Comparative wear test of journal sliding bearings with sintered bronze and Babbitt alloy bushes lubricated by environmentally acceptable/adapted lubricants (EAL)

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Abstract:

A growing awareness of the negative effects of mineral oils on the natural environment has resulted in the introduction of new regulations related to environmental protection. One of these regulations requires the use of environmentally acceptable/adapted lubricants (EAL) to lubricate marine main shaft bearings, in place of the mineral lubricating oils that have been used for decades.

Classification Societies, which supervise the technical condition of ships, record a certain number of failures each year of heavy loaded stern tube bearings lubricated with modern, environmentally friendly lubricants. The reason often given for such failures is that the operating parameters of EAL lubricants are worse than those of mineral oils.

The aim of the experimental research reported here was to compare the operating parameters and wear of EAL lubricated journal sliding bearings. For this purpose, two classic mineral lubricating oils and a group of four alternative lubricants from the EAL group were selected. The tests carried out for two types of bearing bushes – sintered bronze and Babbitt – did not show significant differences in terms of operating parameters or journal and bush wear for the entire group of tested lubricants.

Key words: wear, friction, journal bearings, safety

Introduction

Increasing awareness of the progressive degradation of the natural environment, including the seas and oceans, has led the international organizations responsible for shipping to introduce specific restrictions. An example is the ban on the use of mineral oil to lubricate ships' propeller shaft shells (1). The best solutions available on the market today, which are well known and have been tested many times all over the world, are water-lubricated bearings with a non-metal shell. Apart from their undoubted advantages, they have some limitations, just like any other solution in engineering. The best water-lubricated bearing bushings available have almost perfect properties (2,3,4,5).

When these water-lubricated bearings are properly designed, manufactured and installed, they work in the area of fluid friction, the wear of the journal and the bushing is low, the resistance to movement is low, and they are sufficiently resistant to operation under lubricating conditions with liquids containing solid particles. (6,7). However, this is not always the case: poor quality of the bush material, improper geometry of the bearing, or careless installation can cause an almost immediate failure (8). Therefore, over the decades, more complicated and expensive oil-lubricated bearings with white metal bushings have gained a large following. With this solution, however, the problem is oil leakage, which has a negative impact on the natural environment. It is also surprising how much oil is in the lubrication system of the shaft shell bearings – there can be hundreds of litres. Based on the data from Classification Societies, oil leaks from the stern tubes of all ships globally may amount to anywhere between 4.6 and 28.6 million litres per year (9). Therefore, the restrictions described above were introduced. In response, lubricant suppliers began to offer their customers replacement oils approved by Classification Societies for use in shipbuilding that are based on esters or glycols, which are known as environmentally acceptable or adapted lubricants (EAL for short).

After several years of using EAL in place of mineral oils, the causes of some failures began to be sought in the slightly different physical properties of these lubricants (10,11). According to some researchers, the problem lies in the different viscosity characteristics of EAL compared to mineral oils, as a function of temperature and pressure, the faster than usual degradation of seals, which causes water to penetrate the lubricant, and ultimately the faster aging process of EAL, resulting in a change in properties.

In conclusion, both types of bearing – water-lubricated and oil-lubricated – have advantages and disadvantages. In practice, probably due to tradition or experience, both these very different solutions have their supporters and their opponents. It is also difficult to define clearly which of the solutions is the best.

Recent tests of the properties of a propeller shaft bearing with a journal diameter of 100 mm did not confirm the theory about the different properties of the lubricants for a bearing lubricated with EAL-type oils compared to a bearing lubricated with a mineral oil of similar viscosity (12). As part of this research, four certified EAL oils available on the market were tested, and no differences in the measurements of resistance to motion and thickness of the lubricating film were found.

Not many papers can be found in the literature connected with EAL. A number of them go deeply into problems of chemical content (13,14) or chemical or physical properties (13,15,16). Experimental work by researchers around the world has been conducted on tribometers (17,18,19,20), and the results of real scale journal bearings tests are not common (21,22). It is worth adding that, in some cases, replacing mineral oil with EAL leads to problems with rubber seals (23,24).

When planning the series of costly durability tests of bearings with cast white metal bushings, preliminary tests were carried out to assess the influence of the lubricant on the wear process of two different bearings. After reading the literature and repeatedly discussing this topic with industry representatives, it was expected that, for bearings lubricated with oils of the EAL type, the wear process would be more intensive than in the case of bearings lubricated with the classic mineral oil commonly used to lubricate a ship's shaft sleeve bearings. The results of the measurements were surprising, and it was considered that the tests should be performed again for SAE30 grade mineral reference oil to make the results more plausible. Therefore, below are the results of measurements of tests carried out for two bearings lubricated with mineral oils with identical catalogue parameters and four oils of the EAL type.

Materials and methods

When planning the experimental work, it was decided that the tests should be carried out in conditions similar to those prevailing in a classic oil-lubricated bearing, and therefore a journal bearing was chosen. The experimental work conducted at the University of Aachen (25) inspired this research. When designing the test rig for this research, it was found that, because the procedure to test durability and reliability was time-consuming, it was advisable to test several bearings at the same time. In the final design, which has been operating reliably for several years, it is possible to test two bearings with a different bushing design simultaneously (Fig. 1).

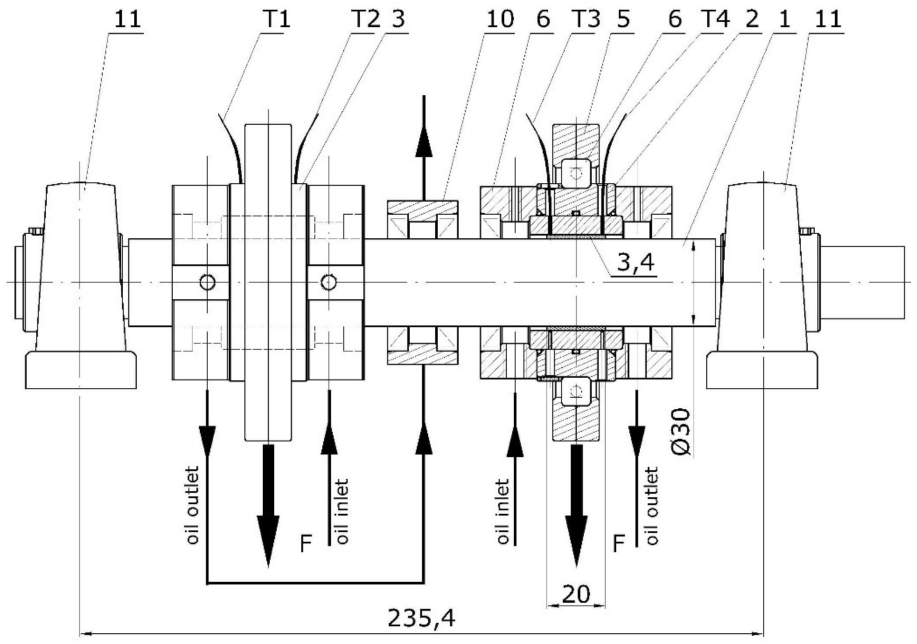

Fig. 1. Main section of test rig for wear test of the two sliding bearings; 1 – shaft, 2 – tested bearing bush housing, 3,4 – tested bearing bush with sleeve, 5 – load ring with ball bearing, 6 – covers with sealings, 10 – two seals module, 11 – self-alignment ball bearing, t – thermocouples

The lubricant is pumped to each of the bearings separately, so it is possible to carry out tests for two different oils simultaneously. The oil flow is carried out by a dosing pump (peristaltic) with flexible tubes that are easy to replace when the oil is changed, so there is no need for time-consuming cleaning of the entire installation – only the tank and the tested bearings need to be cleaned.

Two types of typical, mass-produced bearing bushes were selected for the tests (Table 1). The first is a popular sintered bronze type (Fig. 2). The second is a multi-layered thin-walled bush with a Babbitt sliding layer, such as is often used for a compressor's crankshaft bearings (Fig. 3).

A slightly larger than usual bearing clearance was applied (reducing the diameter of the shaft journal), with the expectation that this would mean imperfect hydrodynamic properties, favouring work in the area of mixed friction; more intensive wear was therefore expected, especially in the starting and stopping phases.

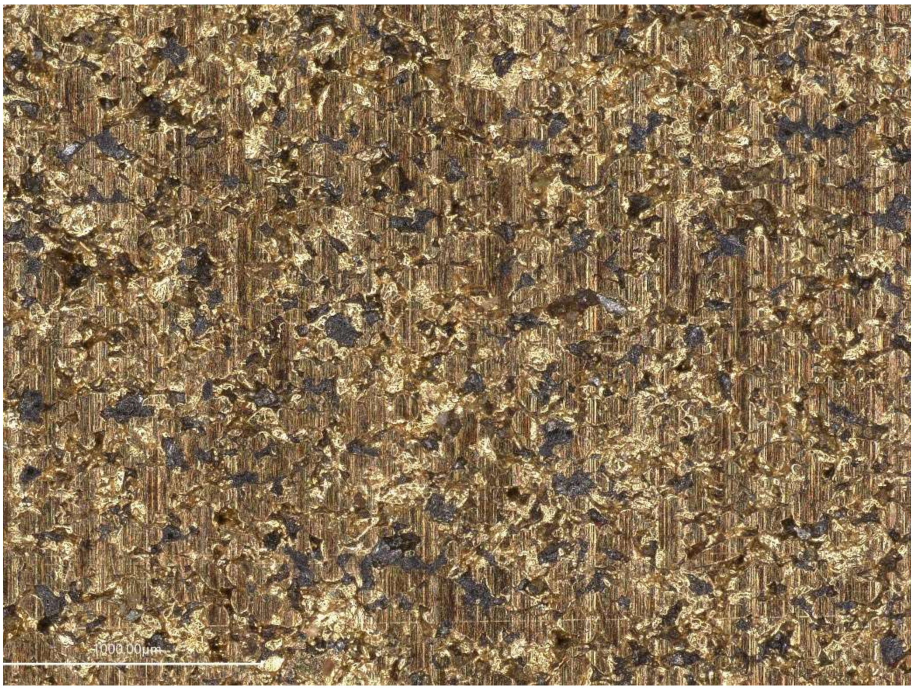

Fig. 2. Sliding surface of the sintered bronze bush

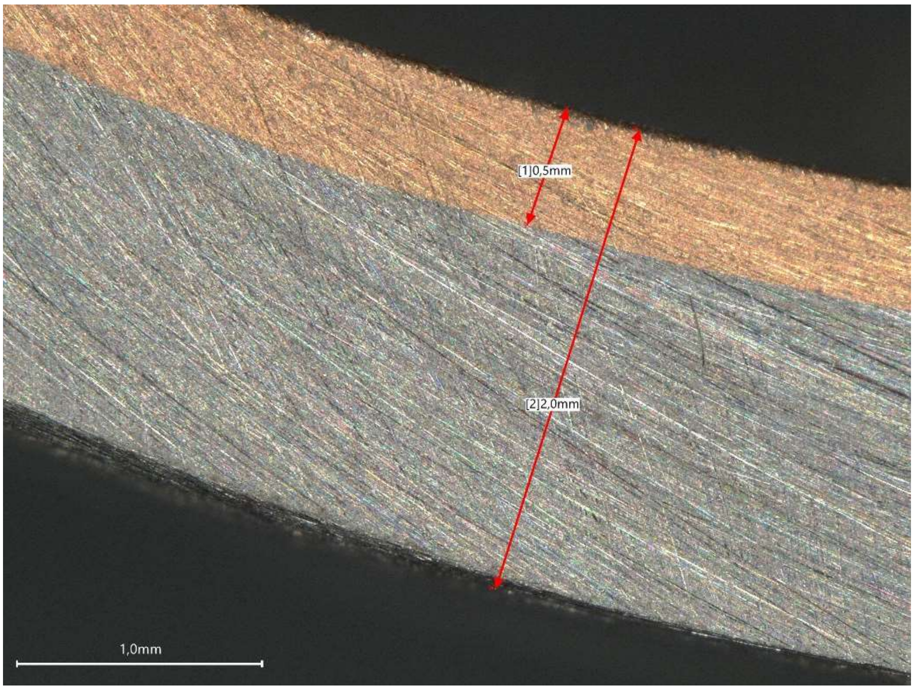

Fig. 3. Cross cut across the multi-layer bush with Babbitt sliding surface

Table 1. Data for the bearings, operational conditions and details of measurement sensors

1	Diameter of the shaft / length of the bushing	30 mm / 20 mm	
2	Shaft material	Carbon steel AISI 1045	
3	Bearing bush material	Sintered bronze Porosity 20% ASTM B438 – bearing bronze	Lead Babbitt PbSn10Cu2 thickness 0.1 mm, sintered bronze 0.4 mm, steel backing 1.5 mm
4	Bearing clearance measured at a temperature of 20°	0.20 mm	0.20 mm
5	Rotational speed of the shaft	0 – 1000 rpm – 15s long cycle, 32h of operation, 7680 start–stop cycles	
6	Radial load [N] / specific pressure [MPa]	1200N / 2MPa	
7.1.	M1 – Reference lubricating oil – mineral, typically used for the lubrication of low-speed internal combustion engines and sliding bearings of the shaft line, -produced by several reputable manufacturers, VG100 (SAE30) (often described as type 3005). Viscosity at 40°C 105 cSt		
7.2.	M2 – Second reference oil – same viscosity properties as M1 but supplied by a different company		

7.3.	EAL(1) Environmentally Acceptable Lubricant – a non-emulsifying synthetic fluid specifically developed for use in stern tubes. Manufactured from fully saturated esters. Viscosity at 40°C 100 cSt, Viscosity Index 137

In order to perform an intensive test of the durability of the sliding couple in the shortest possible time, it was decided to simulate operating conditions that are not sufficient for full fluid friction, that is, frequent and successive starts and stops when the shaft and bushing are in contact with each other cyclically, accompanied by wear. The scheduled test of each pair of bearings was carried out over four days and for eight hours each day. The AC controller controlling the speed of the motor of the test stand was programmed in such a way that during each minute the shaft of the test stand stopped four times and was then accelerated to a speed of 1000 rpm. As a result, after four days of operation (32 hours in total), the shaft of the test rig had performed 7,680 start–stop cycles. Each of the bearings was loaded with a gravitationally applied load causing an average specific pressure of 2 MPa. During the long-term operation of the stand, the acquired measurements were the instantaneous speed of the shaft and the friction force, which were converted into the coefficient of friction (COF), the temperature of the bushing, and the oil temperature at the inlet and outlet of the bearing. The purpose of these measurements was to determine the time needed to stabilize the operating conditions and to control the operating condition of both bearings (Fig. 4).

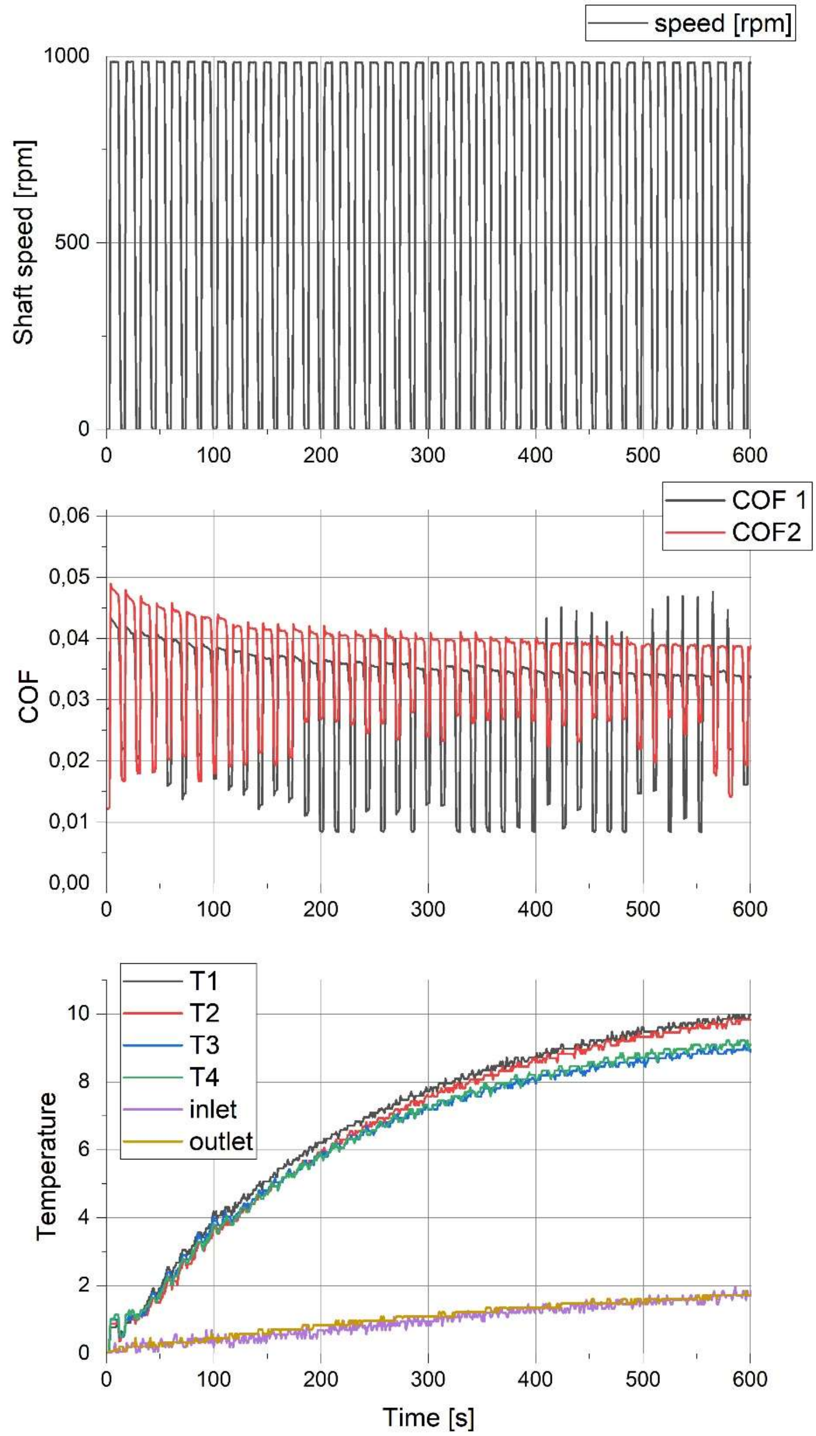

Fig. 4. Measured parameters during tests, early stage of the test after cold start until the test rig had warmed up; shaft speed, COF, and temperature diagrams

Keeping in mind that during the start-up and stopping of the test rig, the values of the friction are in a fluid, rapidly changing phase, and allowing for the assessment of the value of the static friction coefficient and the value during operation in the area of mixed friction, the data acquisition system was run at high sampling speed every few hours so as to be able to measure the friction force during the start-up and stopping of the test rig (Fig. 5).

While the start-up speed was programmed and repeatedly realized by the AC controller, the shaft of the test rig stopped because of the resistance to movement in the bearings and seals. It should be added that the measurement of the resistance of motion in the bearings was burdened with some error. In fact, the friction force of the entire bearing unit, that is, the bushing and the two sealing rings at the same time, was measured. During the design of the stand, this inconvenience was acceptable, because the expected result was the wear measurements of the tested sliding couple, which were taken after disassembly. However, when carrying out the first tests, the functionality of the stand was improved, making it possible to measure resistance to motion and calculate the COF. Therefore, the stand was expanded by installing a module on the shaft between the two tested bearings, which was equipped with two sealing rings (Fig. 1 part no. 10). As a result of this, it was possible to measure the resistance of motion and to calculate the frictional moment of the pair of seals. This allowed a more accurate interpretation of the measured values of the friction torque of each bearing to be made (the measured value of the friction force of a single bearing was subtracted from the value measured on the ring with seals) when calculating the value of the coefficient of friction (COF) (Fig. 6).

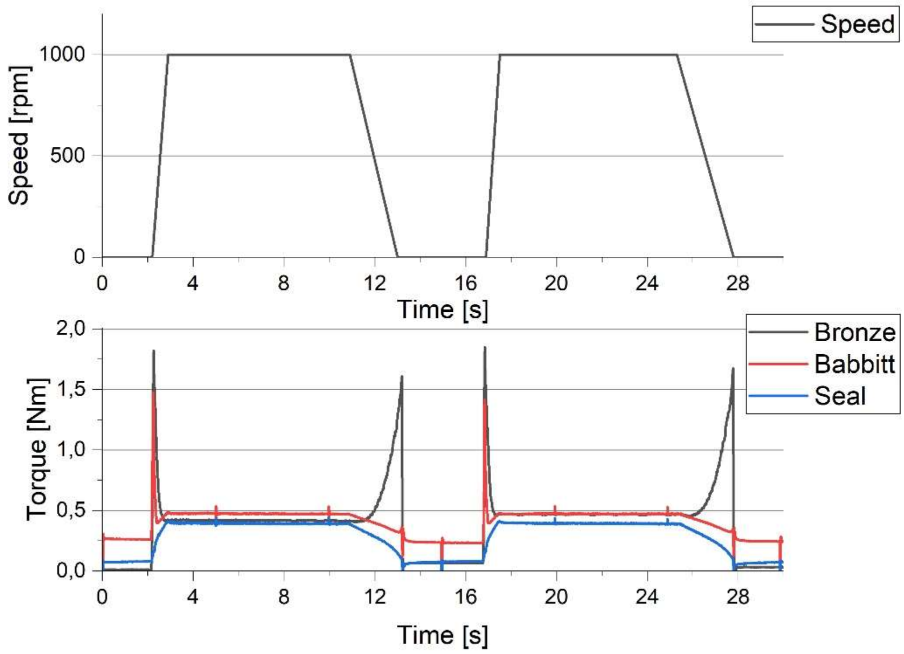

Fig. 5. Measured shaft speed and friction torque of the two tested bearings and the pair of sealing rings

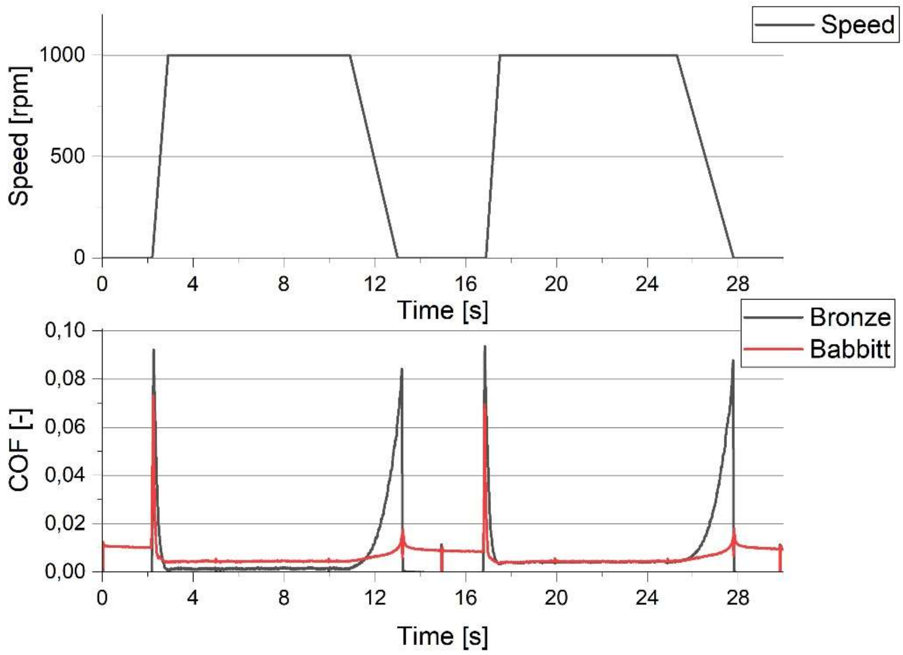

Fig. 6. Measured shaft speed and calculated COF of the two tested bearings during two working cycles

Results and discussion

The tests that were performed allowed a comparison to be made of the results of the operating parameters over time, based on the acquired data and the wear of the sliding pairs, which could be assessed after disassembly of the bearing units. The test results proved that the tested bearings quickly reached a stable operating temperature after about twenty minutes. For oils of the viscosity index VG100, this ranged from 34 to 35°C. For the last of the tested oils, which has a viscosity class of VG150, the temperature reached 39°C, which coincides with the expectation that higher oil viscosity will result in better hydrodynamic parameters, but also greater resistance during operation and thus higher friction losses. The measurements performed for all the tested oils proved that a bearing with a bronze bush works with higher resistance of motion (higher COF) than a bearing with a bush made of a bearing alloy (Babbitt). The operation of the bearing with a sintered bronze bush was accompanied by high resistance of motion during start-up and during shaft run-out (Fig. 6), which proves, among other things, that this bearing has worse hydrodynamic properties. This is due to the poorer surface condition and, in particular, the open pores in the bush structure (Fig. 3), which reduce the value of the hydrodynamic pressure that grows in the lubricating gap. The result of this process is greater wear in the sliding pair.

It is worth to add a comment that measuring very small values of friction during operation under full speed, when bearing operates under full fluid film friction is not simple. In the authors opinion some measurements errors occurs and that is reason why values are not precise enough to analyze them and draw credible conclusions.

The journal wear was determined by measuring the roughness of the shaft journal on a section longer than the width of the bushing (Figs. 7, 8). As a result of this, it was possible to determine the baseline defining the original shape of the shaft surface, and compare it with the outline after the tests. On this basis, the volumetric wear of the journal was calculated. Two pairs of roughness profile charts, prepared for the two types of journal for mineral oil (Fig. 7) and EAL oil (Fig. 8), are presented below.

It is worth noting that the local wear of the journal of the bearing with the Babbitt bush (Figs. 7.b and 8.b) did not exceed 1 µm, which is a very low value after such a hard wearing trial. The bearing journal with a sintered bronze bushing has a shape characteristic of operation with a self-aligning bushing, with the wear at both the edges being greater than the wear in the middle of the bushing length.

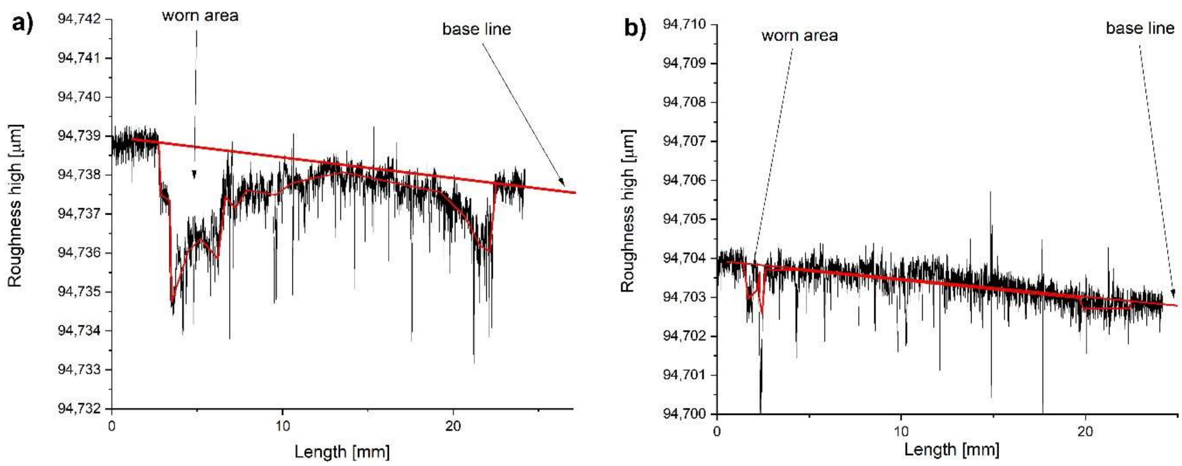

Fig. 7. Roughness profile of the shaft journal after test with bronze (a) and Babbitt (b) bushes lubricated by mineral oil; base line and worn zone marked

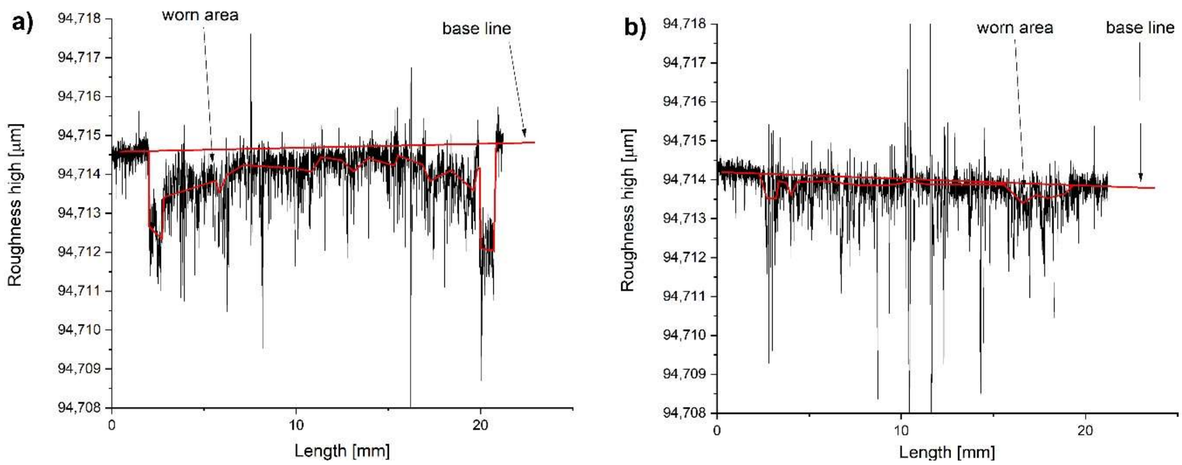

Fig. 8. Roughness profile of the shaft journal after test with bronze (a) and Babbitt (b) bushes lubricated by EAL(3) oil; base line and worn zone marked

The results for the calculated volumetric wear of the shaft journals are presented in the table below (Table 2). The acquired wear values for the sintered bronze bushings are significantly higher than those for the Babbitt bush. The results for the VG150 oil are noteworthy. When starting the research, it was predicted that higher viscosity, at the expense of some friction losses, would increase the thickness of the lubricating film and allow faster transition to the operating range in the area of full fluid friction, so lower wear was expected. It may therefore be surprising that the wear of the white metal bush is higher than the wear for other oils. Analysing the shape of the roughness profile, it can be concluded that the thicker lubricating film caused the self-aligning bushing to become more unstable and, consequently, to wear close to both the edges (Fig. 9). It seems that in a bearing with a fixed relationship between the axes of the shaft and the bushing such an effect would not occur and the wear would be lower. The journal of a bearing lubricated with mineral oils proves to have a low level of wear that is difficult to interpret. There are shallow circumferential scratches on the journal, but also, locally, traces of material transfer from the bush are visible, accompanied by a local, minimal (not exceeding 1 μm) increase in diameter.

Table 2. Volumetric wear of the shaft journals

	Bronze	Babbitt
	[mm^3]	[mm^3]
Mineral M1	1.82	0.03

Mineral		
M2	1.09	0.03
EAL (1)	2.06	-0.38
EAL (2)	2.53	0.14

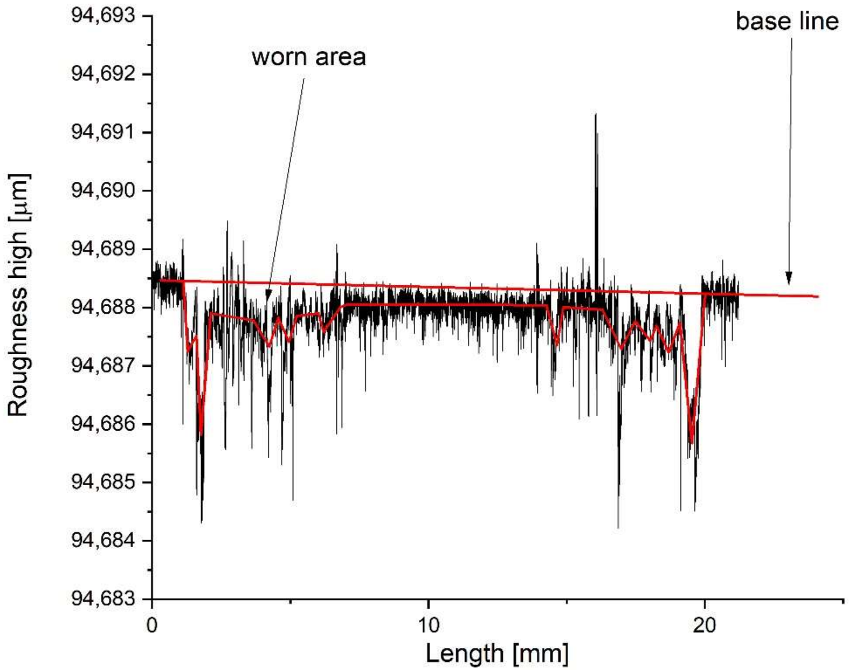

Fig. 9. Roughness profile of shaft journal installed in bearing with Babbitt bush lubricated by VG150 EAL oil

A photograph of the loaded parts of the bearing bushing is shown below (Fig. 10). Traces of wear are visible, but it is difficult to measure the loss of material precisely. The measurements of the bearing bush thickness did not allow a reliable interpretation of the results.

Measurements were carried out using a profilometer, but due to the absence of base surfaces, these only provide information on roughness and not on the amount of wear.

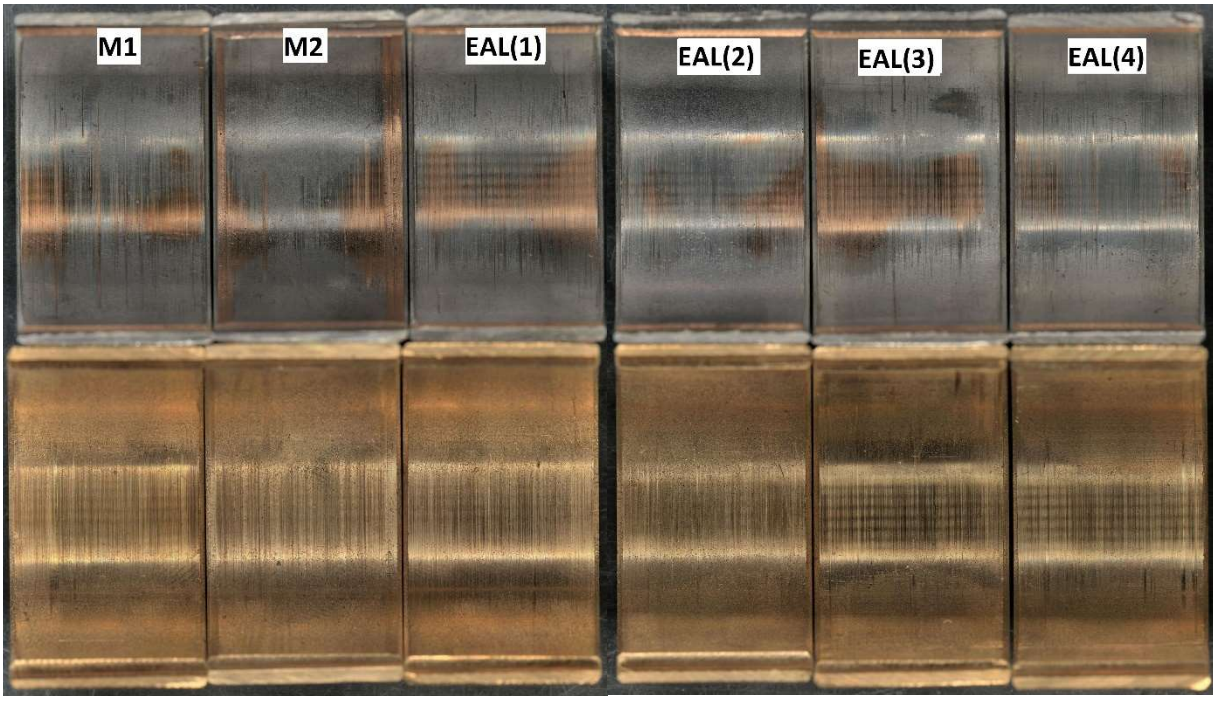

Fig. 10. Bushes after tests; top – bushes with Babbitt sliding layer, bottom – sintered bronze

Another method was therefore proposed that allows for a comparison of the results obtained. This is based on an assessment of the size of the wear area, which can be determined on the basis of a photograph (Fig. 11). It would be ideal if the apparent wear area did not need to be calculated, as the width W can easily be determined (Fig. 10.a). This is the case for bearings with sintered bronze bushings, but not for bushes with a thin Babbitt sliding layer. The resulting dynamically hydrodynamic load-balancing force does not always have the same effect on the self-aligning bushing, and, as a result, we can see greater wear at the edges (Fig. 11. b). A comparative method was therefore proposed, which essentially consists of comparing the surface area of the wear fields and calculating the average width, Wav.

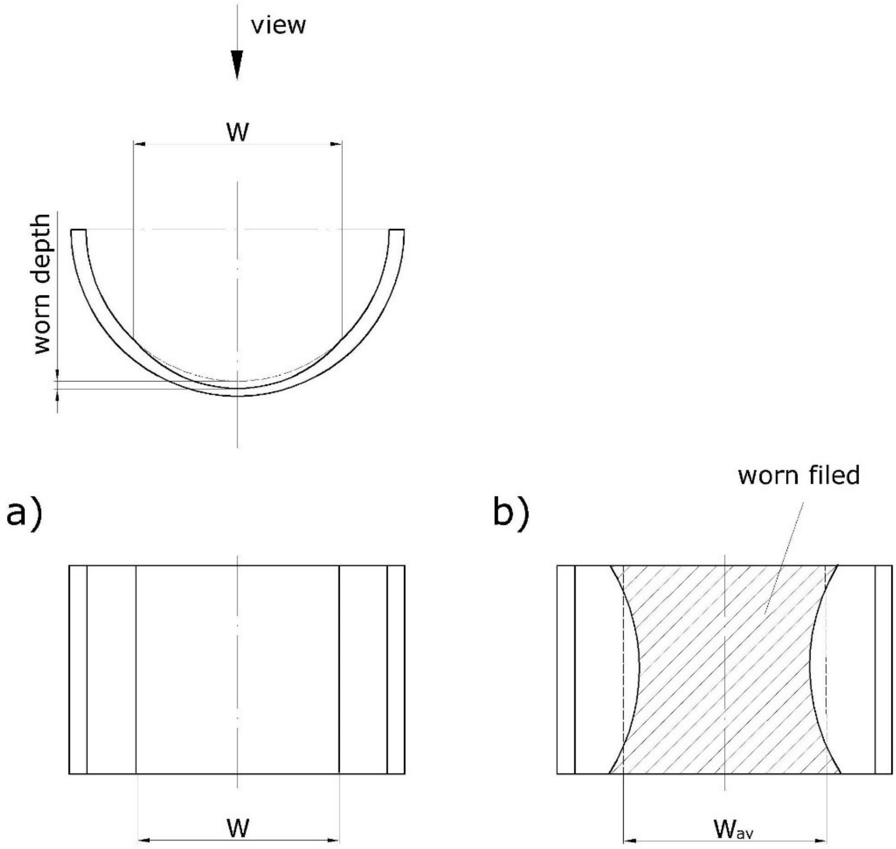

Fig. 11. Worn field width and average worn width on tested bearing bush; perfect location of the worn zone (a) and method for self-aligning bush when worn zone has specific shape(b)

The results of the measurements and calculations of the wear field width are presented in Table 3 below.

Table 3. Average width of worn zone of bearing bushes [mm]

Oil	W_{av} average worn width	
	Bronze	Babbitt
	[mm]	[mm]
M1	12.80	5.21
M2	12.60	5.68

EHL(1)	12.80	5.84
EHL(2)	12.30	3.32
EHL(3)	12.50	7.26
EHL(4)	13.30	3.79

All the results of the wear measurements are presented in the aggregate diagrams below (Figs. 12 and 13). A bearing with a sintered bronze bushing damaged the shaft end much more than a white metal bearing. This is certainly because of the worse hydrodynamic properties and the greater bush hardness compared to the Babbitt bush. The worn depth and width (Fig. 11 – W) of the sintered bush is practically the same for each of the oils used. The bearing with the Babbitt bush showed significantly less wear on both the bush and the journal side. Interestingly, in the bearings lubricated with mineral oils, there was practically no wear of the shaft. The surface roughness deteriorated slightly (from Ra = 0.3 to Ra = 0.64), but it was practically impossible to determine the volumetric wear. In the case of one of the bearings (EAL (1)), the material of the bushing was transferred to the shaft journal. The results of the wear of the bearing bushings lubricated with the higher viscosity EAL (4) oil seem to be burdened with a certain error resulting from the self-alignment of the bearing, which was mentioned earlier.

It is also worth noting that the plain bearing bush with a thin sliding layer (8 ÷ 10 µm) was practically completely worn out, because the substrate was visible.

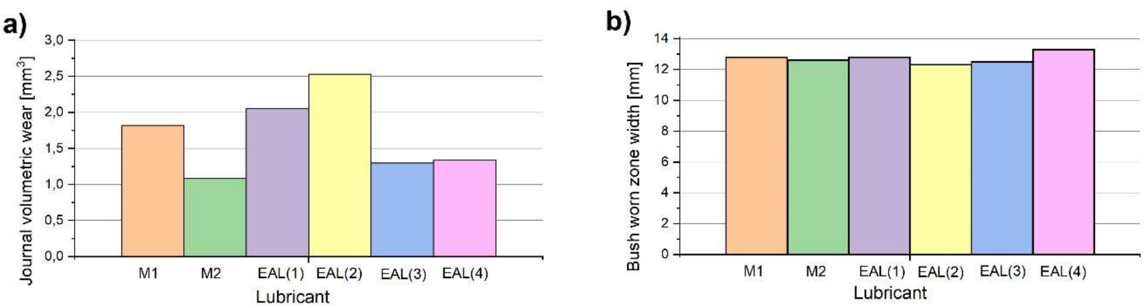

Fig. 12. Graphs of the wear in the bronze–steel sliding couple lubricated by various oils; a) wear of the journal, b) wear of the bearing bush

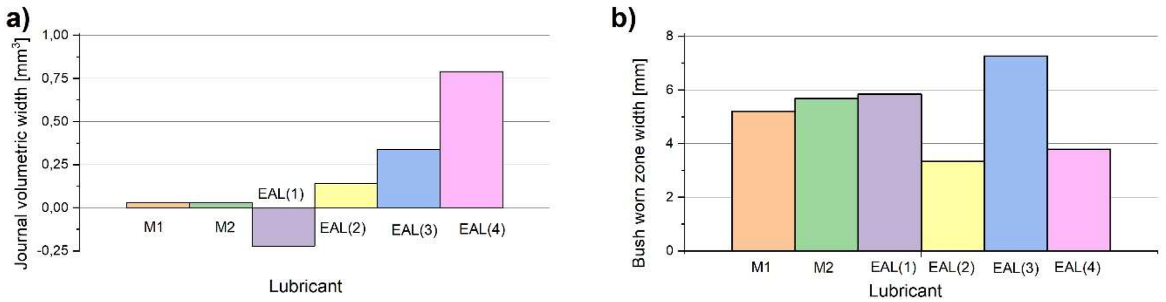

Fig. 13. Graphs of the wear in the Babbitt–steel sliding couple lubricated by various oils; a) wear of the journal, b) wear of the bearing bush

Conclusions

There is no doubt that it is necessary to counteract the ongoing degradation of the natural environment, and therefore replacing oils of mineral origin with oils of other types that are subject to natural decomposition is a step in the right direction. Forcing the use of oils from the EAL group for the lubrication of propeller shaft bearings is a solution that is beneficial for the environment. Unfortunately, however, this solution is not cheap, and various cheaper solutions are therefore sought: for example, bearings with non-metallic water-lubricated bushes are increasingly often used.

Reports of cases in which the crew replaced EAL oil with mineral oil just after an inspection seem unlikely, and would indicate that the regulations were not respected and that the crew was not aware of the environmental concerns.

The conducted tests did not show any significant differences in the properties and durability of bearings lubricated with EAL oils and those lubricated with two mineral oils of the same class. When considering the possible causes of the occasional bearing failures reported by the

Classification Societies, it is difficult to blame EAL oils. According to the authors of this article, these failures could have had different causes, although it is probably easy to blame the new and more expensive oil than the classic solution. It is worth paying attention to the fact that the key parameter responsible for the properties of a hydrodynamic bearing with a specific geometry is the viscosity of the lubricating oil. In the analysed case, the temperature of the bearing bush at the place where the load is highest did not exceed 40°C , which is the temperature at which the viscosity class of the oil is determined. During the tests, the bearings were lubricated with oil circulating in a closed circuit, without a filter. However, the reservoir was so large that natural gravitational sedimentation of pollutants was possible. Nevertheless, the bushings and journals had scratches that are characteristic of lubrication with a liquid containing impurities. The stand has now been rebuilt, with a tank with a funnel bottom and oil filters having been installed. The test results for bearings with bushings made of a bearing alloy are very positive. The wear of the bushing, and especially the shaft end, was small, which in practice is of great importance for the shipowner, as the repair of the shaft usually involves significant costs. Another series of tests will be carried out soon. For these tests, the bushings will be made of the classic white metal alloy used in shipbuilding. It is planned to perform tests for new, used and contaminated oils.

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Authors' Contributions:

Jacek Frost conduct the experiment and analysed and processed the acquired data.

Wojciech Litwin designed the test rig and planned the experiment, and wrote the manuscript. The authors discussed the results and both contributed to the final version of the manuscript.

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