underwent periodic approach–retraction motion. During the approach stage, individual stripes merged into larger sheet-like grease bands. Driven by the push-back effect of

grease accumulated on the V-shaped bearings of the ball holder, these bands migrated toward the contact region and were entrained into the contact inlet, thereby contributing to grease replenishment. The influence of the V-shaped holder on the push-back mechanism was assessed separately through comparison with a holder-free dual-driven configuration, as described in the Appendix. The formation and breakup of grease strings originate mainly from the adhesive and viscoelastic response of the grease under the imposed rolling kinematics, whereas their subsequent interception and redirection by the V-shaped bearings constitute an apparatus-associated mechanical contribution. Meanwhile, grease detached from the stringing region and re-adhered to the ball and disc surfaces, corresponding to the retraction stage. It should be noted that the push-back effect of the V-shaped bearings was influenced by the entrainment speed, as can be observed from the grey curves. As shown by the red curve in Fig. 7a, at low speeds, the peak positions of grease on both sides of the contact zone were initially located relatively far from the contact zone. With increasing speed, however, these peak positions shifted closer to the contact zone. This observation explained the experimental results shown in Fig. 4. At higher entrainment speeds, the push-back effect of the V-shaped holder became more pronounced, driving a larger amount of grease on both sides of the contact zone back towards the inlet, where it participated in the stringing process. As a result, a longer effective operating duration was achieved.



(a)



(d)

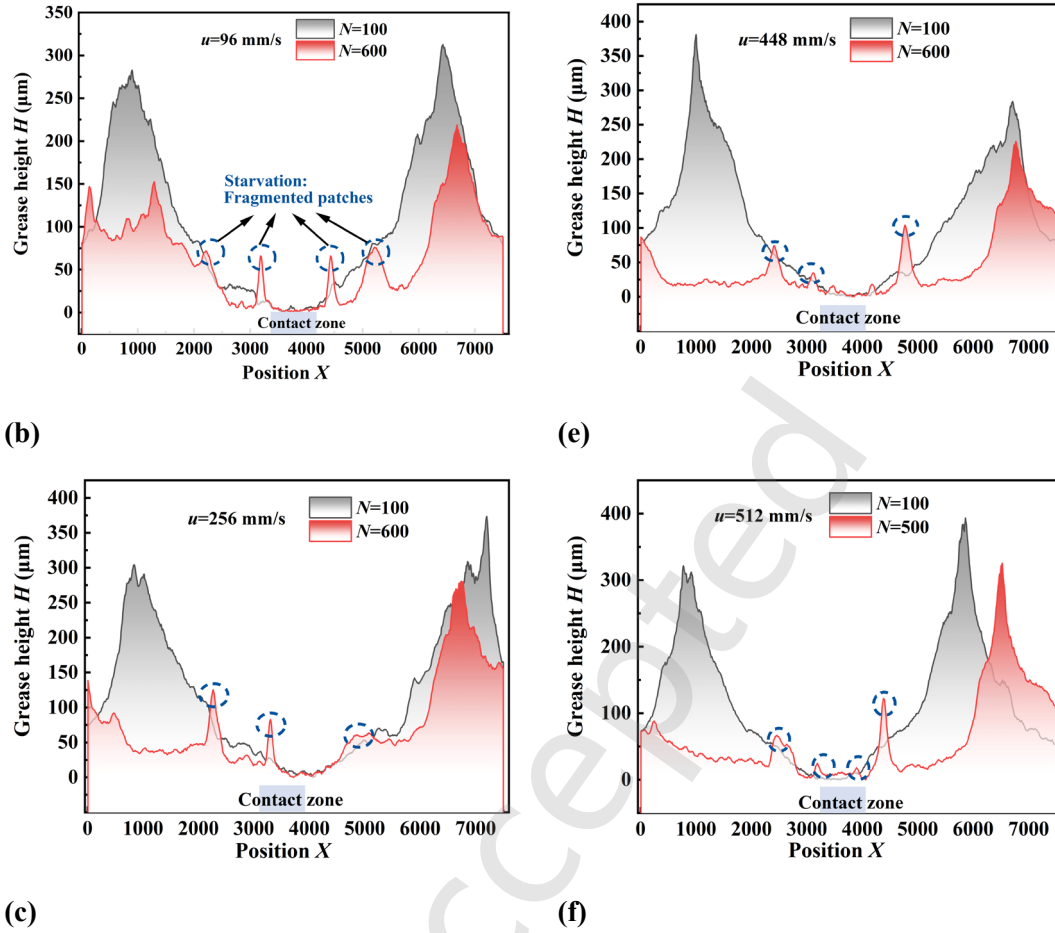


Fig. 7. Grease distribution on the steel ball surface across the inlet of the contact region for PAO-Li-2-50 without cage at entrainment velocities (a) 16 mm/s, (b) 96 mm/s, (c) 256 mm/s, (d) 384 mm/s, (e) 448 mm/s, and (f) 512 mm/s.

As the number of revolutions increased further while the grease supply remains constant, grease on the V-shaped bearings was continuously thrown off due to centrifugal effects and shear thinning. Consequently, the total amount of grease adhering to both the ball surface and the V-shaped bearings decreased. Under these conditions, the grease on the ball could no longer be effectively pushed back towards the contact zone, meaning that no additional grease participated in the replenishment process. The grease distribution at the contact inlet gradually evolved into isolated peak-like structures. After repeated interaction cycles, grease near the contact zone was no longer replenished, and the remaining grease appeared as fragmented residues on the ball. These fragments were unable to reconnect into continuous grease bands and, therefore, couldn't be re-entrained into the contact zone. As a result, severe starvation occurred as indicated by the red curves in Fig. 7. Under these conditions, only a small amount of residual grease remained near the contact zone. This grease predominantly exhibited a

fragmented distribution, forming the BSZ. Therefore, the evolution of grease distribution on the ball surface can be divided into three stages: (i) Lateral ridge formation, (ii) stringing-dominated interaction, and (iii) fragmentation-induced starvation.

Based on the above analysis, the film thickness was governed by both, the volume and spatial distribution of the grease ridges. Therefore, quantitative characterization of the height and position of the residual grease layer on the ball surface using fluorescence based thickness measurements is essential. With such information under different operating conditions, the lubrication state within the contact zone can assist in evaluating.

3.2 Grease distribution at the inlet and within the contact with cage

It is well established that the cage can substantially influence the residual grease layer height at the inlet. Therefore, the lubrication enhancement effect of cages with different clearances was evaluated based on fluorescence measurements of the inlet grease layer height on the ball surface. The evolution of film thickness with the number of revolutions at different entrainment speeds for the grease PAO-Li-2-50 at a cage clearance of $e = 0.2$ mm is shown in Figure 8. Compared with the no-cage results in section 3.1, the film thickness increased substantially, particularly at high speeds, and the effective operating duration before severe starvation was markedly extended. At an entrainment speed of 256 mm/s, severe starvation occurred at approximately $N = 1300$ with the cage, compared to $N=300$ without a cage (Figure 4). At a speed of 512 mm/s, the starvation point increased from $N = 400$ to $N = 1800$.

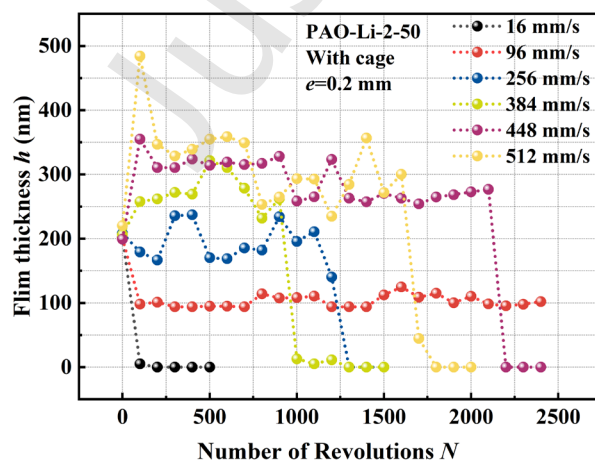


Fig. 8. Central film thickness versus number of revolutions for PAO-Li-2-50 with a cage of clearance 0.2 mm.

Fluorescence images of grease distribution from the two observation perspectives for the PAO-Li-2-50 grease with a cage clearance of 0.2 mm at the entrainment speeds of 256 mm/s and 512 mm/s at different numbers of revolutions are presented in Figure 9a and b, respectively. Compared with the no-cage condition, the effective operating duration before starvation was substantially extended, indicating a clear improvement in lubrication performance in the presence of the cage. The fluorescence images further revealed that the lateral ridges formed on both sides of the contact track during the initial stage were flattened by the scraping effect of the cage and redistributed closer to the contact region. This redistribution facilitated grease supply to the contact inlet. During operation, grease entered the contact zone from the inlet. Under the pressure within the contact area, part of the grease was expelled and formed lateral ridges. As the steel ball rotated, this grease was transported into the cage clearance. Because of the limited space in the clearance, the grease underwent spatial compression and formed a thinner but more uniform layer located closer to the contact zone. Under the scraping action of the cage, this redistributed grease was subsequently carried back to the contact inlet by the rotating ball, resulting in a periodic grease redistribution process.

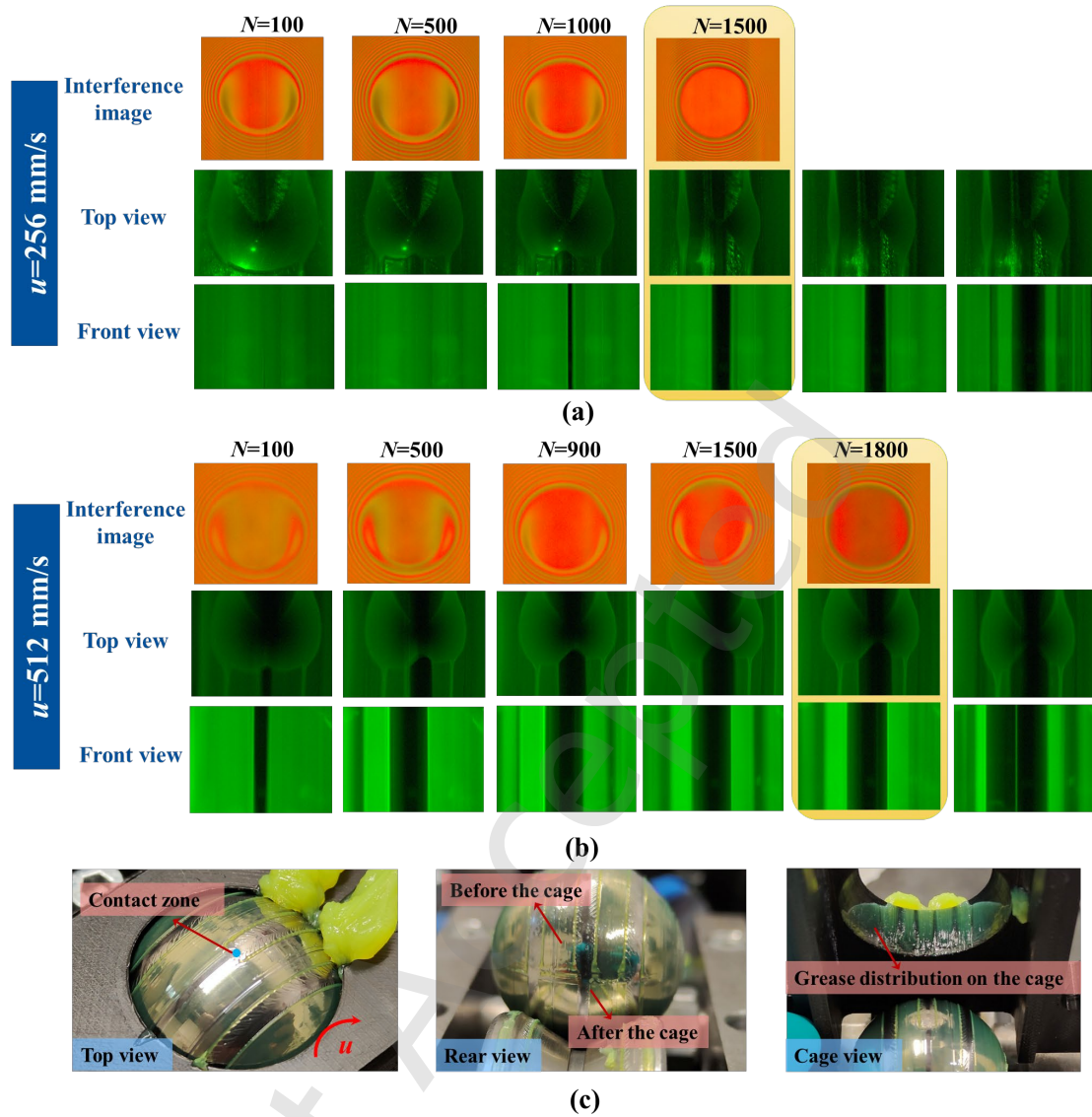
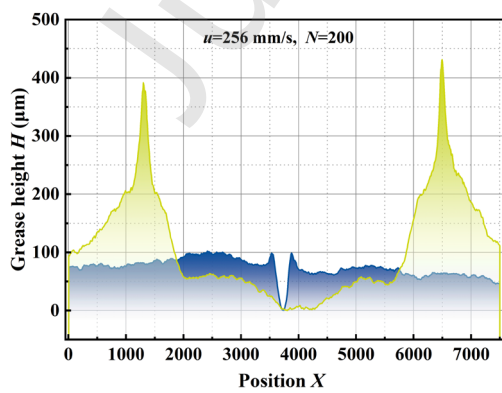


Fig.9. Interference images as well as top and front view fluorescence images after various numbers of revolutions for PAO-Li-2-50 with cage at entrainment velocities (a) $u = 256$ mm/s and (b) $u = 512$ mm/s, (c) Grease distribution on the ball.

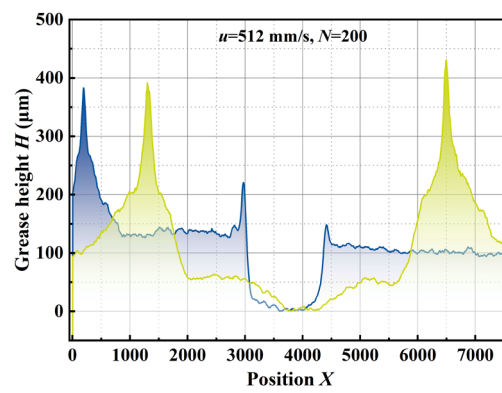
The grease distribution on the steel ball surface and the cage during this process is depicted in Fig. 9c. Grease leaving the contact zone exhibited a dendritic distribution pattern before entering the cage clearance due to the high pressure in the contact zone and spread away from the contact to form lateral ridges. After passing through the cage clearance, the grease was compressed and flattened into a more uniform layer and moved closer to the contact region. In this sense, the cage acted as a scraper, pushing grease towards the center of the contact region and thereby extending the effective lubrication duration. A comparison between the cases with and without the cage indicates that the improvement provided by the cage is substantially more

pronounced than the push-back effect in the V-shaped bearing.

The evolution of grease distribution at the contact inlet on the steel ball surface together with the variation in grease layer height on the ball surface as a function of the number of revolutions is further illustrated Figure 10. The distributions with and without a cage are compared at entrainment speeds of 256 mm/s and 512 mm/s. At the lower speed, during the initial stage ($N = 200$), the lateral ridges without a cage reached a height of approximately 400 μm . In contrast, under the action of the cage, the grease layer was uniformly flattened to about 80 μm and redistributed closer to the contact zone. As the number of revolutions increased, the grease layer height under the cage remained around 100 μm at $N = 500$ and $N = 1500$, although the spatial distribution differed. At $N = 500$, more grease remained near the contact zone, whereas at $N = 1500$ the grease moved more outside from the contact zone, eventually leading to severe starvation. When the entrainment speed increased to 512 mm/s, the grease initially exhibited a relatively uniform distribution. As the number of revolutions increased, however, the distribution gradually became more non-uniform. This behavior can be attributed to the increased height of the lateral ridges formed after contact separation under the influence of centrifugal forces, which intensified with increasing speed. As a result, more grease was removed by the cage and no longer contributed to lubrication. At the same time, the distance between the grease and the contact zone increased with increasing revolutions, eventually leading to severe starvation. Nevertheless, the onset of starvation occurred substantially later than in the no-cage case.



(a)



(d)

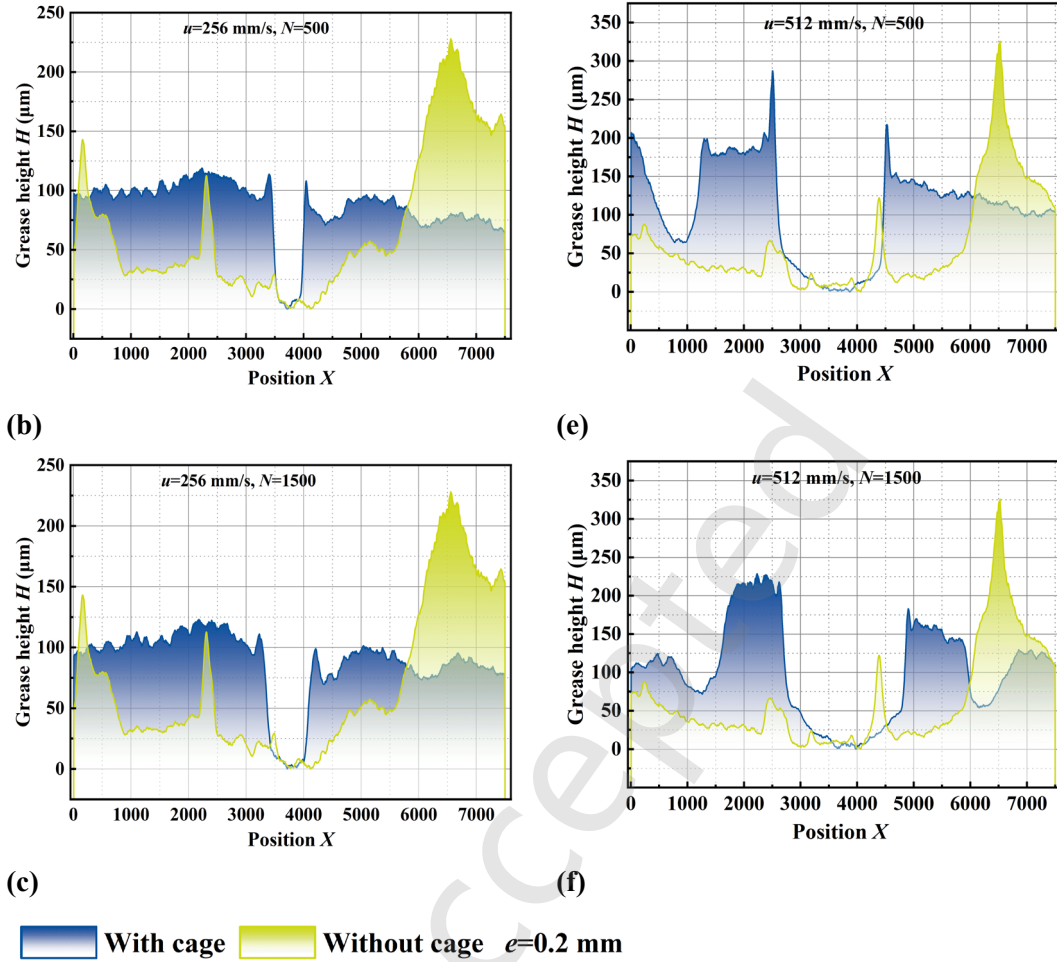


Fig. 10. Grease distribution on the steel ball surface across the inlet of the contact region for PAO-Li-2-50 without and with a cage of 0.2 mm clearance at the entrainment velocities (a-c) 256 mm/s and (d-f) 512 mm/s after (a, d) 200, (b, e) 500, and (c, f) 1500 revolutions.

The film thickness curves for PAO-Li-2-50 and PAO-Li-1-50 greases at entrainment speeds of 256 mm/s and 512 mm/s with different cage clearances are shown in Figure 11. For PAO-Li-2-50, at the lower speed of 256 mm/s, film thickness remained relatively stable at approximately 170 nm when the cage clearance was 0.1 mm. At the higher speed of 512 mm/s, a cage clearance of 0.2 mm maintained a longer operating period before severe starvation occurred, extending to approximately 1600 revolutions. For PAO-Li-1-50, at the lower speed 256 mm/s most cage clearances maintained effective lubrication except for the largest clearance with 0.5 mm. At the higher speed (512 mm/s), effective lubrication was maintained for all cage clearances and the highest film thickness occurred at for a clearance of 0.2 mm. To further quantify the lubrication conditions, the specific film thickness parameter λ was calculated using an estimated composite root-mean-square roughness of 18.20 nm. The corresponding evolution of λ is presented in Fig.

S6 in the Supplementary Material. These results suggest that suitable cage clearances enable stronger interaction between the cage and the thicker grease layers formed near the contact inlet, thereby promoting more effective grease redistribution. When the cage clearance becomes larger (e.g., 0.5 mm), the lateral ridges height formed during the initial stage was smaller than the clearance, as shown in Fig. A2 in the Appendix. As a result, the cage cannot effectively scrape or redistribute these grease accumulations. In addition, PAO-Li-1-50 was expected to exhibit a lower apparent structural strength than PAO-Li-2-50, consistent with its lower NLGI grade, allowing it to more easily overcome the surface tension between the grease and the ball surface. Consequently, with the same cage clearance, PAO-Li-1-50 could be redistributed more easily through the cage clearance, resulting in improved grease replenishment behavior.

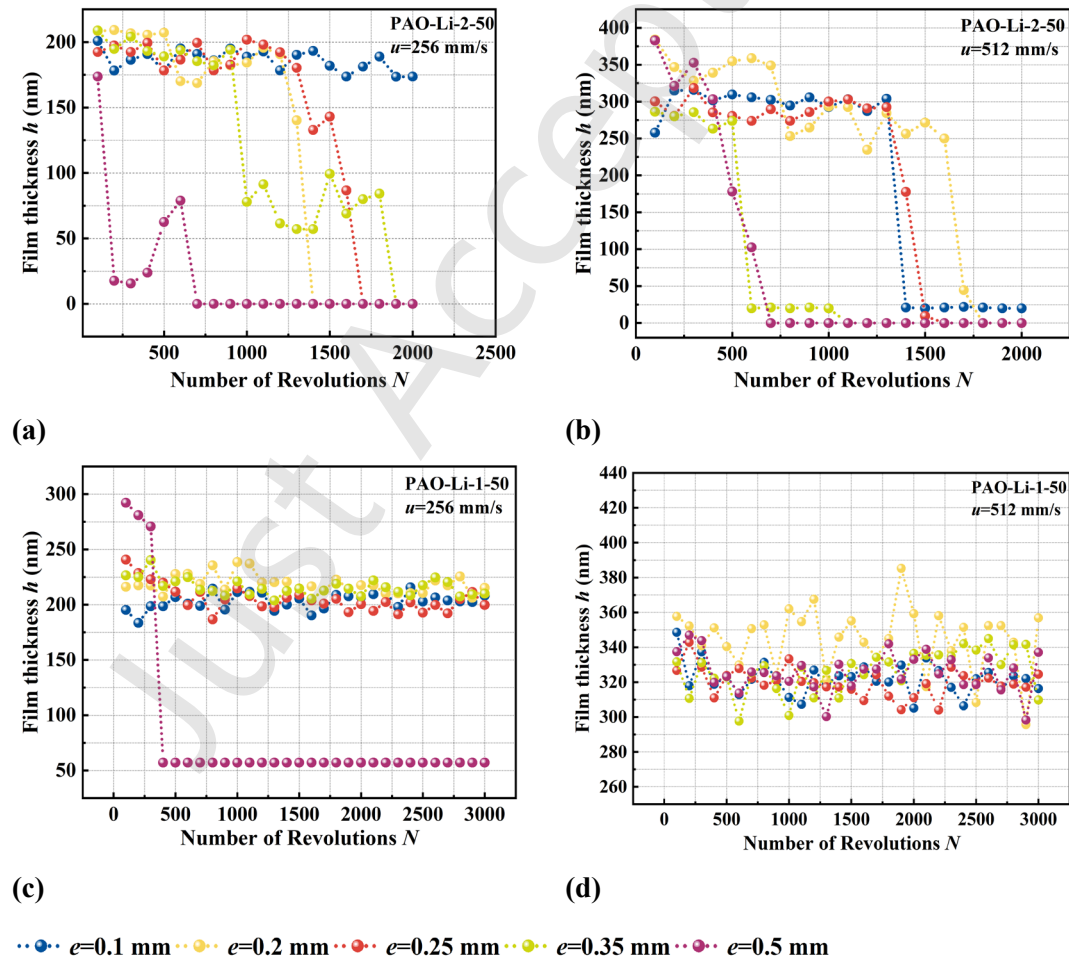


Fig.11. Central film thickness versus number of revolutions for (a, b) PAO-Li-2-50 and (c, d) PAO-Li-1-50 with different cage clearances at the entrainment velocities (a, c) 256 mm/s and (b, d) 512 mm/s.

These experimental results indicate that the spatial distribution of grease at the contact inlet and

the grease layer height on the ball surface near the contact zone govern the grease supply mode. These factors determine the size of the lubricant reservoir within the contact zone and the inlet length from the meniscus to the contact center, which ultimately affects the film thickness. Under grease lubrication conditions, film thickness reflects only the instantaneous lubrication state within the contact zone and cannot directly characterize grease replenishment at the inlet. However, film formation and stability are strongly influenced by the spatial distribution of grease in the inlet region. Therefore, the grease supply state was further evaluated using two parameters: the local average grease layer height H_a and the dimensionless inlet length m . As defined above, H_a represents the average grease layer height within the fixed 350 μm -wide analysis region adjacent to the contact zone, while m characterizes the streamwise extension of the grease-covered region into the effective entrainment zone. The resulting H_a - m map is therefore used to describe the grease supply state near the contact inlet. Based on this framework, the parameters H_a and m were measured for both greases during operation. The H_a - m relationship throughout the entire process from the initial stage of operation to the onset of severe starvation under different cage clearances and entrainment speeds is illustrated in the Figures 12 and 13. The distribution of the data points reflects the evolution of the lubrication state over the course of operation, each data point corresponded to the grease layer height and inlet length throughout the process from N values of 100, 200, 300, \ldots, up to the onset of severe starvation. During operation, the H_a - m distribution effectively forms a two-dimensional lubrication supply space. The parameter m represents the effective inlet length over which grease contributes to film formation. Larger values of m promote the formation of a stable lubricant film, whereas smaller values, especially those approaching unity, are indicative of starvation conditions.

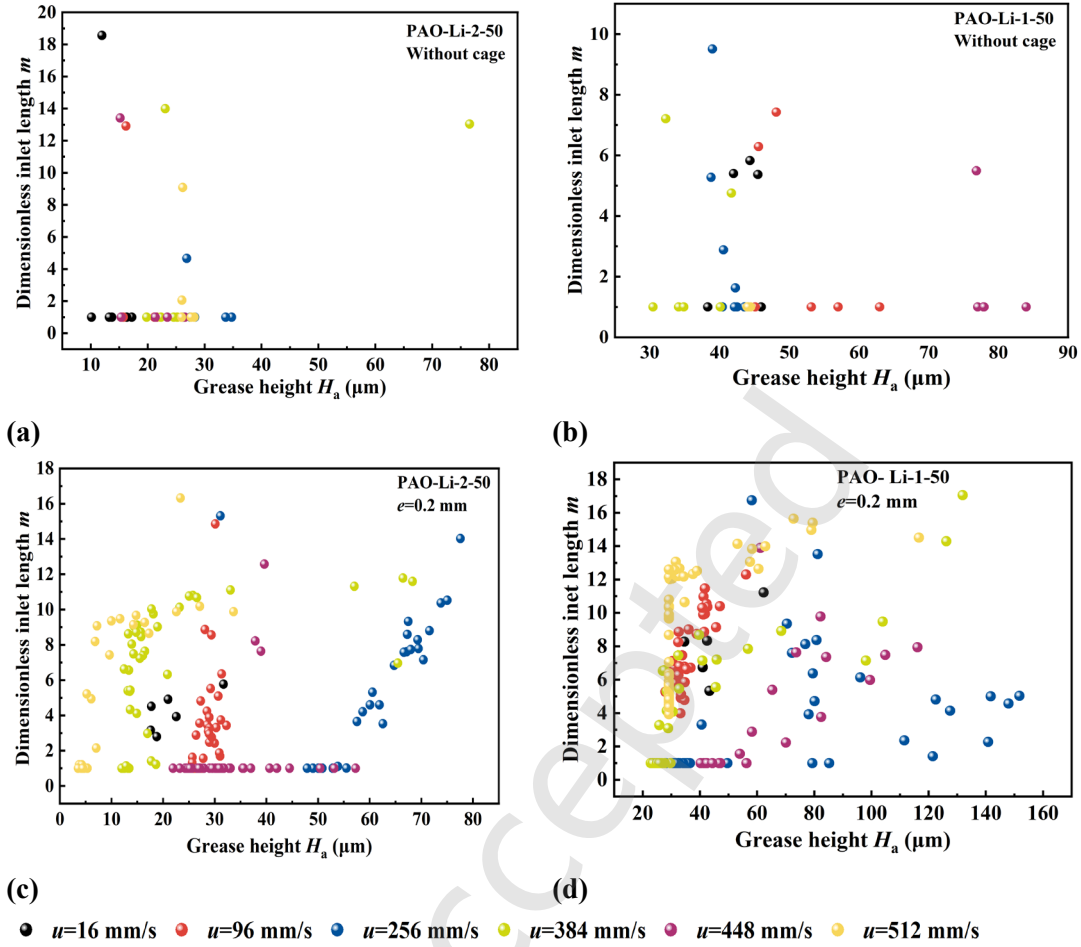


Fig. 12. Distribution of H_a - m (a, b) without and (c, d) with the cage with a clearance of 0.2 mm at different entrainment speeds for (a, c) PAO-Li-2-50 and (b, d) PAO-Li-1-50.

As shown in Fig. 12a, for the PAO-Li-2-50 grease, at the speed of 384 mm/s, under the no-cage condition, the grease layer height on the ball surface at the inlet initially reached approximately 80 μm . However, the effective inlet length was sustained only up to about 300 revolutions. As can be observed from the Fig. 12a, only two effective lubrication points (where $m > 1$ corresponds to a grease-covered inlet length exceeding one Hertzian contact semi-width) were present, and starvation had already occurred at $N=300$. In addition, only a limited number of effective lubrication points are observed at all entrainment speeds. Subsequently, most operating conditions entered a starvation regime, characterized by $m = 1$. A similar trend was observed in Fig. 12b for the cage with a clearance of 0.2 mm. In turn, the distributions in Fig. 12b and d for PAO-Li-1-50 showed a substantial increase in the number of effective lubrication points at different entrainment speeds. With the same cage clearance, the lower consistency allowed it to be more easily retained on the contact surfaces compared with PAO-Li-2-50. As a result, a larger number of effective lubrication points were observed. This is in reasonable

consistency with the film thickness results shown in Fig. 11c and d. In addition, the experimental results showed that, under the influence of the cage, the distribution of effective lubrication points in the H_a - m space tended to exhibit a more densely distributed pattern.

These results revealed a continuous relationship between the grease layer height H_a and the dimensionless inlet length m . In some cases, stable lubrication was maintained even when the grease layer height remained at relatively low or moderate levels, indicating that the grease was effectively entrained into the contact zone. However, situations also occurred in which a relatively large grease layer height did not correspond to an increase in inlet length. In certain operating conditions, the inlet length even decreased. This behavior suggests that grease replenishment at the inlet becomes limited and gradually evolves towards starvation. Therefore, the lubrication state cannot be described by a single parameter but is jointly governed by the grease layer height at the inlet and the effective inlet length, which together control grease supply and film formation within the contact zone. In addition, the data points corresponding to different H_a - m combinations exhibited noticeable dispersion. This reflects the sensitivity of grease distribution behavior to variations in cage clearance and local operating conditions. These observations indicate that the lubrication state should be interpreted as a continuously evolving process, rather than as a set of discrete lubrication regimes. Furthermore, the H_a - m - h distributions of effective lubrication points ($m > 1$) are summarized in Figure 13 for different cage clearances at the entrainment speeds of 256 mm/s and 512 mm/s. Based on the previously defined H_a - m framework, the measured central film thickness h is encoded by the color scale to provide a more direct visualization of the lubrication response. Therefore, the H_a - m - h map should be interpreted as a lubrication-supply state map with film thickness information, rather than as a direct predictive model of h . A high H_a does not necessarily lead to a high h if the grease is located outside the effective inlet region or remains as a weakly entrainable deposit. For cases with a relatively large number of effective lubrication points, curve fitting was further performed to enable a clearer evaluation and comparison of the lubrication performance under different cage clearances. The sensitivity analysis summarized in Table S2 quantifies the extent to which these fitted H_a - m relationships depend on representative systematic perturbations of H_a . The resulting ranges are interpreted as deterministic sensitivity ranges rather than statistical

confidence intervals. A comparison of the fitted results showed that, for the same H_a value, different cage clearances corresponded to noticeably different m values, indicating that the lubrication enhancement efficiency varied with cage clearance. Under the high speed condition (512 mm/s), for a cage clearance of 0.2 mm, the data for PAO-Li-2-50 and PAO-Li-1-50 were well described by an exponential saturation model and a quadratic model, respectively, yielding coefficients of determination of 81% and 78%, respectively. At a given residual grease layer height H_a , this clearance corresponded to larger inlet lengths m compared with other clearances, indicating a more stable grease replenishment behavior. Physically, as the residual grease layer height at the inlet increases, grease supply into the contact region is enhanced. However, as H_a increases within the investigated range, the system approaches a dynamic equilibrium, and the inlet length gradually reaches saturation. In the nominal fitted comparison within the low H_a region, the 0.2 mm cage clearance frequently exhibited relatively large m values, particularly under the high-speed condition. However, the magnitude and separation of these differences depend on the assumed H_a perturbation, as shown in Table S2. This showed good consistency with the film thickness results presented in Fig. 11. When the cage clearance was relatively small (0.1 mm), the H_a - m distribution was characterized by shorter inlet lengths. Although a relatively high initial film thickness was observed, grease replenishment became limited after a certain running period. This behavior can be attributed to the intensified scraping and blocking effects of the cage under small clearances, which hinder the continuous transport of grease toward the inlet region. For intermediate clearances (0.25 mm and 0.35 mm), the H_a - m distributions generally showed an initial increase followed by stabilization or a slight decrease, indicating the existence of an optimal grease supply level. This regime can be regarded as a transition region, where the sensitivity of grease replenishment behavior to cage clearance was reduced. Correspondingly, the stability of film thickness evolution also decreased. When the cage clearance became relatively large, i.e., 0.5 mm, the number of effective lubrication points decreased substantially. Although the grease layer height may increase, the weakened scraping and guiding effects of the cage promote local accumulation of grease rather than continuous replenishment at the inlet, thereby negatively affecting the formation and stability of the lubricant film. It should be noted that these trends are not universally applicable under all operating conditions. Under low-speed conditions, the sensitivity of different greases to cage

clearance varied, and intermediate clearances tended to provide improved lubrication performance. This indicates that the optimal cage clearance should be determined based on specific operating conditions, including entrainment speed and grease type.

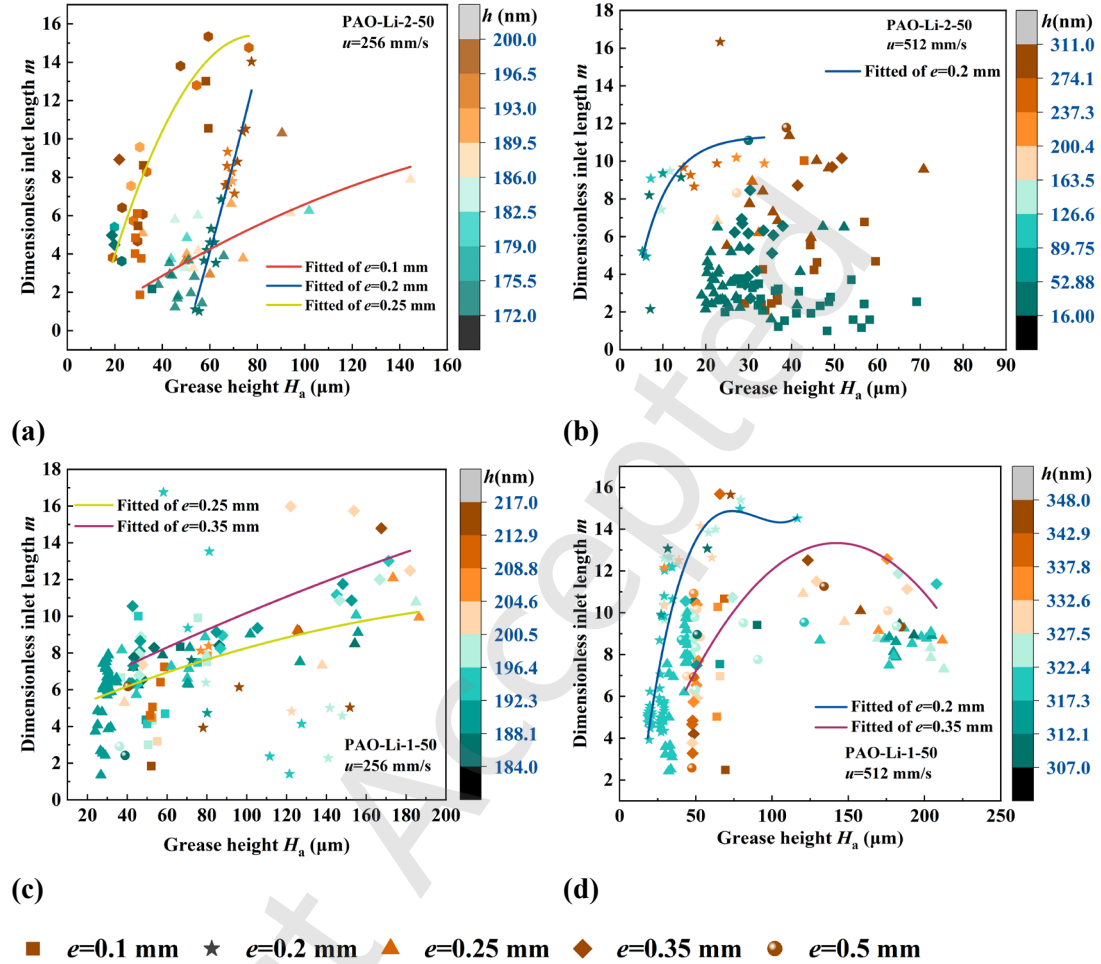


Fig.13. Distribution of H_a - m for (a, b) PAO-Li-2-50 and (c, d) PAO-Li-1-50 without cage and with cages of different clearance at the entrainment speeds (a, c) 256 mm/s and (b, d) 512 mm/s. The color scale indicates the measured central film thickness h , and different symbol shapes denote different cage clearances; separate color scales are used for each panel because the film thickness range varies among operating conditions. Only empirical trend lines with acceptable fitting quality are shown, and they are used only to indicate trends within the measured data range.

Finally, the dimensionless inlet lengths m predicted at the nominal reference heights of $H_a = 40$ μm and $H_a = 60$ μm under low- and high-speed conditions are compared for the two greases in Figure 14. A higher nominal m value indicates a more favorable cage clearance under the specified condition. The sensitivity analysis summarized in Table S2 shows that the magnitude

and separation of some between-condition differences depend on the assumed H_a perturbation. Therefore, $H_a=40$ and $60\text{ }\mu\text{m}$ are used as nominal reference heights for comparison rather than as precise or universal design thresholds. For example, in the nominal comparison at $H_a = 40\text{ }\mu\text{m}$ and $u = 256\text{ mm/s}$, the highest predicted m for PAO-Li-2-50 was obtained at a cage clearance of 0.25 mm . This result suggests that, under the investigated conditions, the cage action at this clearance may facilitate grease transport and thereby extend the effective operating duration. Overall, favorable lubrication performance requires an appropriate matching between grease layer height and cage clearance. A suitable cage clearance promotes an effective grease distribution pattern at the inlet, thereby facilitating stable film formation and improved lubrication performance.

The optical ball-on-disc configuration used in this study was designed to observe grease redistribution, inlet replenishment, cage scraping, and film formation under controlled conditions. Therefore, the H_a - m - h framework is more appropriately regarded as a diagnostic method for describing grease supply states, rather than as a direct tool for predicting quantitative results in real bearings. Compared with the ball-on-disc setup, real bearings involve multiple rolling elements, cage-pocket interactions, raceway geometrical constraints, and thermal effects, all of which can significantly affect grease transport, replenishment, and recirculation, making the lubrication mechanism more complex. Therefore, the H_a - m - h distributions obtained in this study and the evaluated optimal cage clearance should be interpreted as results specific to the present experimental configuration and operating conditions, rather than as direct quantitative predictions for real bearings. Nevertheless, this framework may still provide useful guidance for future bearing-oriented studies on grease supply state analysis and cage design.

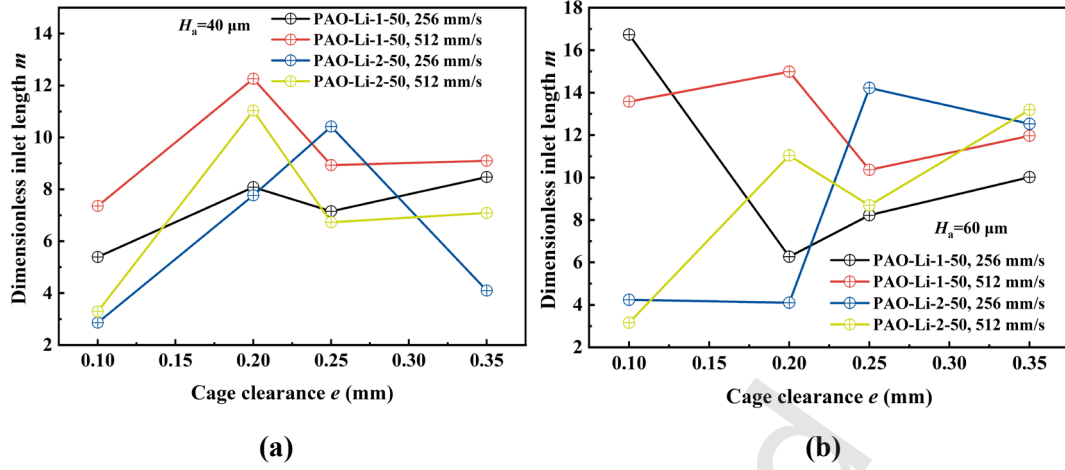


Fig. 14. Predicted dimensionless inlet length m at nominal reference heights of (a) $H_a = 40 \mu\text{m}$ and (b) $H_a = 60 \mu\text{m}$ in case of PAO-Li-2-50 and PAO-Li-1-50 at different entrainment speeds (The plotted values are fitted predictions, and the corresponding calibration-uncertainty sensitivity ranges are reported in Table S2).

4. Summary and Conclusions

In this study, the LED-IF method was employed to quantitatively characterize the grease layer height and meniscus at the contact inlet. Based on this approach, the starvation process in ball-on-disc experiments was directly visualized and its underlying mechanisms were observed and discussed. Furthermore, by combining the grease layer height at the inlet with the inlet length of the lubricant reservoir, a two-dimensional lubrication supply space ($H_a - m$) was constructed. Within this framework, the grease redistribution capability under different cage clearances was evaluated under various operating conditions. The main conclusions are summarized as follows:

- (1) The LED-IF technique enables direct characterization of the spatial distribution of grease on the steel ball surface at the contact inlet. The results show that stringing appears to be a dominant observed redistribution mechanism. As the operating time increases, grease gradually evolves into a fragmented distribution state, in which isolated grease fragments can no longer reform stringing structures, thereby limiting grease replenishment at the inlet and leading to starvation.
- (2) Film thickness is governed by the residual grease layer height and its spatial distribution on the ball surface. The grease layer height H_a was quantitatively measured using the LED-IF technique, and combined with the dimensionless inlet length m to construct $H_a - m$ distributions

under different operating conditions. These results show reasonable consistency with film thickness measurements. The framework may support future bearing-oriented lubrication assessments in cases where direct film-thickness measurement is difficult, although its quantitative transfer to full bearings requires validation under representative bearing kinematics and thermal conditions.

(3) Based on the H_a - m distributions of effective lubrication points, the scraping and redistribution effects of cages with different clearances were systematically evaluated under various operating conditions. The results indicate that effective lubrication performance depends on an appropriate matching between cage clearance and grease layer height, which is strongly influenced by grease type and entrainment speed. In general, excessively small clearances hinder grease transport, while excessively large clearances weaken the scraping and guiding effects of the cage, thereby reducing grease replenishment at the contact inlet.

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CRedit author statement

Y. Liu: Conceptualization, Methodology, Validation, Formal Analysis, Investigation, Data Curation, Writing – original draft, Visualization; **F. Pape:** Writing - Review & Editing, Supervision; **M. Liu:** Conceptualization, Methodology, Writing - Review & Editing; **G. Bayer:** Writing - Review & Editing; **V. Schneider:** Writing - Review & Editing, Supervision; **G. Poll:** Writing - Review & Editing, Supervision; **M. Marian:** Resources, Writing – original draft,

Writing - Review & Editing, Supervision, Funding acquisition.

Data availability statement

The data supporting the findings reported in this study are available from the corresponding authors upon reasonable request.

Conflict of Interest

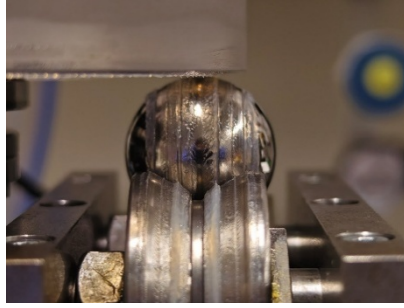
The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this manuscript related to the current study.

Ethical statement

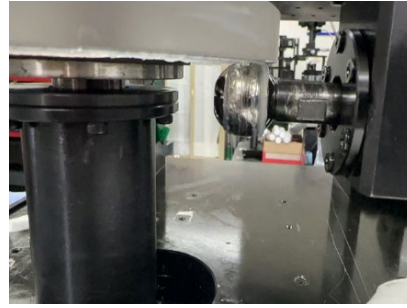
This study did not involve human participants or animals. All experiments were conducted in accordance with relevant institutional safety guidelines.

Appendix

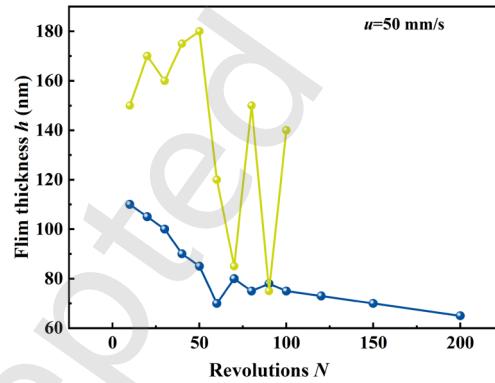
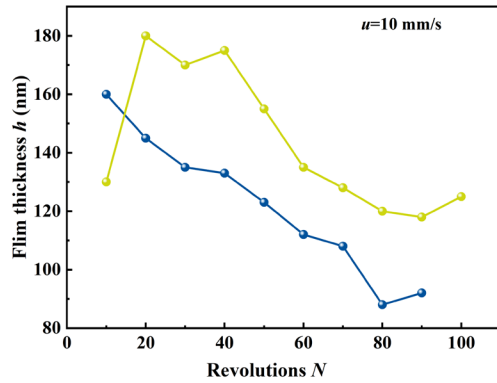
The push-back effect of the V-shaped holder in the ball-on-disc setup (Figure A1a) employed in this study was verified by comparison with a ball-and-disc dual-driven configuration in the absence of a holder (Figure A1b). A respective comparison of film thickness for the two configurations at different entrainment speeds is shown in Figure A1c-e). At low speeds (10 mm/s and 50 mm/s), the film thickness without the holder was slightly higher than that with the holder. However, as the speed increased, the backflow effect became more pronounced. At 200 mm/s, the presence of the holder promoted the push-back effect of grease during the stringing process, pushing more lubricant closer to the contact inlet. As a result, under identical operating conditions, a higher film thickness was achieved.



(a)

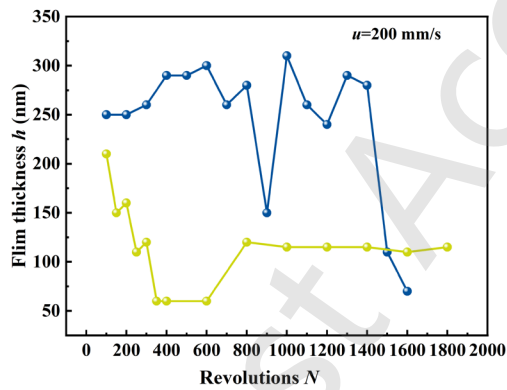


(b)



(c)

(d)



(e)

Fig. A1. Photographs of the test setup (a) with V-shaped holder and (b) with dual-drive configuration without holder as well as central film thickness versus number of revolutions in both cases for lubrication with PAO-Li-2-50 at an entrainment speed of (c) 10 mm/s, (d) 50 mm/s, and (e) 200 mm/s

In Figure A2, the evolution of the grease layer height on the steel ball surface as a function of the number of revolutions for PAO-Li-2-50 grease under the no-cage condition and with a cage clearance of $e=0.5$ mm at speeds of 256 mm/s and 512 mm/s was compared. As shown in the figure, under the action of the cage with a clearance of $e=0.5$ mm, the grease layer height was mainly concentrated around 75 μm , which was significantly smaller than the cage clearance itself. This indicates that the scraping effect of the cage was largely diminished under this

condition. Consequently, the excessively large cage clearance weakened the lubrication enhancement effect provided by the cage.

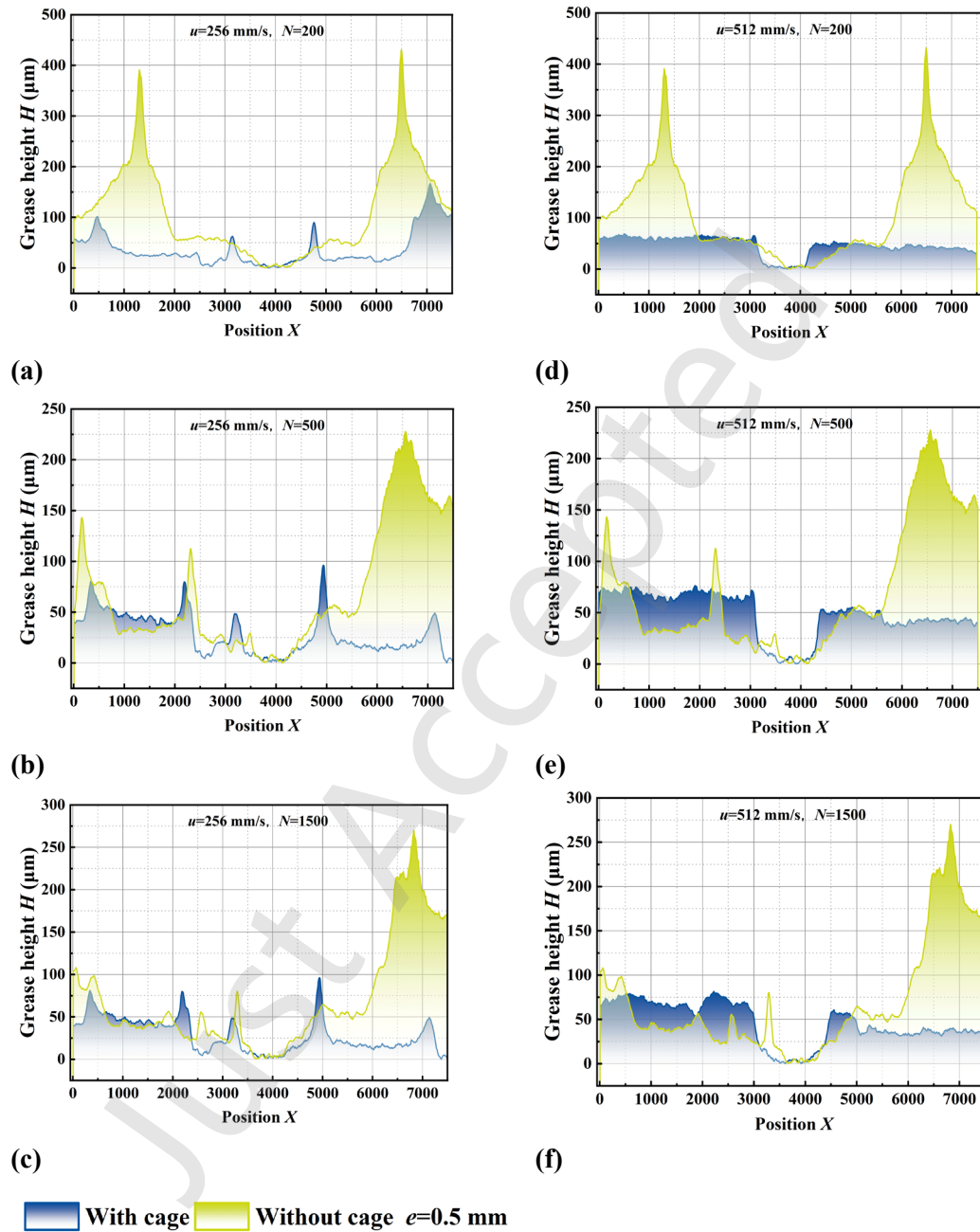


Fig. A2. Grease distribution on the steel ball surface across the inlet of the contact region for PAO-Li-2-50 without and with a cage of 0.5 mm clearance at the entrainment velocities (a-c) 256 mm/s and (d-f) 512 mm/s after (a, d) 200, (b, e) 500, and (c, f) 1500 revolutions.

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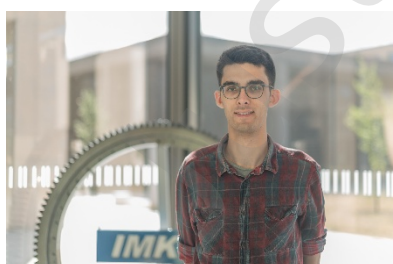
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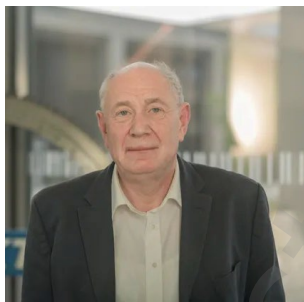
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