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Development of a pre-screening-method for the qualification of environmentally acceptable lubricants for stern tube systems—The evaluation of friction efficiency and scuffing capacity

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Abstract

Lubricants for stern tube systems are subject to increasingly stringent requirements in terms of environmental compatibility, as any lubricant leakage leads to contamination of the sea. As a result, interest in environmentally acceptable lubricants (EALs) is increasing. Biodegradable ester lubricants represent a possible alternative to conventional, mineral oil-based lubricants. In addition to environmental compatibility requirements, these ester lubricants must also meet tribological requirements. In this study, suitable tribometers were therefore combined in a pre-screening method to cost-effectively test lubricants for their tribological requirements (friction efficiency, wear protection and scuffing capacity). For this purpose, the contact parameters are transferred from the real conditions in the stern tube system's journal bearing and from the standardized component tests (FZG for gears and FE8 for rolling bearings) to tribological model tests on a Mini-Traction-Machine (MTM, PCS Instruments). The result of the study is a method for the pre-screening of EALs under the application-related journal bearing and standardized component test conditions, respectively. The results show an influence of the lubricant composition on the friction efficiency in fluid friction as well as the transition point from fluid to mixed friction. Furthermore, an influence of the lubricant composition on the scuffing capacity could be shown. The results of friction efficiency and scuffing capacity confirm also a high reproducibility for those pre-screening tests.

Entwicklung eines Pre-Screening-Verfahrens zur Qualifizierung von umweltverträglichen Schmierstoffen für Stevenrohrsysteme – Bewertung der Reibungseffizienz und der Fresstragfähigkeit

Zusammenfassung

Schmierstoffe für Stevenrohrsysteme unterliegen immer strengeren Anforderungen an die Umweltverträglichkeit, da jeder Schmierstoffaustritt zu einer Verschmutzung des Meeres führt. Daher steigt das Interesse an umweltverträglichen Schmierstoffen (EALs). Biologisch abbaubare Esterschmierstoffe stellen eine mögliche Alternative zu herkömmlichen Schmierstoffen auf Mineralölbasis dar. Neben den Anforderungen an die Umweltverträglichkeit müssen diese Esterschmierstoffe auch tribologischen Anforderungen genügen. In dieser Studie wurden daher geeignete Tribometer in einem Pre-Screening-Verfahren kombiniert, um Schmierstoffe kosteneffizient auf ihre tribologischen Anforderungen (Reibungseffizienz, Verschleißschutz und Fresstragfähigkeit) zu testen. Dazu werden die Kontaktparameter aus den realen Bedingungen im Gleitlager des Stevenrohrsystems und aus den genormten Komponententests (FZG für Zahnräder und FE8 für Wälzlager) in tribologische Modellversuche auf einer Mini-Traction-Machine (MTM, PCS Instruments) übertragen. Das Ergebnis der Studie ist eine Methode zur Vorauswahl von EALs unter den anwendungsbezogenen Gleitlager- bzw. genormten Komponententestbedingungen. Die Ergebnisse zeigen einen Einfluss der Schmierstoffzusammensetzung auf die Reibungseffizienz in der Fluidreibung sowie den Übergangspunkt von Fluid- zu Mischreibung. Des Weiteren konnte ein Einfluss der Schmierstoffzusammensetzung auf die Fresstragfähigkeit nachgewiesen werden. Die Ergebnisse der Reibungseffizienz und der Fresstragfähigkeit bestätigen auch eine hohe Reproduzierbarkeit für diese Pre-Screening-Versuche.

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1 Introduction

Ester-based environmentally acceptable lubricants (EALs) have become established for many applications, for example in the field of hydraulics, as they exhibit good biodegradability, low toxicology and low coefficients of friction. In the marine sector, the requirements for the environmental compatibility of lubricants are steadily increasing. This is due to legal requirements justified by the current annual pollution of the world's oceans with around 80 million liters of mineral oil-based lubricants [1]. A significant portion of this oil loss is due to operational leakage from the ship's stern tube system. The stern tube system guides the propeller shaft from the ship's interior to the marine environment and, in the case of a classical propulsion concept, includes an oil-lubricated journal bearing that is sealed both to the sea and to the ship's interior. In the case of an Azipod propulsion concept, the stern tube system includes an additional gearbox. The gearbox consists of gears and rolling bearings and, due to its integrated design, usually shares a lubricant circuit with the stern tube journal bearing. Therefore, it is necessary that stern tube lubricants meet the tribological requirements, e.g. friction reduction, wear protection and scuffing capacity, of all machine elements of the stern tube system (journal bearing, rolling bearing, gear). Testing the tribological requirements on the entire stern tube system or in component tests (e.g. rolling bearing test FE8, gearbox test FZG) is usually associated with such high costs and material expenses that a comprehensive investigation of different lubricants is not practical. The subject of this article is therefore the selection of suitable test methods and test parameters for simple tribometer tests. These test methods are combined within a multi-phase pre-screening method to increase efficiency in terms of material expense and cost. This pre-screening method aims to make a pre-screening of the different lubricants for stern tube systems for the subsequent component tests.

2 State of the art

In the stern tube system, the contact conditions (lubricant temperature, bearing pressure, sliding speed) generally depend on the operating condition of the ship, i.e. steady running ahead, starting or stopping. During steady running ahead and start/stop operation, lubricant temperatures of approximately 40 °C are present in the journal bearing [2]. Nominal operation at steady running ahead is not considered to be critical for wear [2–4] and is therefore not considered in detail.

Critical for wear is the transition to mixed friction at low sliding speeds, for example, due to start/stop operations [3, 5]. In the literature, simulations have shown that locally higher contact pressures of about 10 MPa occur during start/stop operations at mean bearing pressures of 2 MPa, which are more critical in terms of wear [5].

Therefore, for a pre-screening of a lubricant, the conditions in start/stop operation should be transferred to a tribometer test. In [6] tribometer tests with a pin-on-disc setup were used to evaluate journal bearing materials with different lubricants. The replication of start/stop operation of a journal bearing on a pin-on-disc tribometer however has not yet been described in the literature.

In addition to the reliable function of the lubricant under the journal-bearing-specific operating conditions, the requirements of the upstream gear unit must also be met in the case of the Azipod design. Due to the requirements of the gearbox, typical lubricants for stern tube systems belong to the group of gear oils (CLP). In addition to the tribological requirements typical of gear oils in terms of the lubricant's scuffing capacity and wear protection, friction reduction by the lubricant plays an important role. A high friction reduction in the machine elements affects the energy efficiency of the entire drive system. Lubricant suitability for the machine elements rolling bearings and gears is usually tested in standardized component tests (FE8, FZG). These component tests are used to assess the wear protection (according to DIN 51819-3) and the scuffing capacity (according to DIN 14635-1). The component tests described are associated with high costs and high material input. To reduce costs and material input, pre-screening tests are therefore often carried out using tribometer measurements to select suitable materials and lubricants at an early stage of development. Ball-on-disc tribometers [7–9] are used in the literature as pre-screening tests for the wear protection, in addition to other standardized tests such as the four-ball test [10]. As a test method for wear protection, the disc of the ball-on-disc tribometer oscillates while the ball is stationary [7, 9] or rotating [8]. As a result of the oscillating motion, the wear is concentrated in a relatively small area of the disc and is therefore easy to measure. The rotating movement of the ball reduces the wear slightly compared to the variant with pure sliding movement. This reduction in wear means that the geometry of the specimens remains almost identical and the contact pressure is slightly reduced. A pre-screening test for wear protection following the lambda values and the rolling/sliding movement of the FE8 rolling bearing test is not known.

For the evaluation of friction efficiency, the ball-on-disc tribometer was able to achieve comparable results in the literature for the gears [11] and the rolling bearing [12] as in the respective component tests.

A pre-screening test for the scuffing capacity of lubricants following the FZG gearing test could be performed in the literature using various two-disc tribometers [13, 14], as well as a Timken [15] and four-ball tester [16]. The disadvantage of these pre-screening tests is that the tribological stress is controlled by a stepwise increase of the load [17–19]. As a result of the load increase, the contact area increases, which leads to previously unloaded and unprotected areas entering the contact zone, thus increasing the risk of premature scuffing [17–19]. Furthermore, pre-screening tests with pure sliding motion (Timken tester, four-ball tester) result in high wear on the stationary contact partner, which leads to a constant increase of the contact area during the test [17–19]. Due to the increasing contact area, the contact pressure and therefore the tribological stress decreases during the test. This can lead to the fact that the wear protection of the tested lubricant is decisive for the result of the scuffing capacity [17–19].

Due to these existing problems with conventional pre-screening tests for determining scuffing capacity, INGRAM has developed a pre-screening test for the Mini-Traction-Machine (MTM) in the ball-on-disc configuration. In this pre-screening test, the ball and disc are rotated in opposite directions and the sliding speed is gradually increased. By rotating the test specimens in opposite directions, INGRAM can reduce wear and also decouple the sliding speed from the rolling speed and thus from the EHL film thickness [19]. INGRAM'S pre-screening test was improved by PENG in terms of reproducibility by changing the rolling speed during the test phase and adjusting the conditions of the running-in phase. In addition, he added a mapper step before each sliding speed step to analyze the boundary layer formation using the spacer layer imaging method (SLIM) [18]. PENG'S improved method was adapted by BAYAT with regard to the test specimen (replacing the ball with barrel) in order to be able to achieve higher pressures. The higher pressure increases the tribological stress on the lubricant, which means that higher additive gear oils can be investigated in terms of scuffing capacity [17].

It can be concluded that a pre-screening method at the tribometer level for the efficient and reproducible investigation of the tribological behavior of stern tube lubricants under contact conditions close to the application and standards requires a combination of suitable sample geometries and tribological loads. Based on the suitable combination, the real tribological contact conditions on the one hand and the requirements with regard to standardized component tests for a gear oil release according to CLP on the other hand can be covered.

3 Pre-screening method

The reproduction of the tribological contact conditions (material, roughness, temperature, pressure, speed) from journal bearings, gears and rolling bearings is carried out with the MTM (PCS Instruments). The MTM test rig configurations used in this pre-screening method are ball-on-disc (Fig. 1), barrel-on-disc (Fig. 2) and pin-on-disc (Fig. 3). The developed pre-screening method was divided into different phases on the basis of the relevance for the operational safety as well as the effort (material, time). In the first phase, the EALs are classified in terms of their friction efficiency with the ball-on-disc configuration, as the effort in terms of material and time is lower compared to the other pre-screening tests.

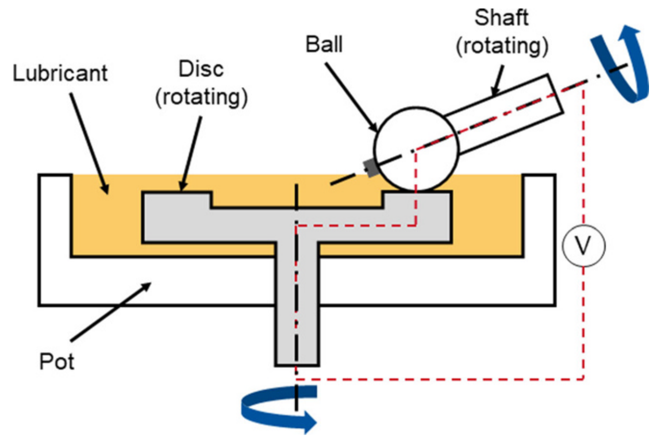

Fig. 1 Schematic representation of the ball-on-disc configuration of the MTM

In the configuration shown (Fig. 1), a ball with a diameter of 19.05 mm and a disc are used as test specimens. Both test specimens are made of 100Cr6 bearing steel and have Ra values of 0.02 µm (ball) and 0.02 µm (disc). For the evaluation of the friction efficiency, the ball and the disc are rotated in a bath with 60 °C lubricant temperature and loaded with a force of 75 N (1.25 GPa). The slide-roll ratios

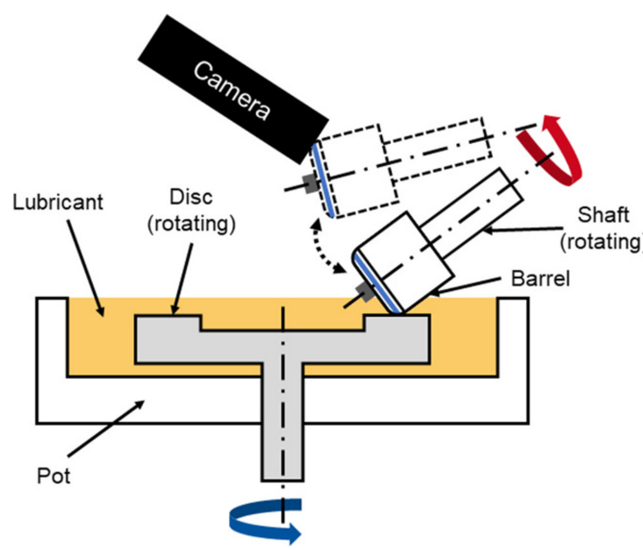

Fig. 2 Schematic representation of the barrel-on-disc configuration of the MTM

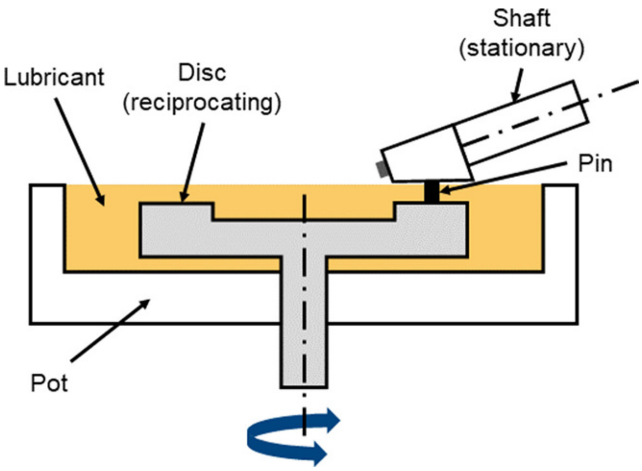

Fig. 3 Schematic representation of the pin-on-disc configuration of the MTM

(SRRs) are 30%, 60%, 90% and 120%. These parameters were chosen in accordance with BJÖRLING, who was able to achieve comparable trends with regard to friction as in the FZG test rig using a ball-on-disc tribometer [11]. The rolling speed is reduced stepwise from 2500 to 10 mm/s during the tests. In addition to the friction coefficient, the electrical contact resistance (ECR) is also considered as a measurand to enable differentiation between the full-film and mixed lubrication. Therefore, a balance resistor value of 100 Ω was chosen, because rolling speeds of less than 1000 mm/s are considered relevant for the transition point.

Subsequently, in phase two, the wear protection in rolling/sliding contact as well as the scuffing capacity are evaluated, since the stresses on the gear components (rolling bearing, gear) are considered more critical compared to the journal bearing of the stern-tube system. Pre-screening of the wear protection of a lubricant in rolling/sliding contact is also done with the ball-on-disc configuration (Fig. 1). The specimens used differ from the specimens of the friction efficiency test in terms of the Ra value of the disc (previously: 0.02 µm; now: 0.15 µm). In this configuration, a rolling speed of 20 mm/s, a force of 75 N (1.25 GPa) and a lubricant temperature of 80 °C are set. This combination leads to a strong mixed friction regime (lambda value of 0.04), which is comparable to that in a FE8 test according to DIN 51819-3 (lambda value of 0.03). For the SRR the highest value occurring in the bearing is used since wear is high in areas with high sliding components [20]. In the case of the axial cylindrical roller bearing (81212) typical for FE8 tests, this corresponds to an SRR value of ± 14% [21]. In addition to the gravimetric wear measurement typical of FE8 tests, the width and depth of the wear track on the disc are recorded by profile measurements in the radial direction at the end of the test to evaluate wear protection. In contrast to the tribometer tests with oscillating motion commonly used in the literature, this pre-screening test is based on the conditions of the FE8 rolling bearing test (lambda value, form of motion).

Pre-screening of the scuffing capacity of the lubricants is carried out according to the method of BAYAT with the barrel-on-disc configuration (Fig. 2; [17]). In this configuration, only the ball is exchanged for a barrel (Ra value = 0.09) made of 100Cr6 compared to the ball-disc configuration of the friction efficiency pre-screening test. During the tests, the barrel rotates at a lubricant temperature of 120 °C in the contrary direction to the disc. As a result of the contrary rotation of the test specimens, sliding speeds of 1200 to 7600 mm/s are achieved in the contact at constant rolling speed (200 mm/s) during the tests. The sliding speeds are increased stepwise with a step duration of 30 s. The sliding speed profile is run successively at three different maximum pressures (1.97 GPa, 2.67 GPa, 3.07 GPa) to further increase the tribological stress for the fully formulated EALs. In contrast to other pre-screening tests, in which tribological stress is controlled by gradually increasing the pressure, running-in phases with the respective pressures are carried out before each pressure increase. The running-in phase allows tribological layer formation to take place in the fresh areas of the contact zone and prevents premature scuffing. In this pre-screening test, scuffing damage is present as soon as the coefficient of friction exceeds a value of 0.2 for 5 s or strong vibrations and high noise levels emanating from the MTM. In order to compare the occurrence of scuffing damage and thus the scuffing capacity of different lubricants, the friction energy intensity (FPI) is determined in the literature [17, 18, 22]. The FPI is the result of the multiplication of the mean pressure in the contact (p_{mean}), the coefficient of friction before scuffing occurs (*COF*) and the sliding speed in the moment of scuffing failure (u_s):

$$FPI = p_{\text{mean}} \cdot COF \cdot u_s \qquad (1)$$

In addition to the determination of the FPI, the SLIM method is used to evaluate the formation and degradation of the tribological layer during the test. For this purpose, a SLIM image is taken before each change in sliding speed and after the scuffing damage.

In the final phase, an evaluation of the EALs for use in the journal bearing of the stern-tube system takes place. The pin-on-disc configuration of the MTM (Fig. 3) is used to represent the contact conditions of journal bearing during start/stop operations. In the pin-on-disc configuration, a 3 mm diameter white metal pin and a 100Cr6 disc are used as test specimens. The initial roughness is in the range of Ra 2.2 µm for the pin and Ra 0.15 µm for the disc. The disc oscillates at a certain angle with a frequency of 6 Hz. Due to this oscillating angular movement, a stroke length of 16 mm (distance between the turning points) is set in the

Table 1 Characterization of the tested lubricants

	Kin. Viscosity at 40 °C (mm^2/s)	VI (–)
Mineral oil-based lubricant	100	~94
Polyol + AW1	104	~135
Complex + AW1	105	~160
Complex + AW2	105	~160

contact point on the disc to replicate the start/stop operations. The pin remains stationary so that a maximum sliding speed of 300 mm/s is achieved.

During the tests, the pin is loaded with a force of 70 N, which corresponds to a nominal surface pressure of 10 MPa. The lubricant temperature is controlled to the highest lubricant temperature occurring in the application (40 °C). The measurement parameters are the coefficient of friction as well as the decrease in length and weight of the pin as a result of wear. From the measurement parameters, the wear protection and friction efficiency of a lubricant in the journal bearing are evaluated. The reproduction of the start/stop processes could be represented straightforwardly and with a high number of cycles on the MTM by using the oscillating motion of the pin-on-disc tribometer.

In this paper, the test results of the friction efficiency analysis (first phase) and the evaluation of scuffing capacity (second phase) are presented. For these tests three fully formulated environmentally acceptable lubricants (EAL) and one mineral oil-based lubricant (reference) of ISO VG class 100 were used. The used EALs differ in terms of their base oil (polyol and complex ester) and their EP/AW additive package (AW1 and AW2). The mineral oil-based lubricant is a commercially available gear oil that meets the requirements of the CLP standard. All lubricants used contain ashless EP/AW additives based on phosphorus and sulfur. A characterization of the used lubricants is shown in Table 1.

The lubricants were tested with different configurations of the MTM. For the first phase (analysis of friction efficiency) the ball-on-disc configuration was used (Fig. 1). The parameter set used in this phase is in accordance with the previously described pre-screening test for friction efficiency (60 °C; 1.25 GPa; 30%, 60%, 90%, 120%; 2500–10 mm/s). The two best performing lubricants of the first phase are further investigated with regard to their scuffing capacity (second phase). For the evaluation of scuffing capacity, the barrel-on-disc configuration (Fig. 2) and the parameter set of the corresponding pre-screening test (120 °C; 1.97 GPa, 2.67 GPa, 3.07 GPa; 250–3600%; 1200–7600 mm/s) was used. In each phase, three separate tests are performed for each lubricant.

4 Results

The results of the first phase (analysis of friction efficiency) are presented exemplary for 60% SRR in the diagrams in Fig. 4. The diagrams show the ECR (a) and the coefficient of friction (b) versus the logarithmic rolling speed. The curves in those diagrams represent the mean values of the ECR and coefficient of friction from three individual tests. The upper diagram (a) shows the ECR values of the polyol with AW1 (PAW1; blue circle), complex ester with AW1 (CAW1; green cross), complex ester with AW2 (CAW2; yellow triangle) and the mineral oil-based lubricant (MIN; brown rectangle) versus the logarithmic rolling speed. PAW1 has an ECR value of 100% for the rolling speed range from 2500 to 50 mm/s. For the complex esters the ECR value is 100% for the rolling speed range from 2500 to 400 mm/s (CAW1) and from 2500 to 60 mm/s (CAW2). The mineral oil-based lubricant (MIN) has an ECR value of 100% for the rolling speed range from 2500 to 90 mm/s. These rolling speed ranges represent the fluid friction regime. The transition from fluid to mixed friction (transition point) is at rolling speeds of 50 mm/s for PAW1, 400 mm/s for CAW1, 60 mm/s for CAW2 and for

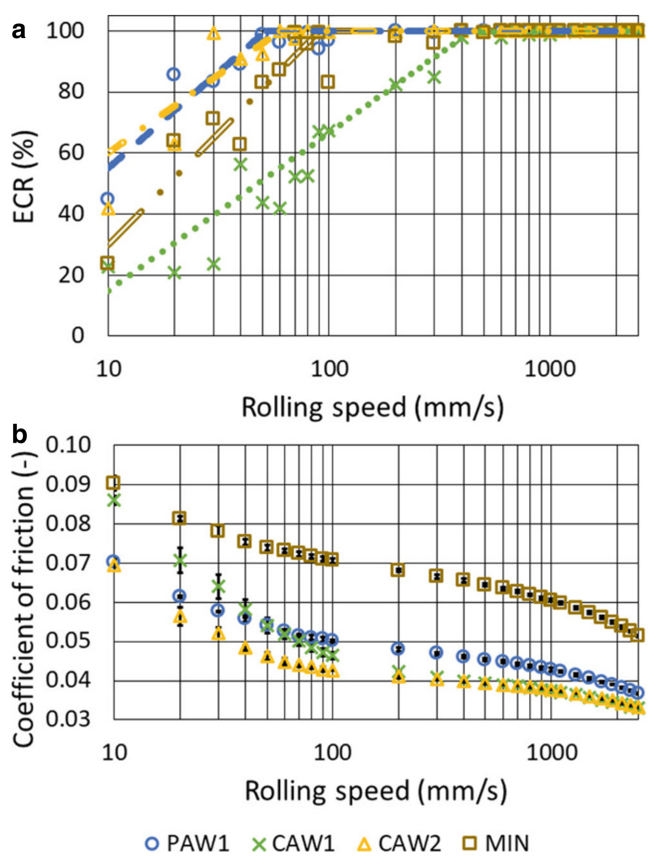

Fig. 4 Electrical contact resistance (**a**) and coefficient of friction (**b**) versus rolling speed for 60% SRR for PAW1 (*blue circle*), CAW1 (*green cross*), CAW2 (*yellow triangle*) and MIN (*brown rectangle*)

MIN at 90 mm/s. The comparison of the transition points shows that the solid body contact occurs at lower rolling speeds for PAW1, CAW2 and MIN than for CAW1. The transition points of PAW1, CAW2 and MIN are similar. The reason for the later occurrence of the solid contact between the lubricants could be due to a different pressure viscosity index of the lubricants or to a thicker and more stable adsorption layer [23]. An adsorption layer forms on the metal surface as a result of the adsorption of polar lubricant components [24]. Higher polarity of these lubricant components could lead to a higher accumulation of more stable components and thus to a thicker and more stable adsorption layer. In addition, the bond between the metal surface and lubricant components with higher polarity is stronger [25]. Due to the stronger bond, lubricant components attach to the metal surfaces and remain there even at low rolling speeds and thus low lubricant film heights. Therefore, a separating lubricating film remains at low rolling speeds. Since the relevant rolling speeds for the transition points are below 1000 mm/s the choice of the balance resistor value was correct.

In the next step the friction efficiency was evaluated using the coefficients of friction. The coefficients of friction of PAW1 (blue circle), CAW1 (green cross), CAW2 (yellow triangle) and MIN (brown rectangle) versus the logarithmic rolling speed are shown in the bottom diagram (b) of Fig. 4. First, the fluid friction regime is considered (rolling speed >400 mm/s). The coefficient of friction in the fluid friction regime decreases for all lubricants with increasing rolling speed due to the shear heating in the contact zone [26]. The shear heating in the contact zone had no influence on the temperature of the lubricant, which remained constant during the short test period (~15 min per test). The values for the coefficient of friction in the fluid friction regime differ depending on base oil type and not on the additive package used. The two complex esters (CAW1 and CAW2) have similar and lower coefficients of friction than the polyol ester (PAW1). The reason for the difference between the two ester types lies in the different ester structures of the base oils used, because the base oil represents the largest part of the lubricant components in contact in this friction regime. Here, the chain length and the degree of branching of the ester structure are decisive for the coefficient of friction [27]. Compared to MIN, all EALs have lower coefficients of friction. The reason for the difference between esters and mineral oils could be the different lubricant properties such as the VI index or the pressure-viscosity coefficient [28].

In the mixed friction regime (PAW1, CAW2: <50–60 mm/s; CAW1: <400 mm/s) the coefficient of friction for all EAL increases more sharply than in the fluid friction regime due to the increasing solid body contact. However, MIN does not exhibit a sharp increase in the coefficient of friction in the mixed friction regime (MIN: <90 mm/s). As the rolling speed decreases, the coefficients of friction of CAW1 and MIN as well as PAW1 and CAW2 converge. Nevertheless, the coefficients of friction of MIN are higher than those of EALs in the entire range of mixed friction.

The reproducibility of the pre-screening test for the evaluation of friction efficiency was investigated by comparing the maximum standard deviation of the coefficients of friction of the lubricants tested. Figure 5 shows the maximum standard deviations of the coefficients of friction from each lubricant. The highest maximum standard deviation of the coefficient of friction is achieved by CAW1 with 0.0032 and the lowest by PAW1 with 0.0012. These values of the maximum standard deviation correspond to a percentage deviation of 4.59% (CAW1) and 1.68% (PAW1), which indicates a high reproducibility for the presented pre-screening test.

For further investigation in terms of scuffing capacity (second phase) the two lubricants with the best performance from the first phase (friction efficiency) were selected. In the first phase, CAW2 showed both a low transition point (60 mm/s) and the lowest friction values over the entire

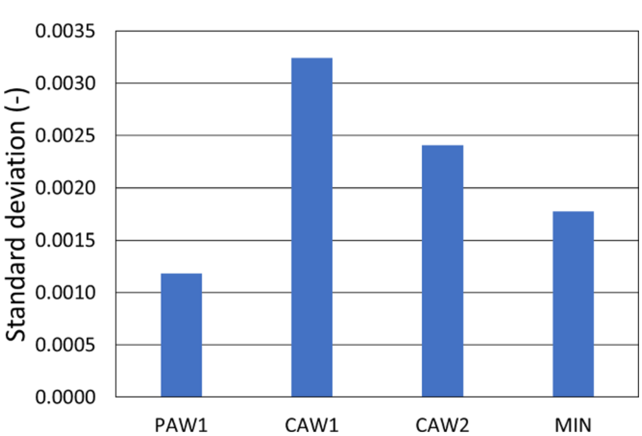

Fig. 5 Comparison of the maximum standard deviations of the tested lubricants

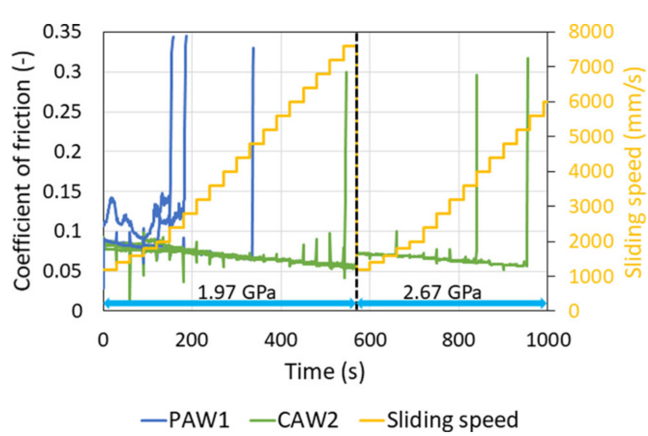

Fig. 6 Coefficient of friction during the pre-screening of scuffing capacity for PAW1 (*blue*) CAW2 (*green*)

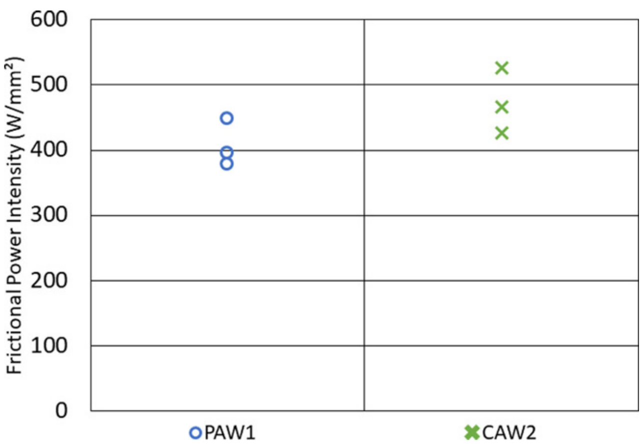

Fig. 7 Comparison of the frictional power intensities of PAW1 (*blue circle*) and CAW2 (*green circle*)

range of rolling speeds tested. PAW1 was chosen as the second lubricant for the second phase because the sum of low transition point (50 mm/s < 400 mm/s resp. 90 mm/s) and low friction is superior to the other lubricants (MIN, CAW1).

For this reason, the results of the pre-screening of scuffing capacity of CAW2 and PAW1 are shown in the diagram in Fig. 6. The diagram shows the time course of the coefficient of friction for PAW1 (blue line) and CAW2 (green line) during the stepwise increase of the sliding speed (yellow line) at two different maximum pressures (1.97 GPa, 2.67 GPa). Each lubricant was tested in three separate tests. The scuffing damage can be seen in the diagram by a sharp increase in the coefficient of friction above a value of 0.2. The comparison of failure times due to scuffing shows that CAW2 fails at higher sliding velocities (7600 mm/s) and at a higher pressure (2.67 GPa) due to scuffing than PAW1. This result indicates a higher scuffing capacity for CAW2 compared to PAW1. One reason for this could be the higher VI index of CAW2 (~160) compared to PAW1 (~135) [17]. Due to the higher VI index, the viscosity decreases more slowly with increasing contact temperature, which means that a protective lubricating film is maintained even at higher sliding speeds and pressures and thus higher contact temperatures.

For further investigation, the FPI values were calculated based on Eq. 1 with the mean pressures 1.31 GPa (maximum pressure of 1.97 GPa) and 1.79 GPa (maximum pressure of 2.67 GPa). The calculated values for PAW1 (blue circle) and CAW2 (green cross) are shown in the diagram in Fig. 7. The mean values of the three FPI values from Fig. 7 show a lower mean FPI value for PAW1 (~407 W/mm²) than for CAW2 (~473 W/mm²), indicating a better scuffing capacity for CAW2. These mean FPI values are comparable to the results of BAYAT who achieved a mean value of 446 W/mm² for an similar EAL [17]. Also, the results in Fig. 7 show a high reproducibility of the used pre-screening test since the standard deviations are 30 W/mm² for PAW1 and 40 W/mm² for CAW2. These standard deviation values correspond to a percentage deviation from their mean FPI value of 7.4% (PAW1) and 8.6% (CAW2).

5 Summary and outlook

Due to legal requirements for environmental compatibility, there is a motivation to use environmentally acceptable ester lubricants in stern tube systems. For the cost-efficient development of corresponding lubricants, a pre-screening method for component and standard-related tests was developed to reduce the necessary component tests for suitability testing and thus save costs. For the development of the pre-screening method, suitable test methods were selected according to the tribological requirements for a lubricant in the stern tube system (friction reduction, wear and scuffing protection) and the contact conditions for tribometer tests (MTM) were derived. The results of the first phase (analysis of friction efficiency) and preliminary results of the second phase (evaluation of scuffing capacity) were presented in this study. For the first phase, an influence of the lubricant composition on the transition from fluid to mixed friction was shown. The reason for the later transition to mixed friction could be a different pressure viscosity index of the lubricants or the formation of thicker and more stable adsorption layers. One explanation for the formation of thicker and more stable adsorption layers could be that lubricant compositions with more polar components are able to form stronger bonds with the metal surfaces. Due to this stronger bond the lubricant components remain in the contact zone at low rolling speeds and low lubricant film heights. Therefore, these lubricant components ensure a separating lubricating film even at lower rolling speeds. Also, the dominating influence of ester structure in fluid friction regime was shown. The complex esters showed lower friction than the polyol esters in the fluid friction regime. The complex ester with AW2 exhibited the highest overall friction efficiency over the entire range of rolling speeds tested, followed by the polyol ester with AW1. These two lubricants were further investigated regarding their scuffing capacity (second phase). The results of the second phase showed a higher scuffing capacity for the complex ester with AW2 than for the polyol ester with AW1. One reason for this could be the higher VI index of CAW2 which results in thicker EHL films at higher contact temperatures. The presented pre-screening tests for the evaluation of friction efficiency and scuffing capacity showed a high reproducibility for the tested lubricants. In the next step, the wear protection of the lubricants in rolling/sliding contact (phase two) and in journal bearings during start/stop operations (phase three) will be investigated with the tests of the pre-screening method.

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