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RESEARCH ARTICLE

Friction behavior and wear protection ability of selected base lubricants

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Abstract: This paper reports a comparative study on the tribological characteristics of base oils in four groups, synthetic esters, mineral oils, polymerized alpha olefins (PAOs), and poly alkylene glycols (PAGs), by means of viscosity, friction, and wear measurements. Friction coefficients for the lubricants in each group, measured with a pin-on-disk tribometer, are summarized in the form of Stribeck curves. Wear of the disk specimens due to rubbing in the interfaces lubricated with the tested oils is evaluated by surface topographic changes. The results indicate that for surfaces of similar roughness, viscosity has the predominant influences on the friction and wear-protection properties of these lubricants.

Keywords: friction; lubricants; viscosity; Stribeck curves; wear loss

1 Introduction

Lubricants with a strong film forming ability can effectively separate contact surfaces, leading to low friction in the boundary and mixed lubrication regimes and low wear of interacting surfaces. Therefore, measurements of friction and wear are essential to understand the effectiveness of a lubricant. Masjuki and Maleque [1] proved the effectiveness of boundary lubrication due to the presence of a polar structure of a fatty acid of palm oil methyl ester composition through anti-wear and friction coefficient reduction ability of the lubricant even at high load. Masjuki et al. [2] carried out a comparative study of wear, friction, viscosity, and lubricant degradation etc., to demonstrate the performance of the mineral oil based lubricating oil. Gryglewicza et al. [3] proved, with a four-ball test apparatus, the lubrication improvement due to the addition of esters of dicarboxylic acids through the formation of a hydrodynamic film at high temperatures and high shear velocity. Martins et al. [4] compared the influence of biodegradable ester and mineral oil in terms of friction coefficient between the gear teeth and the mass loss. Lee et al. [5] evaluated the friction forces and wear amounts in boundary lubricated sliding condition using a pin-on-disk tester to evaluate a PAG Oil. Mia et al. [6] tested the tribological properties (low-temperature behavior, bulk property, frictional coefficient and wear behavior) of PAO oils and a mineral oil to prove that PAO oils exhibited better frictional property and low-temperature behavior. Qian et al. [7] experimentally investigated lubrication failure of polyalphaolefin (PAO) oil film by measuring the traction coefficient and surface profiles of the tribopairs based on a ball-on-disc configuration.

The friction variation with operating conditions at interfaces lubricated with different lubricants may be compared in terms of friction by means of the Stribeck curves, which are for friction as a function of the Hersey number, $\eta N/P$ composed of viscosity η, speed N, and average pressure P [8]. The Stribeck curve concept [9] has been accepted as an overall observation of friction variation in the entire range of lubrication, including

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the hydrodynamic, mixed and boundary lubrication regimes [10−13]. Many researchers used this concept as the basis to analyze the influences of lubricant viscosity related to both base oil and additives and operating conditions on tribological properties of lubricants and contact interfaces. Lu and Khonsari [14] explored the effects of load, oil type, size, depth, and shape of dimples in a journal bearing bushing on the friction characteristics by changing these parameters and plotting Stribeck curves. Moshkovich et al. [15] investigated the influences of sliding velocity and load on the friction coefficient and wear of copper specimens, in which elastohydrodynamic, mix and boundary lubrication regions were presented in the form of the Stribeck curve. Zhu and Wang [16] analyzed the lambda ratio, defined as average film thickness divided by composite root mean square (RMS) roughness, in the mixed lubrication regime. Zhu et al. [17] plotted the Stribeck curves in a wide range of speed and lubricant film thickness based on simulation results that considered various types of contact geometry and machined rough surfaces of different orientations; they also analyzed the relationships between friction and film thickness in a wide range of speed. In addition, wear as a result of rubbing due to the relative motion of surfaces in contact is another element for evaluating the tribological properties of lubricants. Siniawski et al. [18] reported a study on the initiation and development of the surface wear of 52100 steel balls run against B_4C-coated disks, using a pin-on-disk tester, and extended their analysis to wear data obtained from surfaces that experienced more wear cycles. Siniawski et al. [19] discovered that the abrasion rate at any point of any material pair can be predicted accurately with a wear equation, in which the only parameters required for any material pair are the initial abrasiveness and the initial rate at which the abrasiveness decreases with cycles. Obviously, wear in the first cycle can best show the rubbing between the two materials because the interference of wear debris is largely excluded. Such a consideration can be introduced to lubricant studies because it also reveals the surface protection ability of a lubricant without the complication of wear debris.

Developing more energy-effective lubricant formulations requires the knowledge of base fluids. The work reported in this paper explores the tribological properties of a number of lubricants in terms of viscosity, friction coefficients, and the wear losses of the disk surfaces lubricated by the lubricants. Apparently, the Stribeck curve concept can be utilized in this study to summarize the friction behaviors of the targeted fluids, and wear of the fresh disk surface should offer an impartial view of the protective effect of these base fluids.

2 Lubricants studied and the experiments

2.1 Lubricants

This research studies eighteen base oils from four groups, including nine synthetic esters, three mineral oils, three poly alpha olefins (PAOs), and three poly alkylene glycols (PAGs)). The kinematic viscosities of these lubricants measured at 40 ℃ and 100 ℃ with the SYD-265D-1 kinematic viscometer are listed in Table 1.

PAGs are mixtures of $[O(CH_2)_aO_c(CH)_b]_n$, which are polymerized by $HO(CH2)_aO_c(CH_2)_bOH$. Mineral oils are mixtures of high molecular-weight hydrocarbons and non-hydrocarbons. Such an oil with a higher

Table 1 Kinematic viscosity of lubricants (V40, V100 are the kinematic viscosities measured at 40 ℃ and 100 ℃).

Lubricants	V40 (mm^2/s)	V100 (mm^2/s)
PAG a	4.21	1.42
PAG b	143.16	25.43
PAG c	1630.07	284.60
Mineral oil 1	27.25	5.11
Mineral oil 2	85.95	9.54
Mineral oil 3	132.08	9.43
PAO a	28.11	5.67
PAO b	209.62	20.16
PAO c	598.40	63.80
Ester 1	22.00	4.78
Ester 2	38.75	7.03
Ester 3	28.59	5.85
Ester 4	10.95	3.15
Ester 5	35.99	5.12
Ester 6	69.91	7.43
Ester 7	14.74	3.54
Ester 8	46.06	9.41
Ester 9	75.66	9.75

viscosity is composed of higher molecular weight hydrocarbons than that with a lower viscosity. PAOs are composed of linked alkyl structures prepared by oligomerization (joining) of olefins having a carbon–carbon double bond between the first and second atoms; these molecules have the general formula of C_nH_{2n} [20]. They have relatively lower side branching, comparatively shorter molecular chain lengths, and greater consistency in performance than most mineral oils [20, 21]. Among the ester lubricants, Ester 1 is a pentaerythritol ester whose general structure is shown in Fig. 1(a), Ester 2 is a dipentaerythritol ester (general structure in Fig. 1(b)), Esters 3 and 4 are linear decyl esters with the general structure shown in Fig. 1(c), Esters 5 and 6 are orthophthalics with the general structure shown in Fig. 1(d), Esters 7 and 8 are trihydroxy esters with the general structure is shown in Fig. 1(e), and Ester 9 is a 1,3-benzene (Fig. 1(f)).

2.2 Experimental apparatus and pin-disk specimens

A RTEC multifunctional friction tribotester (MFT) was used in this study. The pin-on-disk rotation module was employed to conduct the tests and the in-line white-light interferometry to scan the surface morphology of specimens in the contact area.

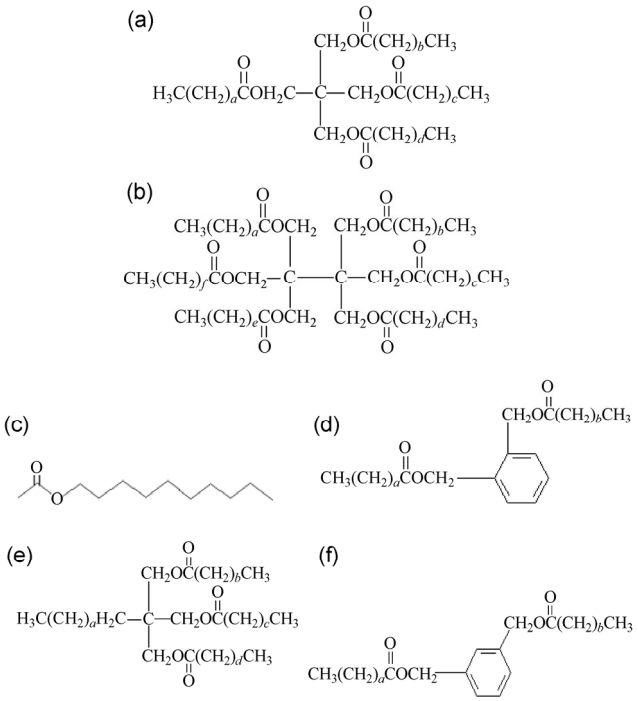

Fig. 1 General molecular structures for the esters. Variables *a*, *b*, *c* in each general molecular structure can be the same or different. (a) Pentaerythritol esters (Ester 1). (b) Dipentaerythritol esters (Ester 2). (c) Decyl esters (Esters 3 and 4). (d) Orthophthalics (Esters 5 and 6). (e) Trihydroxy esters (Esters 7 and 8). (f) 1, 3-benzenes (Ester 9).

Steel disk specimens are 75 mm in diameter and 6.50 mm in thickness, made of 304 stainless steel. The surface hardness is HV189.0. The ball specimens are 9.5 mm in diameter, made of 440C stainless steel. The centerline average roughness, Ra, of the disk specimen surfaces is about 0.8 μm, and such roughness should result in a certain amount of asperity rubbing at low speed conditions. The ball surfaces were processed by superfinishing, and the surface roughness could be neglected, compared with disk surface roughness. The hardness of the steel balls is about HRC62 and the sphericity is about 0.000024 mm. The compositions of the disk and ball materials are listed in Tables 2 and 3. The intention of using disks of the same rougher surfaces is to allow the comparison of the lubricant performance throughout a wide range of lubrication status but similar levels of roughness influence so as to reveal the influences of the targeted lubricants.

2.3 Experimental procedures

Two types of tests, circular-tracks and spiral-tracks, were conducted in the experiment. The circular-track tests are designed for the evaluation of continuous friction change, to support the datum summary in the form of Stribeck curves. In each circular-track test, a 304 stainless steel ball was pressed on a disk surface under a 5 N normal load (the Hertzian pressure on the virgin surface is about 504 MPa), and all comparisons are based on this initial condition. For each lubricant, 60 groups of tests were conducted with the velocity changing from 0.01 m/s to 0.6 m/s. Higher velocities were avoided in order to exclude the thermal effect. All experiments were repeated three times, and the average friction coefficients were calculated. A new ball-disk set was used for groups of the tests along the tracks of different radii. An ultrasonic cleaner was used to clean the specimen surfaces with isopropanol before and after the test, and each cleaning took about 15 minutes.

The surface wear pattern formed during the first cycle can have a profound influence on wear, where the effect of wear debris can be ignored [18, 19]. Comparison of wear of fresh surfaces allows a direct

Table 2 Chemical constitution of the disk material (%).

Material	C	Si	Mn	Cr	Ni	P	S
0Cr18Ni9 (stainless steel 304)	⩽ 0.07	⩽ 1.0	⩽ 2.0	⩽17.0–19.0	⩽ 8.0–11.0	⩽ 0.035	⩽ 0.03

Table 3 Chemical constitution of the steel ball material (%).

Material	C	Si	Mn	Cr	Mo	P	S
440c stainless steel	0.95−1.20	1.00	⩽ 1.00	16.0−18.0	⩽ 0.075	0.04	⩽ 0.03

evaluation of the surface protective abilities of the lubricants under investigation without the uncertainty due to debris contamination. The spiral-track tests were so designed that the ball travels along a spiral path over the fresh disk surface, resembling the wear tests on the first track that exclude the influence of wear debris. Thus the disk surface material loss from each test is essentially from asperity rubbing under the influence of a particular lubricant used. The spiral-track wear tests were conducted to evaluate wear of the disks lubricated with different fluids. The materials and sizes of the ball and disk specimens are the same as those used in the circular-track experiments. The normal load is 15 N in these spiral-track tests.

2.4 Surface morphology measurements for roughness change and surface wear

A representative area, approximately 0.9 mm^2 (0.9 mm × 1 mm), of the stainless steel disk specimen surface was selected as the observation region for each test to analyze the influence of lubricants on wear. The white light interferometer (WLI) was utilized to quantify the disk surface topography. Three dimensional (3D) digital surface topography maps were obtained by scanning the worn zone of 1.171 mm × 0.937 mm into 640 by 512 pixels, up to 327,680 data points. Centerline average roughness (Ra) of each surface was analyzed.

3 Results and discussions

3.1 Stribeck curves.

Friction coefficients measured at the pin-disk interfaces lubricated with the fluids listed in Table 1 at different sliding velocity under the same normal load are shown in Fig. 2. The friction coefficients generally decrease with the rotational speed at the lower speed side, but they increase with the rotational speed at the higher speed side. Each curve has a trough region where the friction coefficient reaches the minimum, and in the trough region, friction coefficients do not vary much with speed.

The central film thicknesses can be calculated with the Hamrock-Dowson film-thickness formula of point contacts (Eq. (1)) [22]:

$$H_c = 2.69U^{0.67}G^{0.53}W^{-0.067}(1-0.61e^{-0.73k}) \quad (1)$$

where

$$U = \frac{\mu_0 u}{E' R_x}$$

$$\frac{1}{E'} = \frac{1-v_1^2}{E_1} + \frac{1-v_2^2}{E_2}$$

$$G = \alpha E'$$

$$W = \frac{F}{E' R_x^2}$$

$$k = a/b$$

$$u = \frac{1}{2}(u_1 + u_2)$$

in which μ_0 is the absolute viscosity at the test temperature; u_1, u_2 are the velocity of surfaces 1 and 2, u is the average sliding velocity; E_1, E_2 and v_1, v_2 are elastic moduli and Poisson's ratio for body 1 and 2; R_x is the equivalent curvature radius in the direction of motion (x); α is the pressure-viscosity coefficient of lubricant; a is the contact ellipse dimension perpendicular to the direction of motion, which is the contact area radius in the current cases; b is the contact ellipse dimension parallel to the direction of motion, which equals a for the current cases, and k is ellipticity ratio, which is one for the current cases. The pressure viscosity index, α, is about 2.1×10^{-8} GPa [23].

The λ ratio, defined as average film thickness divided by composite root mean square roughness,

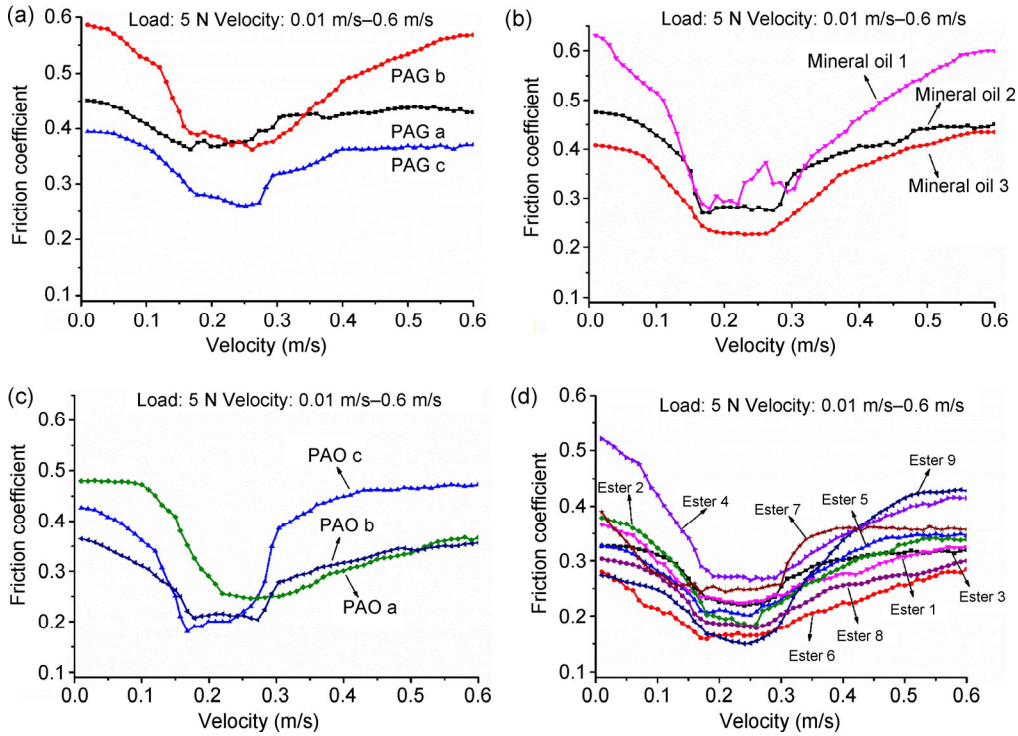

Fig. 2 Friction coefficient (the max error is 0.028) versus sliding speed under the same normal load, 5 N, for different types of the lubricants. (a) PAGs. (b) Mineral oils. (c) PAOs. (d) Esters.

in the range of 0.6–1.0 can be considered as the upper limit of mixed lubrication, and λ = 0.01–0.05 can be viewed as the lower limit of mixed lubrication according to Zhu and Wang [16]. Therefore, the film thickness of the mineral oils in the mixed lubrication regime is approximately in the range 0.008–0.04 μm < H_c < 0.48–0.8 μm, considering the fact that the disk average roughness is about 0.8 μm.

The central film thicknesses for the test pairs lubricated with the mineral oils were calculated at the speed of 0.15 m/s and 0.30 m/s, and the results are listed in Table 4, where H_{c1} is the central film thickness when the lubricating state shifts into the mixed lubrication from the boundary lubrication (when the speed is about 0.15 m/s), and H_{c2} is the central film thickness when the lubricating state shifts into the hydrodynamic lubrication from the mixed lubrication (when the speed is about 0.30 m/s).

These film values can be used as reference film thicknesses for most of the PAG and PAO lubricants, suggesting similar ranges of mixed lubrication in their corresponding Stribeck curves. The minimum friction coefficients of all lubricants appear in the speed region of 0.2–0.3 m/s confirms such agreement, which also suggest that they are in the transition to the full-film lubrication.

Table 4 Central film thicknesses of mineral oils for speed = 0.15 m/s and 0.30 m/s at 40 ˚C.

Lubricants	H_{c1} (μm)	H_{c2} (μm)
Mineral oil 1	0.01	0.46
Mineral oil 2	0.02	0.50
Mineral oil 3	0.03	0.64

Figure 3 presents the minimum friction coefficients in the trough regions together with the viscosities at 40 ˚C and the original Ra values of the disk specimen lubricated with different lubricants; where the minimum friction coefficients with the corresponding viscosity can be inversely correlated.

Different from the Stribeck curves obtained through journal-bearing sliding friction experiments [10, 23–27], the Hersey number for the friction coefficients from the pin-on-disk point-contact tests in the current work may be adjusted to $\eta V/(P{\cdot}a)$, where P is the average contact pressure defined in Eq. (2) [28, 29]:

$$P = 4F/(\pi a^2) \quad (2)$$

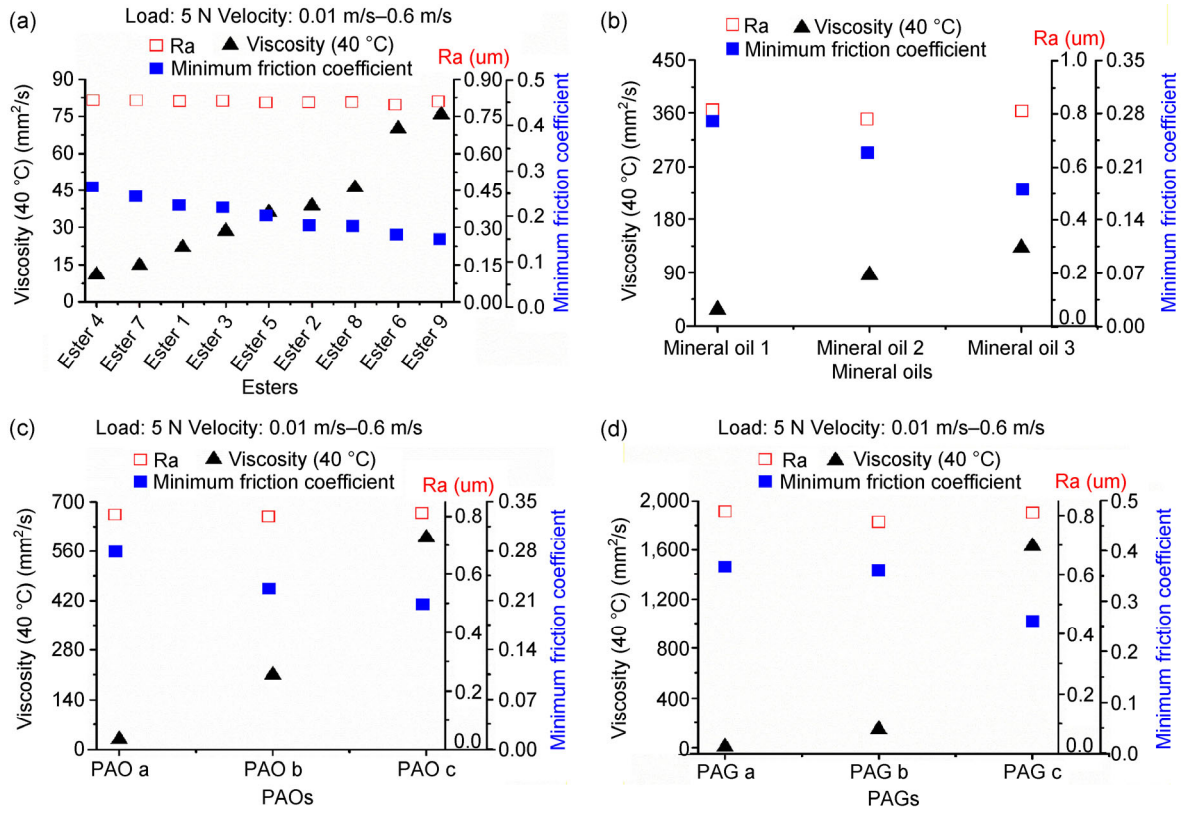

Fig. 3 Viscosity (40 ˚C) and minimum friction coefficient for different lubricants. (a) Esters. (b) Mineral oils. (c) PAOs. (d) PAGs. The original disk specimen surface roughness Ra is also shown as a reference.

in which F is the normal load and a is the Hertz contact diameter which is given in Eq. (3) expressed by ball radius R and material equivalent modulus E':

$$a = 2\left(\frac{2}{3}\cdot\frac{FR}{E'}\right)^{\frac{1}{3}} \tag{3}$$

Figure 4 plots the Stribeck curves of the four group lubricants against the Hersey number that combines the effects of viscosity, load and speed. Again, the initial pressure and contact area are used for these plots. They look different from the plots in Fig. 1 due to the inclusion of viscosity. The trough locations are altered; however, the appearances of the minimum friction coefficient are not changed.

It is interesting to view the Stribeck curves (Figs. 4(c) and 4(d)) for the ester lubricants together with their general molecular structures (Fig. 1). For the esters with a branched structure, the minimum friction coefficient for Ester 2 (dipentaerythritol esters) is lower than that for Ester 1 (pentaerythritol esters), which reveals that increasing the number of branches or the number of the carbon atoms may benefit the low-friction performance of these ester lubricants in the mixed lubrication regime. For Esters 3 and 4, both are decyl esters with a linear chain structure, but the former has more carbon atom than the latter, and the minimum friction coefficient for Ester 3 is lower than that for Ester 4; For the esters with a phenyl group, the minimum friction coefficients for Esters 5 and 6 (orthophthalics) are higher than that for Ester 9 (1,3-benzenes), indicating that the low-friction performance of 1,3-benzenes outperforms the orthophthalics ester in the mixed lubricant regime. Furthermore, comparing the minimum friction coefficients for the ester lubricants with the chain structures (Ester 2) with those for the esters with a phenyl group (Ester 5) suggests that the former is better for lower friction than the latter in the mixed lubrication regime when their viscosities are similar.

3.2 Surface wear

The surface topographic data were obtained through

scanning the disk surfaces with the white-light interferometer. Plots (a) and (b) in Fig. 5 show a typical disk surface topography digitized before and after a spiral-track test, clearly showing the grinding traces in the former and a worn track in the latter. In such an experiment, a mark was made on each disk surface to ensure the surface comparison at the same location. The morphologic changes of the worn disk specimen surfaces lubricated with different PAG lubricants (Figs. 6(a)–6(c)) can be captured from the width and depth of the wear tracks. The wear groove for PAG a is wider and deeper than that for PAG c. The wear loss of the disk surface lubricated with PAG a is larger than that with PAG b, and the wear loss associated with PAG c is smaller than that with PAG b. Similar topographic analyses were also conducted for the worn disk surface lubricated with mineral oils and ester lubricants for wear comparisons.

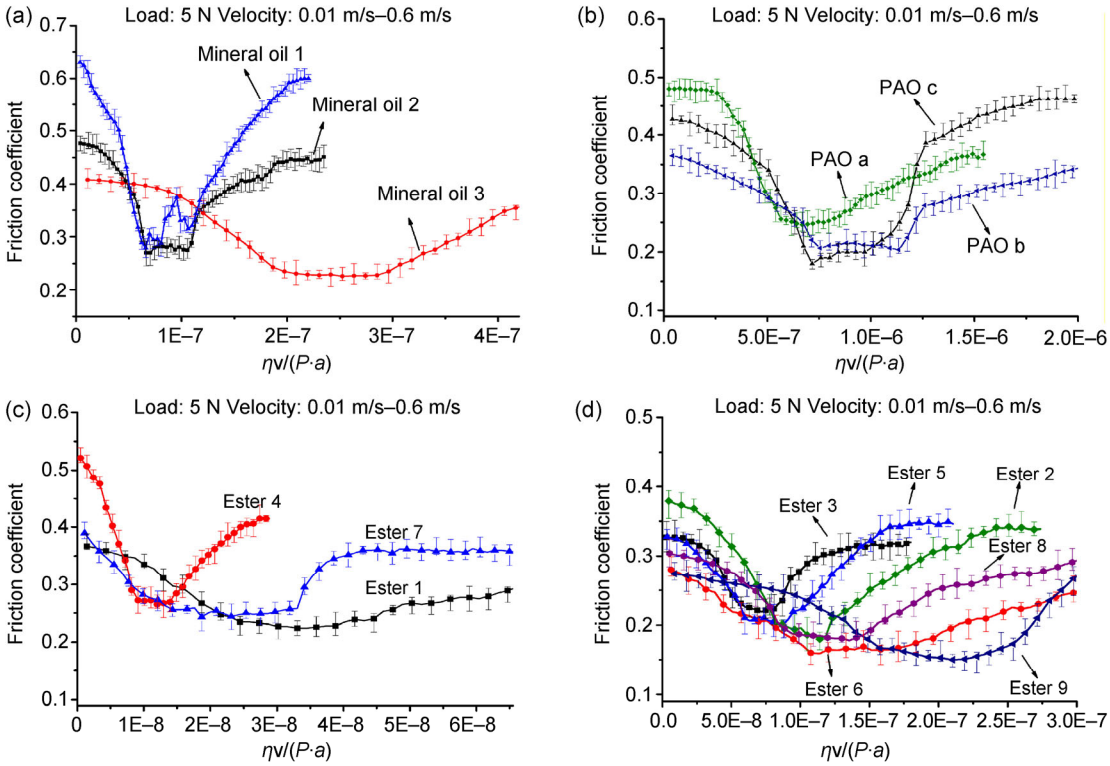

Fig. 4 Stribeck curves for different lubricants for friction coefficients vs. the Hersey number. (a) Mineral oils. (b) PAOs. (c) and (d) Esters.

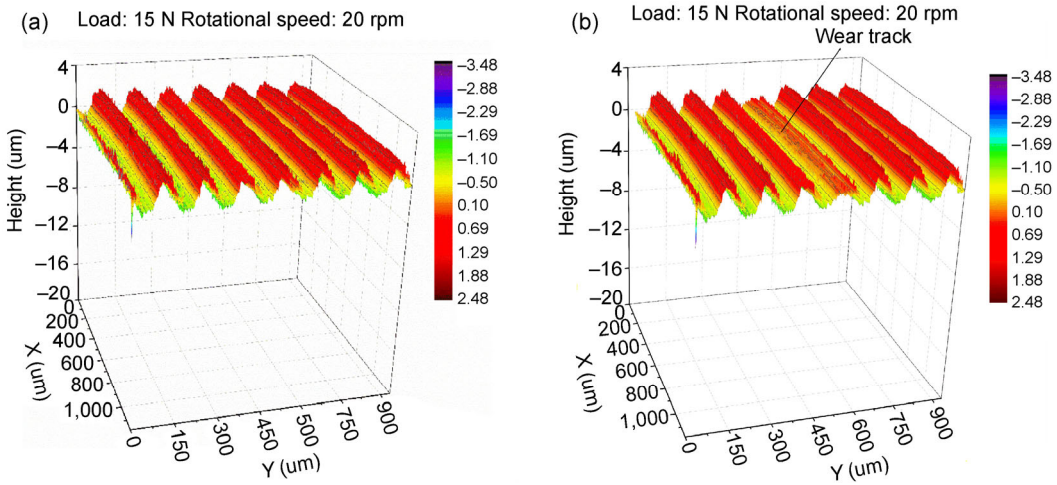

Fig. 5 Typical 3D topographic data for the disk surface before (a) and after (b) the spiral-track test using Ester 1 as the lubricant.

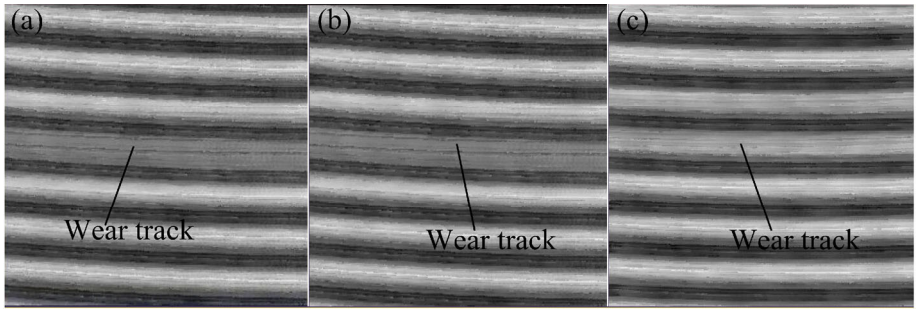

Fig. 6 Top topographic view, by means of the optical profilometer, of the worn disk specimen surfaces lubricated with different PAG lubrications at the conditions of load = 15 N and rotational speed = 20 rpm. (a) PAG a; (b) PAG b; (c) PAG c.

Wear is an important aspect to evaluate the effectiveness of a lubricant. Wear can be quantified with the changes in volume [30]. In this study, the wear volume of the disk specimen were calculated through the area difference of the cross sections before and after the wear test (Fig. 7). To do so properly, the profiles at the marked locations were monitored, which was the depth cross-section contour at x = 585 μm of each 3D disk surface topography, as shown in Fig. 7. The relationship between disk surface wear loss and viscosity of the corresponding PAGs are shown in Fig. 8, in which disk surface wear, viscosity of each PAG lubricant, and the average roughnesses of disk specimen before and after the spiral-track tests, Ra_1 and Ra_2, are plotted. This figure reveals that Ra_1 is always greater than Ra_2 in this group of experiments, suggesting a running-in process, as expected for the first-cycle wear of the fresh surfaces. Because the average roughness of every disk surface used in this study is about 0.8 μm, the influence of disk surface roughness on wear loss can be ignored.

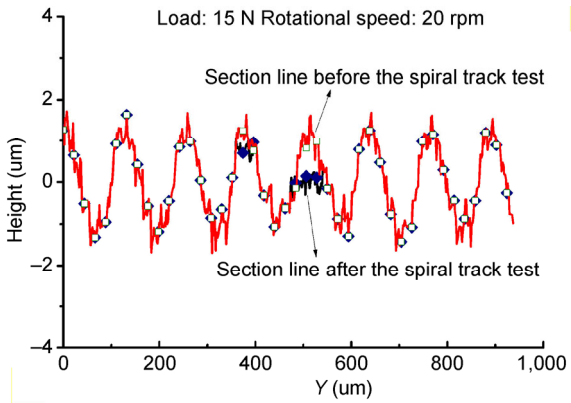

Fig. 7 Typical section profiles along the disk radial direction before, labeled with □, and after, labeled with ◆, the spiral-track test using Ester 1 as the lubricant.

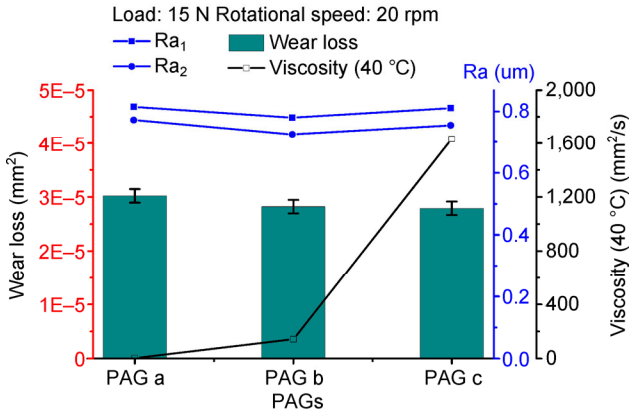

Fig. 8 Average roughnesses and wear losses of the disk specimens tested with different PAG lubricants. Ra_1 and Ra_2 are the average roughnesses of the disk specimens before and after the spiral-track tests, respectively.

Obviously, the disk surface wear losses subjected to different PAG lubricants decrease with the increase in viscosity when tested under the same experimental conditions.

Similarly, for the ester, mineral oil, and PAO lubricants, disk surface wear loss, viscosity of each lubricant, the average roughness of the disk specimen before and after the spiral-track tests with this lubricant are shown respectively in Figs. 9, 10 and 11, where similar conclusions can be obtained.

The authors have noticed some inconsistence between the trends of viscosity and wear when cross comparing different types of oils, which deserves a special attention. A comparative plot about the minimum friction coefficients and wear losses of the disk specimens lubricated with mineral oil 1 (viscosity: 27.25 mm^2/s), PAO a (28.11 mm^2/s), and Ester 3 (28.59 mm^2/s), which process similar viscosities, are shown in Fig. 12, revealing that the wear loss from

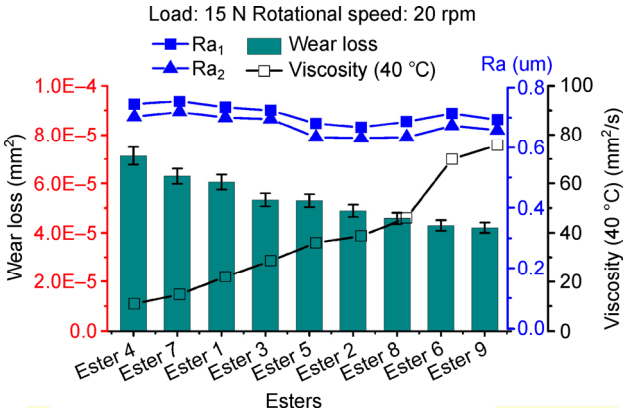

Fig. 9 Average roughnesses and wear losses of disk surfaces lubricated with different ester lubricants. Ra_1 and Ra_2 are the average disk roughnesses before and after the spiral-track tests, respectively.

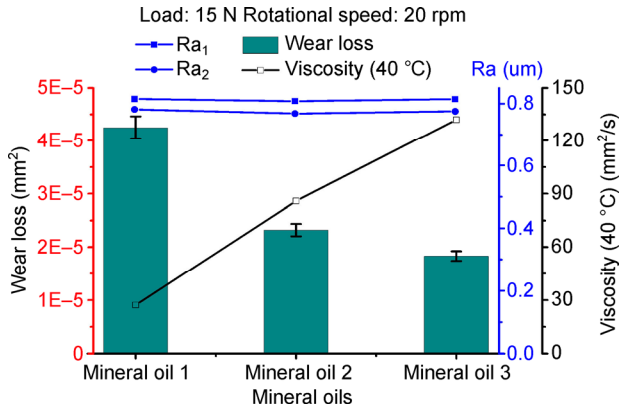

Fig. 10 Average roughnesses and wear losses of disk surfaces lubricated with different mineral oils. Ra_1 and Ra_2 are the average disk roughnesses before and after the spiral-track tests, respectively.

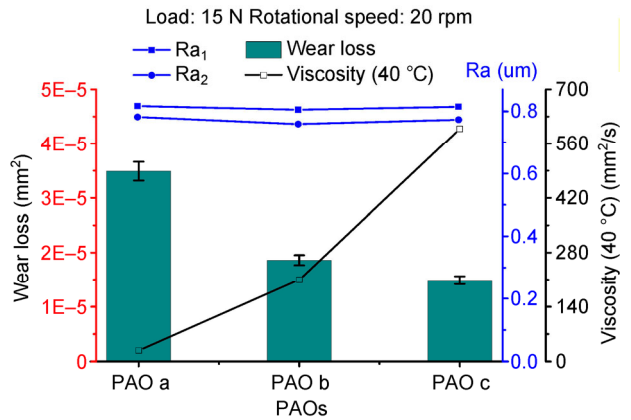

Fig. 11 Average roughnesses and wear losses of disk surfaces lubricated with different PAOs. Ra_1 and Ra_2 are the average disk roughnesses before and after the spiral-track tests, respectively.

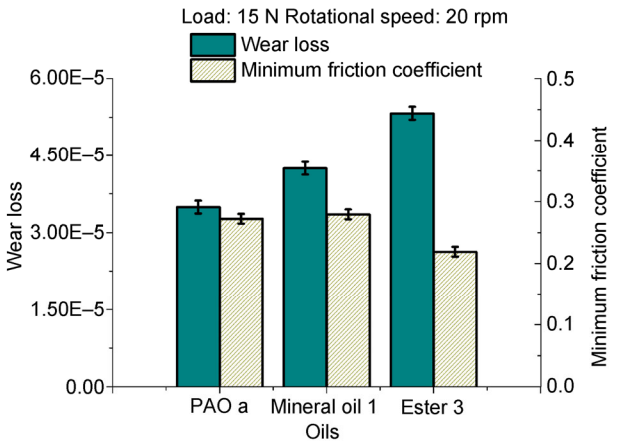

Fig. 12 Minimum friction coefficients and wear losses of the disk specimens lubricated with mineral oil 1, PAO a and Ester 3 which process similar viscosity.

the disk specimen lubricated with PAO a is the lowest among the three cases although the lowest minimum friction coefficient is associated with ester 3. Ester 3 is a linear decyl esters with the chemical structure of lone pair electrons on the oxygen atom of the eater group. Such polar molecules can effectively work in boundary lubrication for the reason that they tend to form stable physical bonds with metal surfaces [31]. However, the reason why a higher wear was found on the disks lubricated with Ester 3 should be further explored in a future study.

4 Conclusions

Tribological performances of eighteen lubricants (PAGs/Mineral oils/PAOs/Esters) were studied using a RTEC pin-on-disk tribometer. Centerline average roughness Ra of the disk specimen surfaces is about 0.8 μm, to allow a certain amount of asperity rubbing at low speed conditions and to facilitate the comparison of the lubricant performances throughout a wide range of lubrication status but similar levels of roughness influence so as to reveal the influences of the targeted lubricants. Two test track conditions, the spiral-track and the circular-track, were used. The former allows direct wear comparison without the wear debris influence, while the latter supports an overall friction observation and comparison in the form of the Stribeck curves.

The results indicate that the friction performances of the selected PAGs, mineral oils, PAOs, and esters generally obey the trend of a typical Stribeck curve, and that the increase in viscosity decreases the minimum friction coefficient value in the trough of each Stribeck curve.

Friction and the lubricant molecular structures may be correlated. For the esters with a branched-chain structure, increasing the number of branches or carbon atoms can improve the low-friction performance of the ester lubricants. For the esters with the phenyl group, the 1,3-benzenes ester outperforms the orthophthalics ester. Tribological performance of the esters with the chain structures seems to be better than those with the phenyl group when the difference in their viscosities is small. In most cases, wear can be inversely correlated with the viscosities of the oils of the same type.

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