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POLISH MARITIME RESEARCH 4 (112) 2021 Vol. 28; pp. 160-166
10.2478/pomr-2021-0058

ASSESSING THE POTENTIAL REPLACEMENT OF MINERAL OIL WITH ENVIRONMENTALLY ACCEPTABLE LUBRICANTS IN A STERN TUBE BEARING: AN EXPERIMENTAL ANALYSIS OF BEARING PERFORMANCE

Jerzy Kowalski
Gdańsk University of Technology, Poland

* Corresponding author: *jerzy.kowalski@pg.edu.pl (J. Kowalski)*

ABSTRACT

This study compares the performance of a plain bearing, with a similar structure to a tail shaft stern bearing, lubricated with either mineral oil or an environmentally acceptable lubricant (EAL). The main characteristic of the bearing is its length/diameter ratio of <1. Measurements are carried out with the bearing operating under loads from 0.5 to 1 MPa and seven speeds ranging from 1 to 11 rev/s. The bearing lubricated with either mineral oil with a viscosity grade of 100 or an environmentally acceptable lubricant (EAL) with a viscosity grade of 100 or 150 is investigated according to the ISO standard. Bearing wear is simulated by increasing the clearance circle by 0.1 mm. According to the results obtained, the use of an EAL in place of mineral oil does not cause significant changes in the bearing performance, regardless of the value of the clearance radius. The pressure distribution in the oil film, bearing load carrying capacity, eccentricity and friction coefficient have similar values for the entire load and speed ranges considered, and the discrepancies in the results are within the range of the measurement errors. Only an increase in EAL viscosity causes significant changes in bearing performance and these changes comply with the general theory of lubrication.

Keywords: environmentally acceptable lubricants (EALs), mineral oil, stern tube bearing, bearing wear, marine bearing

INTRODUCTION

A tail shaft scabbard bearing is an integral part of the propulsion system of a ship. The standard structure of this tribological node is a hydrodynamically lubricated transverse plain bearing. In this case, mineral oil in a circulating system acts as the lubricant. Such a system allows for control of the flow, temperature and cleanliness of the oil, which translates to durability, reliability and high energy efficiency of the bearing. In contrast, oil circulation is isolated from the marine environment by a sealing system, which, despite its complex structure, is not perfect. According to [1], ~2.6 litres per day of lubricating oil leak into marine environments from the tail shaft scabbard of an average ship. The accumulation at sea of oil from shipboard tubes causes significant environmental degradation.

For this reason, lubricants are sought that meet the requirements set for oils intended for lubricating plain bearings and are simultaneously environmentally friendly. One of the first legal standardisations in this area was presented by the United States, which, as early as 2013, introduced a new class of lubricating oils, referred to as environmentally acceptable lubricants (EALs) [2]. Generally, EALs are defined as biodegradable, non-bioaccumulative and minimally harmful to aquatic environments. The simplest solution is to use water as a lubricant [3][6], which is undoubtedly an environmentally inert substance and a common lubricant in historical guava bearings. However, its low viscosity contributes to a relatively low bearing capacity. In recent years, virtually all major lubricant manufacturers have introduced EALs into their range of lubricating oils. For the most part, these are oils produced on the basis of hydrocarbons of biological origin with various structures [7], although, according to the manufacturers' declarations, some substances are water based. Commercially available EALs have physical parameters similar to the parameters of mineral oil. Despite this, shipowners continue to use mineral oil for fear of, among other things, EALs becoming old and possibly losing the required bearing capacity of stern bearings and intensifying bearing pan wear as a consequence. This phenomenon is clearly visible in the engine lubrication oils [8]

The presented condition constitutes the primary motivation for the undertaken research, which seeks to compare the operation of a plain bearing, with a structure similar to that of a stern bearing of a ship, lubricated with either mineral oil or an EAL. Although the issue of EAL aging was presented in a multi-faceted manner in [9], an analysis of the literature on the subject confirms that there are relatively few studies that address this issue. The performance parameters for mineral oil and EALs with relatively high viscosity under elastohydrodynamic lubrication conditions have been investigated [10], while in [11], the rheological properties of EALs were addressed. A comparison of mineral oil and EAL performance was presented in [12]; however, this was for a relatively narrow bearing with a structure not used in tail shaft scabbards and a limited range of load conditions. A broader range of studies on water-lubricated stern bearings can be found, as this structure solution is increasingly used in the shipbuilding industry. For example, problems of structure and the performance of water-lubricated bearings have been presented [13] and [14], while [15] and [16] considered the influence of stern tube bearing deformations on the lubrication parameters. For this reason, this study compares a wide range of performance criteria of a stern bearing under varying load conditions and degrees of wear and lubricated with mineral oil and two types of EALs.

LABORATORY TESTS

Measurements were made on the test stand demonstrated in Fig. 1 and the pan structure in Fig. 2. A shaft with a diameter of 100 mm was supported by two roller bearings and driven by an electric motor. The structure of the stand allowed the shaft speed to be adjusted and measured. A 200 mm wide alloy bearing pan casing was mounted on the shaft, complete with a seal and lubricating oil inlet and outlet ports. The casing could accommodate 16 sensors for measuring the high-speed hydrodynamic pressure and four shaft position sensors relative to the pan. The pan casing was supported from below by an independently powered hydraulic cushion, which allowed the plain bearing to be loaded. By using a cushion, it was also possible to measure friction in the bearing. Table 1 shows the parameters of the tested lubricants. Measurements were made using a DV-1 rotational viscometer and a DMA35N petrol densitometer.

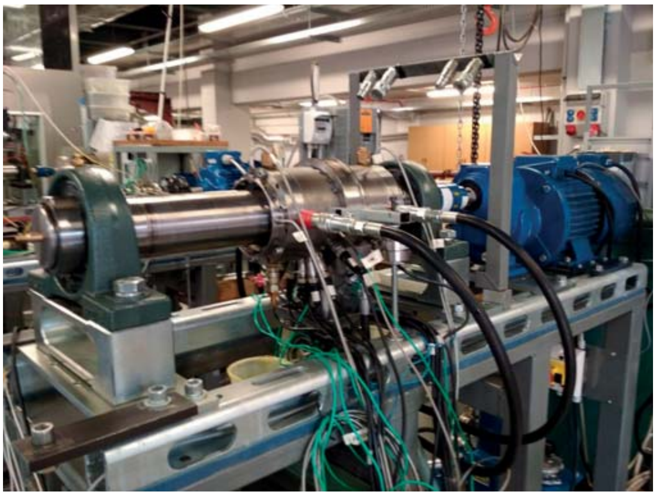


Fig.1. Laboratory stand

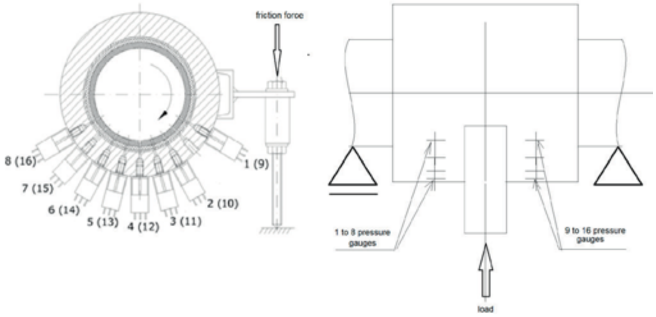


Fig.2. Laboratory stand scheme

Tab. 1. Parameters of mineral oil and EALs

Lubricant	Viscosity@25 °C [mPa s]	Viscosity@40 °C [mPa s]	Density@40 °C [kg/m^3]	Viscosity index*
Mineral oil	222.38	93.52	873.8	97.4
EAL 1	176.70	94.29	910.3	98.5
EAL 2	290.53	145.25	914.7	98.6

*Viscosity@100 °C is extrapolated

The bearing parameters were measured in six stages each with three observations whilst maintaining the oil temperature in the bearing constant at 40±0.5 °C. A white bearing alloy pan with a length/diameter ratio of two was used. This value is close to the aft bearing structure of the tail shaft. Each stage measured the operating parameters of the plain bearing lubricated with the tested oil for three average pressures of 0.50, 0.75 and 1.00 MPa and seven rotational speeds of 11, 9, 7, 5, 3, 2 and 1 rev/s. The hydrodynamic pressure in the bearing lubrication wedge was measured at 16 measurement points, as shown in Fig. 2. The position of the shaft relative to the pan at both ends of the pan, the oil temperature before, after and inside the pan, the shaft speed, the bearing friction torque and the force loading of the bearing were also determined. The parameters of the measuring sensors are presented in Table 2. The load and speed ranges are given in Table 3.

Tab. 2. Parameters of measurement gauges

Gauge	Method	Accuracy	Range
Pressure	Piezoelectric	0.3%	0–6 MPa
Rotational speed	Tachometric	±2.5 rpm	0–6000 rpm
Temperature	K-type	±0.5 °C	0–100 °C
Shaft centre position	Proximity probe	±1%	0–10 mm
Friction force	Tensometric	±0.25%	0–100 N

Tab. 3. Measurement stages

Stage*	Lubricant	Diameter clearance [mm]
1	Mineral oil	0.34
2	EAL 1	0.34
3	EAL 2	0.34
4	Mineral oil	0.44
5	EAL 1	0.44
6	EAL 2	0.44

*Each stage consists of three average pressures of 0.50, 0.75 and 1.00 MPa and seven rotational speeds of 11, 9, 7, 5, 3, 2 and 1 rev/s. Three observations for each stage are considered.

In order to test the influence of bearing wear on the bearing performance, measurements were also carried out for a reduced shaft diameter (larger clearance circle) of 0.1 mm.

RESULTS AND DISCUSSION

This study compares the performance of a stern bearing of a ship lubricated using mineral oil and EALs. A mineral oil with a viscosity grade of 100, according to the ISO standard (kinematic viscosity in mm^2/s at 40 °C), was adopted as the base oil. In order to select a representative EAL, some viscosity and density tests were conducted for commercially available EALs. Figure 3 presents the measurement results. According to the presented results, it can be seen that EALs of the same viscosity grade have a higher density compared to mineral oil. In Table 1, it can also be observed that the EALs relative to mineral oil show a reduced viscosity change together with the temperature change. This phenomenon is confirmed by the viscosity index values.

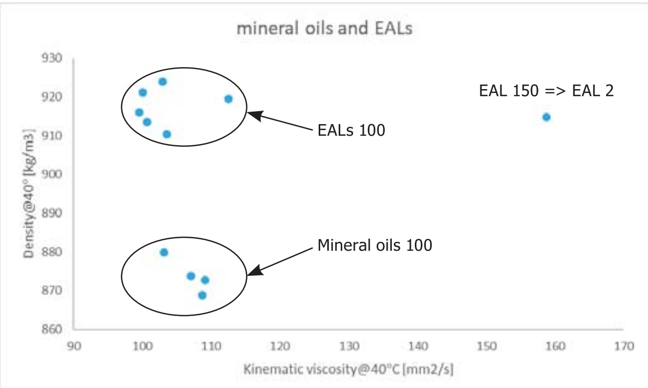


Fig.3. Parameters of mineral oils and EAL's

The essential parameter of the stern bearing from the perspective of the energy balance of the ship is its friction loss. Obviously, the friction coefficient in a bearing lowers the viscosity of the lubricating oil. Unfortunately, the reduction in oil viscosity also contributes to a substantial reduction in the load-bearing capacity. Figure 4 presents the influence of bearing speed on the friction coefficient. An increase in the bearing speed increases the friction coefficient for both the mineral oil and EALs, as expected. The Hersey–Stribeck relation illustrates the characteristics of these changes. It is noteworthy that replacing the mineral oil with an EAL does not cause significant changes in the friction factor, assuming that the viscosity of both oils is similar. A slight increase in the friction coefficient is observed for EAL 1 compared to the mineral oil at a load of 0.75 MPa, but this change does not exceed the measurement error. Only the use of the EAL with a higher viscosity grade results in a significant increase in the friction coefficient. It is also interesting that an increase in bearing load causes a significant decrease in the friction coefficient.

When the measurements were conducted, the oil film pressure was also recorded using 16 pressure sensors, arranged as presented in Fig. 1. The pressure distribution in the oil film in the two bearing cross sections was obtained on this basis. As expected, the highest pressure was observed at the location of the sensors numbered 5 and 13. These are sensors with a banding angle of 105° counting in the direction of rotation of the bearing. Example measurement results for a load equal to 1 MPa and a speed of 11 rev/s are shown in Fig. 5(a). The horizontal axis shows the angle position of the sensors. The maximum pressures for all considered bearing speeds and loads are presented in Figs. 5(b)-(d).

According to the results presented, an increase in rotational speed causes a decrease in the maximum pressure in the oil film. The classic theory of hydrodynamic lubrication presents an increase in the relative velocity of the lubricated surfaces with a decrease in the Sommerfeld number, which translates to a reduction of the eccentricity of the bearing (e). In contrast, the bearing eccentricity is in the basic equation of pressure in the third exponent in the denominator. Despite the increase in sliding speed (and rotational speed), the pressure in the bearing decreases. The results presented in Fig. 5 also show that the maximum oil film pressure increases with increasing load. Using EALs instead of mineral oil with the same viscosity does not cause quantitative or qualitative changes in the maximum pressure. Only the use of EALs of higher viscosity causes a noticeable reduction in the maximum oil film pressure, but only at relatively low bearing speeds. This result is consistent with the classic theory of lubrication. An increase in oil viscosity results in a decrease in the Sommerfeld number and bearing eccentricity.

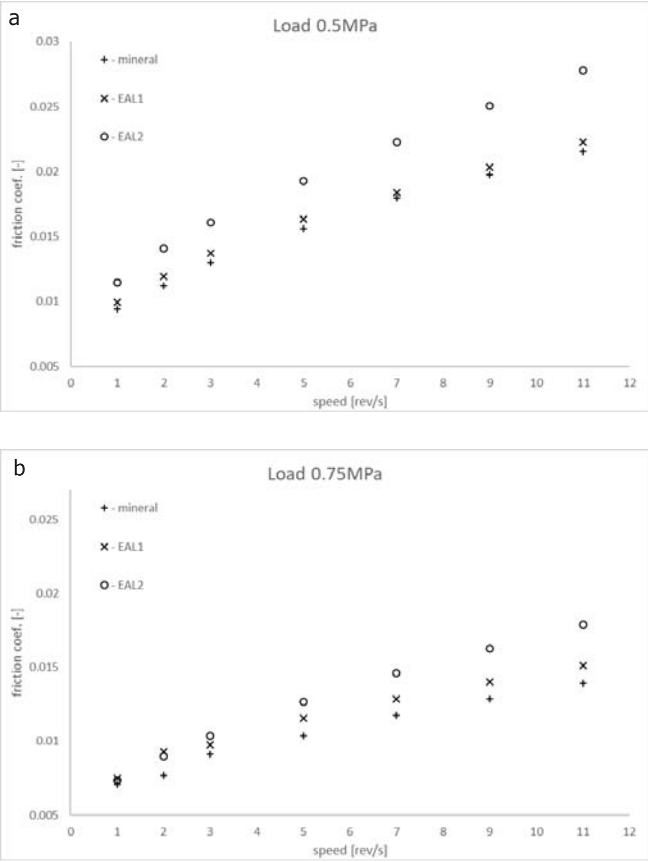


Fig.4 – Friction coefficients for mineral oil and EAL's

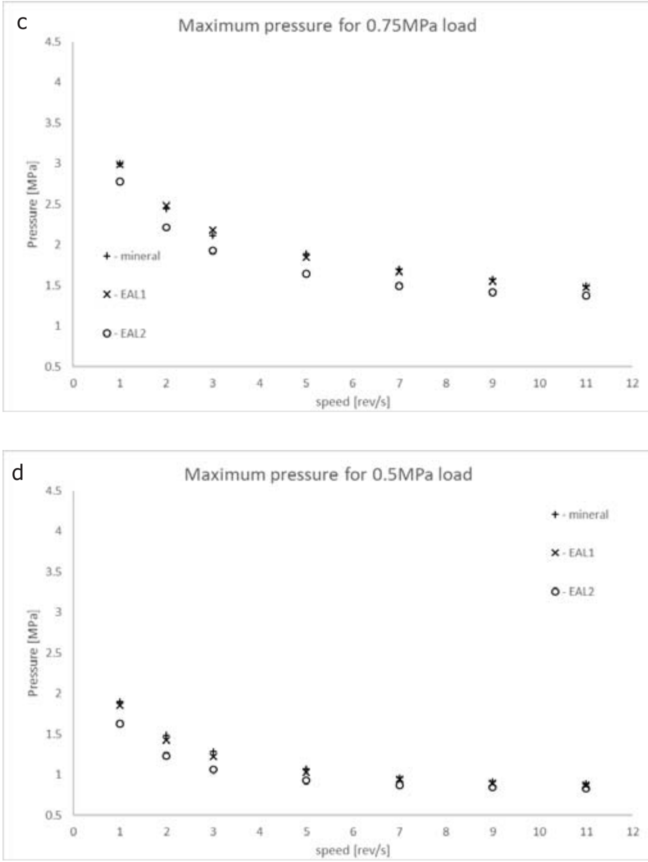
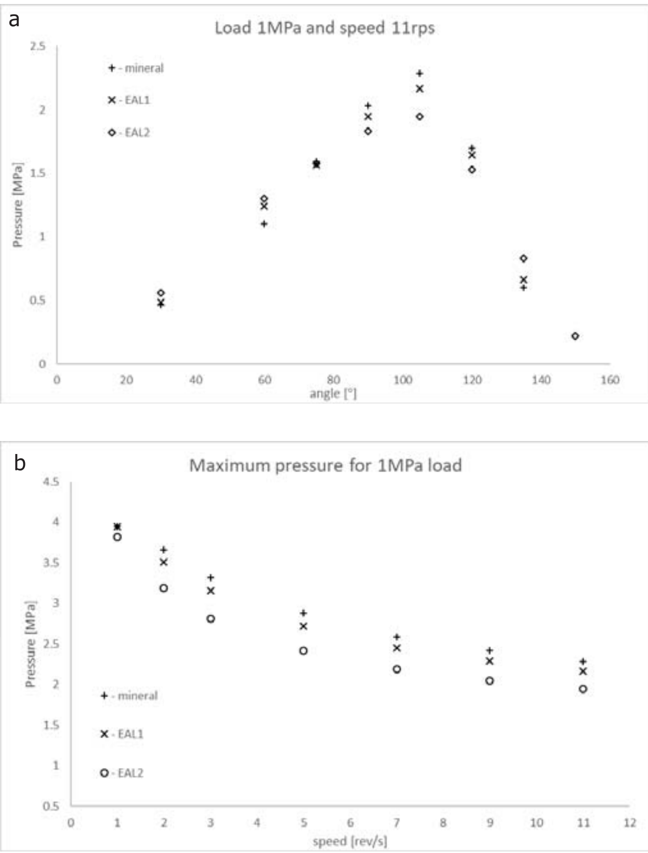


Fig.5. Pressures in the oil film; a – example of pressure values for 1MPa and11rev/s, b-d – maximum pressure for different bearing loads and speeds



The instantaneous position of the shaft in the bearing was also recorded during the measurements. According to the obtained results, the position of the shaft oscillates during bearing operation with a frequency proportional to its rotational speed. This phenomenon is neglected in the classic theory of lubrication. The instantaneous variable position of the shaft is reduced to eccentricity, with a constant value representing the average value of the position of the shaft axis relative to the pan axis. Figure 6 presents the eccentricity for the considered bearing speeds and loads. The results show that a rotational speed of 5 rev/s causes a decrease in eccentricity and thus an increase in oil film thickness. A further increase in the rotational speed does not change the eccentricity value of the bearing. An increase in eccentricity with increasing load was also observed. The analysis also showed that, according to Fig. 6, the use of an EAL with a viscosity close to that of mineral oil does not affect the shaft positions in the bearing and an increase in viscosity reduces the eccentricity and increases the oil film thickness.

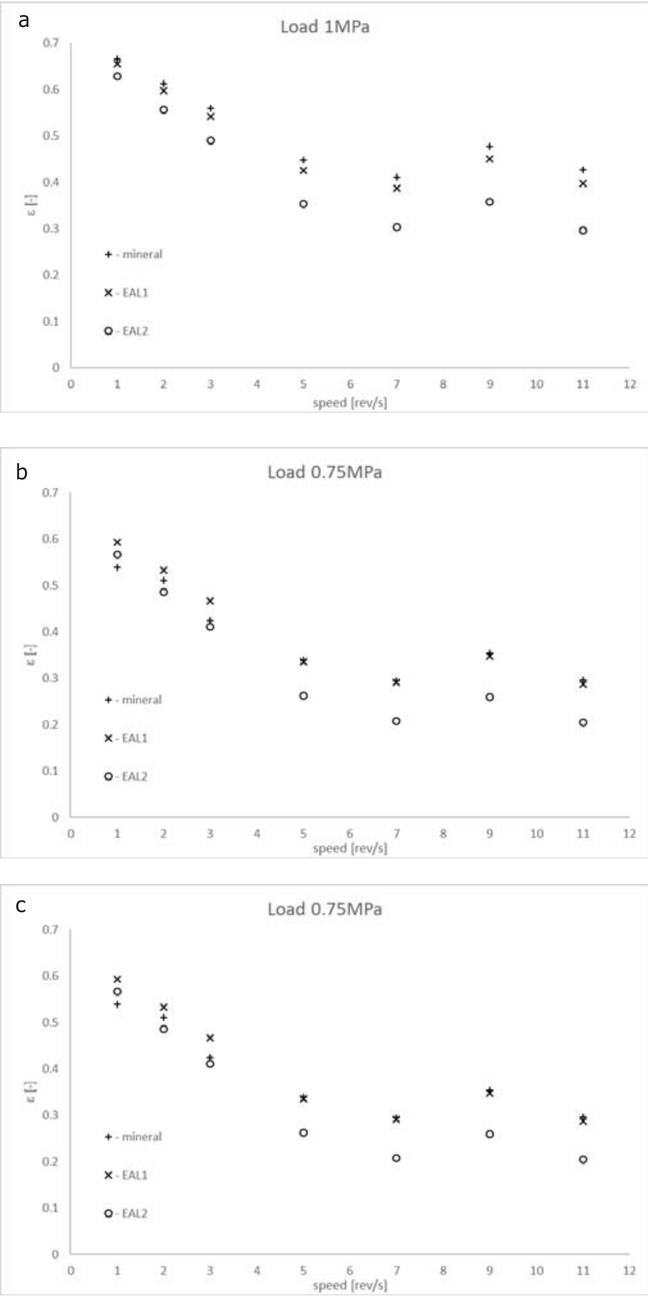


Fig. 6. Eccentricity for different bearing loads and speeds

In order to verify the performance of the bearing lubricated with an EAL after its wear, the performance of the bearing with increased clearance radius was measured. As previously mentioned, wear was simulated by reducing the shaft diameter by 0.1 mm. Figure 7 presents the wear effect on the friction coefficient. An increase of 0.1 mm in diameter clearance does not cause significant changes in friction. However, changes in the friction coefficient not exceeding 0.002 can be observed for low loads and speeds. Despite slight changes in the friction coefficient, an increase in clearance radius changes the pressure distribution in the oil film. Figure 8(a) shows an example of the pressure measurement results for EAL 1 with a load of 1 MPa and a speed of 11 rev/s. An increase in clearance radius increases the maximum hydrodynamic pressure and decreases the pressure in the extreme areas of the oil film. This means that although the load remains constant, the maximum unit pressures increase with increasing bearing wear. The use of EAL instead of mineral oil does not cause any changes to the pressure distribution in the bearing even after it has worn out. It can be observed in Figs. 8(b)-(d) that the increase in maximum pressure is more pronounced at high loads and low speeds. The observed maximum pressure difference when increasing the diameter clearance by 0.1 mm reaches 0.7 MPa regardless of the oil used.

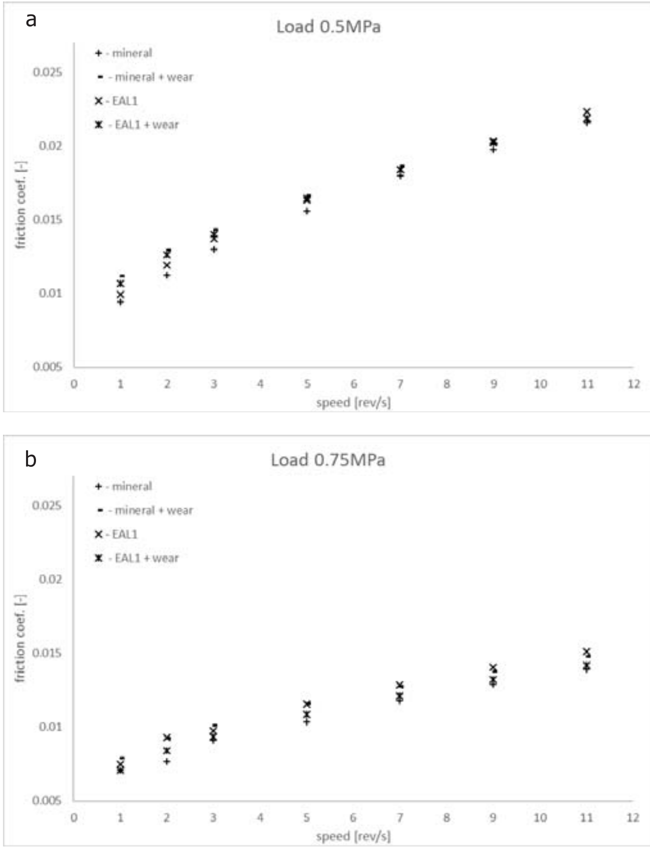
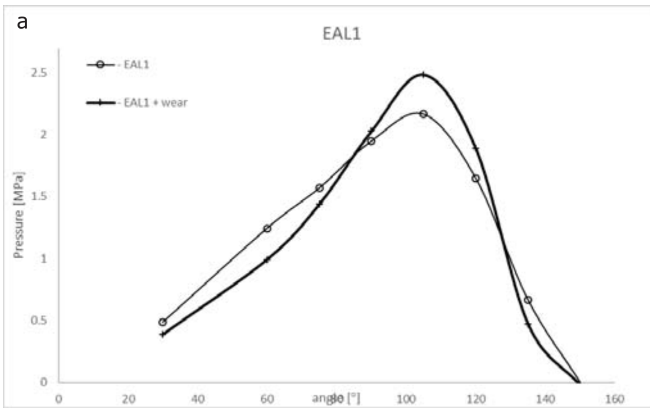


Fig. 7. Friction coefficient for different diameter clearances of bearing



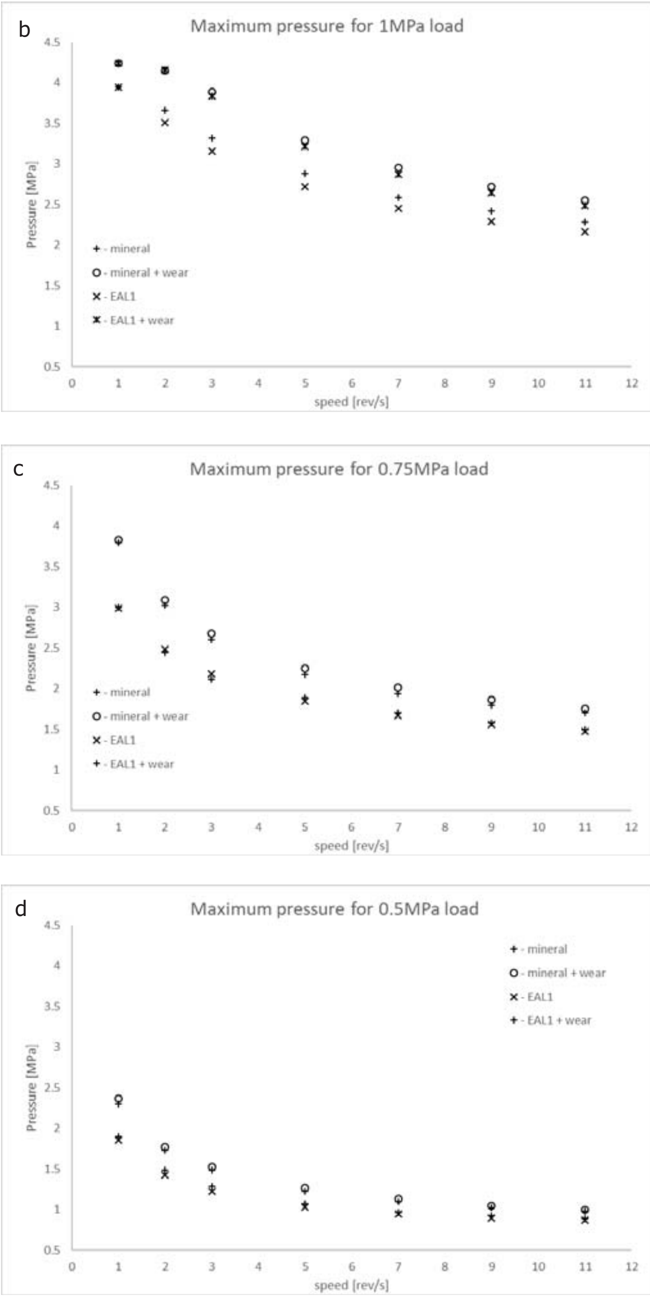


Fig. 8. Pressures in the oil film for different diameter clearances of bearing; a – example of pressure values for EAL1, b-d - maximum pressure for different bearing loads and speeds

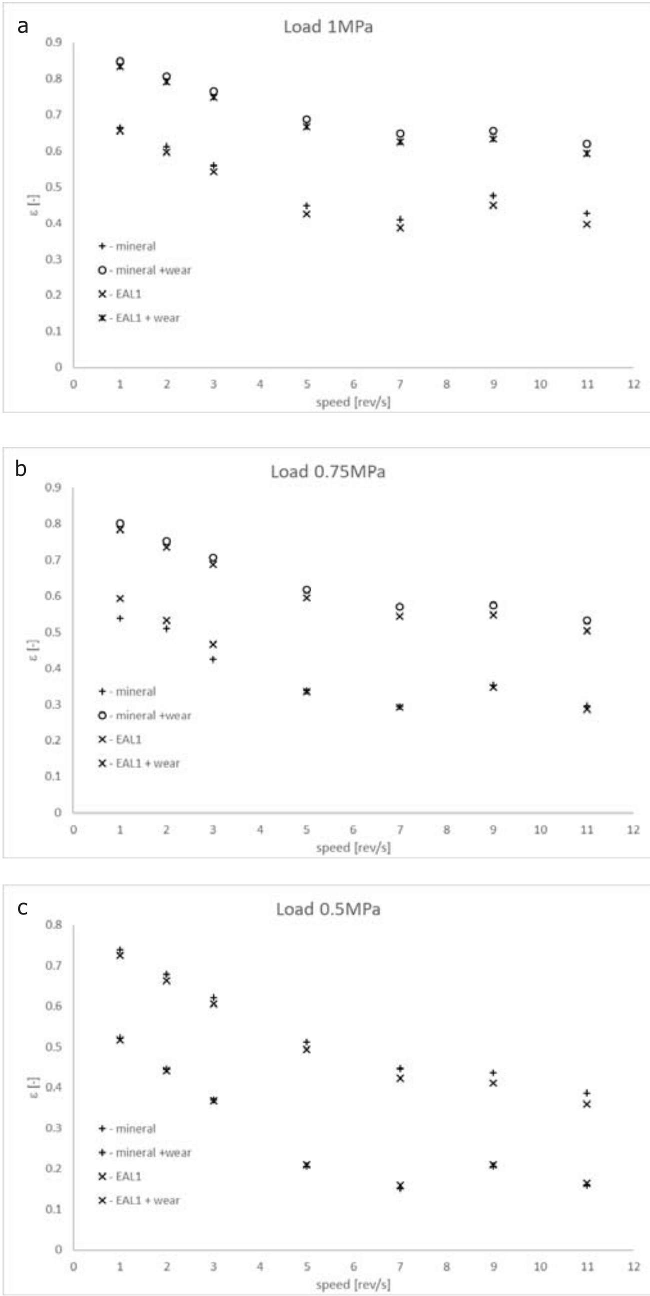


Fig. 9. Eccentricity for different diameter clearances of bearing

Figure 9 presents the influence of bearing wear on its eccentricity. According to the results presented, the eccentricity value increases with increasing wear, with the differences increasing with decreasing load and not depending on speed. Changing mineral oil to EAL of the same viscosity grade does not affect the eccentricity of the bearing even after it has worn out.

CONCLUSIONS

This study compared the performance of a plain bearing with a length/diameter ratio of two lubricated with mineral oil or an EAL. The oil film pressure distribution, friction coefficient and instantaneous shaft axis position were measured. According to the results presented, the following general conclusions can be drawn:

– Changing mineral oil to an EAL of the same viscosity grade in the stern tube bearing does not affect its performance. Both the friction force and pressure distribution in the oil film and the average position of the shaft axis in relation to the pan axis do not differ by values greater than the measurement error of the measuring system. This means that the oil composition does not have a significant influence on the bearing operation, so an EAL of the same viscosity can replace mineral oil without the need to make any structural changes to the bearing. The influence of the EAL on the wear and aging of seals and the aging phenomenon of the EAL itself was not addressed in this article.

– The bearing operating parameters do not change during operation with increased clearance radius resulting, e.g., from wear.
– The conclusions presented are valid for the replacement of mineral oil by an EAL with not only a similar viscosity but also a similar viscosity index. This condition does not have to be fulfilled when the bearing operates in a small temperature range.

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CONTACT WITH THE AUTHOR

Jerzy Kowalski
e-mail: jerzy.kowalski@pg.edu.pl

Gdańsk University of Technology,
Narutowicza 11/12, 80-233 Gdańsk,
Poland