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Influence of Lubrication Water Contamination by Solid Particles of Mineral Origin on Marine Strut Propeller Shafts Bearings of Ships

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ABSTRACT

The stern tube or strat bearings are key important component for safety of shipping. Modern global regulations required a low as possible negative impact on environment including lubricants leaking to the sea. The ship owners are looking for reliable and durable solutions. The costs of each ship components are carefully studied. The water lubricated bearings for ship propeller shafts are an environmentally friendly solution. That is the reason why water lubrication is an interesting option to consider. Because simplicity of the ship design is expected the open lubricating system with only one seal module, when sea water is a lubricant is often recommended by manufacturer.
This research was inspired by a series of failures or premature excessive wear of the propeller shaft strut bearings of real ships. The experimental tests were conducted in the laboratory on custom designed and build test-rig. The large group of tested bearings was delivered by certified manufacturers.
The results of long time wear tests clearly prove that the cause of this premature wear was abrasive wear resulting from water contamination with solid particles of mineral origin. The range of the wear strongly depends on bearing bush materials and bearing interspace geometry. The wear was low for bearings with elastic bushes especially when hydrodynamic phenomena takes place.
The general conclusion is that water lubrication is an interesting option to consider when design of the ship is discussed. The simplicity resulting an attractive price, low maintenance costs and proven durability and reliability of some bearings materials cause that this technology is becoming popular and often applied
.

Keywords: stern tube bearing, green tribology, water lubrication, particles, contamination, energy efficiency

INTRODUCTION

Despite significant progress in engineering at the end of the 20th and the beginning of the 21st century, shipping still plays a key role in global transport. It has been estimated that there are over 100,000 ships in operation around the world, with an average age of over 20 years. The lifespan of a ship is often stated as 30 years, although in practice, this lifespan depends on many factors [1].

An important component of any modern ship is a safe, reliable power transmission system. In the case of a ship equipped with a classic shaft line, the propeller shaft and its bearings are particularly important. Despite over 150 years of experience with ship propeller propulsion, a significant number of failures are recorded every year, resulting in unplanned, costly downtime and repairs to ships [2][3].

Faster ships, such as car and passenger ferries, passenger ships, warships and some container and gas carriers have a twin-screw propulsion system. This solution has been used for over 100 years, most frequently on war ships, but it still gives rise to certain problems. In the past, water-lubricated bearings installed inside the struts were commonly used, and the failure of these propulsion systems caused some shipowners to demand the use of oil-lubricated strut bearings. To make this possible, the shaft support was connected to the hull by a pipe called a stern tube, consisting of a jacket protecting the shaft, filled with circulating lubricating oil. This solution is relatively complicated and sensitive, due to the need for a complex sealing system to ensure the tightness of the lubrication system. These are not 100% environmentally safe solutions, as there is a risk of seal damage and oil leakage into the sea. To extend the durability of the seal, some permanent operational leakage of oil into the water is often allowed (up to two litres per day from a properly functioning system). It is also common that seawater enters the lubricating oil, which may degrade the properties of the lubricant. In the case of some environmentally friendly oils (i.e. the so-called EAL), water ingress may cause the formation of chemical compounds such as acids that can degrade rubber seals, among other components. The desire to reduce the cost of components at the construction and operation stages and to avoid the risk of contamination of lubricating oil from seawater has caused some shipowners to willingly use water-lubricated bearings (Fig. 1b). A cantilever bearing has a relatively simple structure, since it is lubricated by the surrounding water.

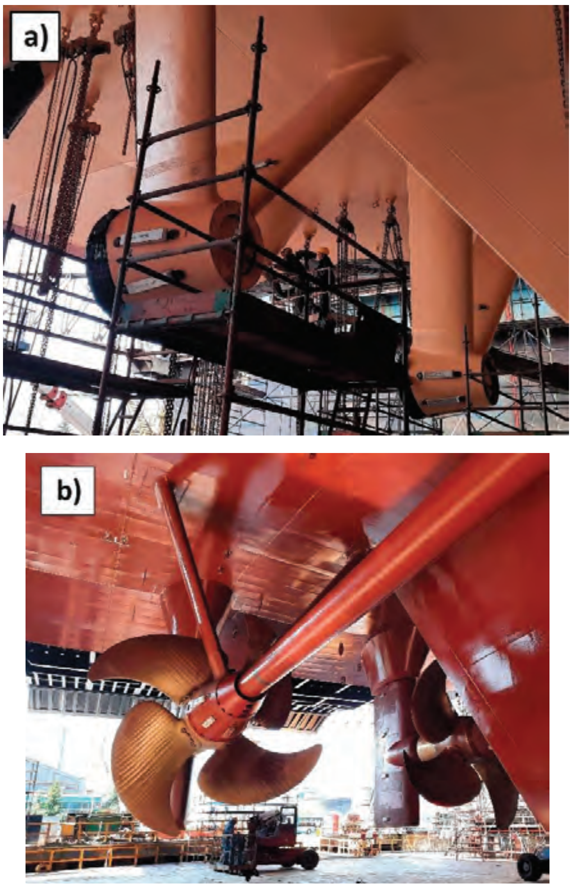

Fig. 1. Strut propeller shaft bearings for passenger and car ferries: (a) a propeller shaft strut before installation of the propeller shaft; (b) two shafts with propellers after final assembly

However, this solution also has some significant disadvantages. An unprotected propeller shaft is exposed to corrosion, and must be carefully protected. The bearing is lubricated with water, which is not filtered, meaning that it may contain various impurities, including hard solid particles of mineral origin, which are particularly dangerous. This was the case for the shaft shown in Fig. 1. After only two years of operation, the wear of the bushings was so deep that it was necessary to replace the bearings and regenerate the journals of both propeller shafts (Fig. 2).

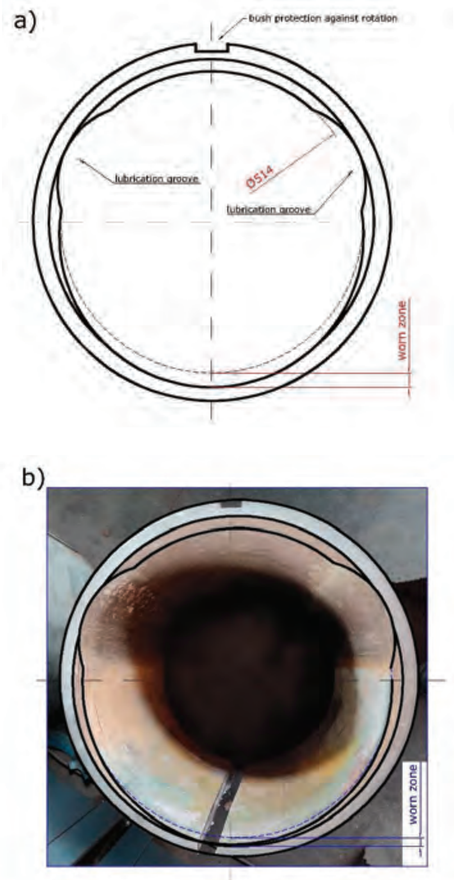

Fig. 2. A ship tail shaft strut bearing bush after dismounting from the passenger and car ferry shown above: (a) simplified drawing of the bushing; (b) photo of a dismantled, worn bushing (the worn area is marked; a slice of the bush was cut and removed by shipyard workers to make disassembly easier)

The problem of lubrication with a fluid containing impurities has been studied by scientists in the past [4]. In recent years, research has focused on some specific types of sliding couples used in practice, including in shipbuilding, and both theoretical and experimental studies using tribometers have been carried out [5][6]. Some of this work has focused on specially prepared journal bearing test stands [7][8][9][10][11][12]. The test results allow conclusions to be drawn regarding the wear resistance of specific sliding couples. In a real object, for example in the case of the ship discussed above, the shaft diameters are significant, and the radial loads resulting from the mass of the rotating shaft and propeller exert average pressures not exceeding 0.6 MPa in typical cases, meaning that despite lubrication with a low-viscosity liquid, it is possible to obtain full hydrodynamic lubrication. Theoretical and experimental research conducted in recent years has confirmed this observation [13][14][15][16].

However, the latest research shows that the loads on the propeller shaft bearing do not result solely from the weight, but also from the propeller, which operates under conditions that are unique to each ship. Traditionally, it was assumed that the average pressure on stern bearings with a length-to-diameter ratio of two (L/D = 2) was less than 0.6 MPa, but the latest research shows that significant radial forces may arise from the operation of the propulsor in a non-uniform pressure field influenced by the shape of the underwater part of the ship [17][18][19]. It turns out that when manoeuvring the ship, the rudder affects the pressure field and hence the forces acting on the shaft and stern bearing [20]. The hydrodynamic phenomena occurring around the stern of the ship in the case of two- and four-screw drives are very complex, and in practice require a separate study for each ship [21]. Experimental research is currently being carried out around the world with the aim of learning about and understanding the impact of variable pressure fields on the forces acting dynamically on a ship's propulsion system [22][23]. It is believed that forces resulting from the hydrodynamic phenomena accompanying the operation of a ship's propeller may be important in unsteady states, especially during manoeuvres. Swimming with a partially submerged propeller is also potentially dangerous, as the propeller wings come out of the water and then hit the water surface [24][25].

Under steady-state conditions, when the speed is constant, the forces resulting from the operation of a large-sized propeller operating under non-uniform pressure (which arise not only from the disturbances caused by the flow of water around the hull but also from the different hydrostatic pressure resulting from the depth) usually make up a percentage of the total thrust force. On a ship, this effect is compensated by a constant rudder turn of about 1–2°.

The phenomenon of hydrodynamic lubrication has a significant impact on the friction and wear process. It has been proven that if solid particles are smaller than the thickness of the lubricant film, they are carried by the fluid through the lubrication gap without damaging the surfaces of the sliding elements, and do not crumble. It is therefore common today to use bushings with lubrication grooves only in the upper part of the bushing, so that they do not have a negative impact on the process of forming the load-bearing lubricant film. However, as operational experience proves, this solution does not guarantee durable and reliable operation of the bearing for the period of 10 years that elapses between two successive required inspections of the ship in dry dock, when disassembly of the propulsion system is usually obligatory. There have been cases where the measured shaft drop, which results from the wear of the bush and journal of the propeller shaft, has forced the need for premature, unplanned, and costly repair.

It is interesting to note that bushings made of synthetic rubber (nitrile rubber, NBR), a solution patented in 1922, are still available on the market and are provided by many manufacturers [26]. This solution is considered obsolete by many engineers, but it turns out that such bearings perform perfectly under contaminated water lubrication conditions, even when they have a geometry that prevents the formation of a load-bearing lubricant film [27]. This solution is very popular in some countries, and is widely used even for large shaft diameters, despite the requirements imposed by classification societies that the sleeve length must correspond to four shaft journal diameters (the proportion of the length L to the diameter D must be equal to four). Concerns may therefore arise about the precision of long bushings; for example, in the case of a shaft with a diameter of 500 mm, the bushing length must be 2 m, and the typical bearing clearance can be calculated using the formula given in the regulations of classification societies as 0.7 mm. For most other materials, such as polymers or composites, the legally required (accepted by the classification society) length-to-diameter ratio is half as large, and amounts to two diameters of the shaft [28].

In order to objectively analyse the advantages and disadvantages of the solutions discussed above, the significant advantages of rubber bushings should be mentioned, such as their ability to dampen vibrations, the lack of stress concentration in the event of misalignment of the shaft and bush axes, the relatively low price of this solution, and the widespread availability of raw material for their production. The aim of the current work is therefore to investigate the wear process of typical sliding couples used in ship propeller shaft strut bearings.

METHODS

This research focused on a group of typical materials prepared for use in shipbuilding (Table 1) with four typical, popular geometries (Fig. 3). It is worth noting that the solution with grooves around the entire circumference (Fig. 3a) is considered to be a classic design, and is still being produced, although attention is now being paid to hydrodynamic phenomena, which is why bushings in which the loaded part has no lubrication grooves at all are becoming increasingly popular (Fig. 3b, c). A compromise can be found in the form of a geometry with wider grooves (Fig. 3d), and it is sometimes possible to find the opinion that a wider arrangement of lubrication grooves favours the flashing out of impurities from the friction zone, and that limiting the sliding surface to 120° of circumference does not limit the hydrodynamic potential of such a solution. All the bushings tested here were manufactured and provided by global manufacturers, and were made extremely carefully. The bearing clearance was selected each in each case in accordance with the manufacturer's recommendations and experience. Each of the materials tested here has also been accepted for use in shipbuilding by at least one of the well-known classification societies, and is in practical use in shipbuilding.

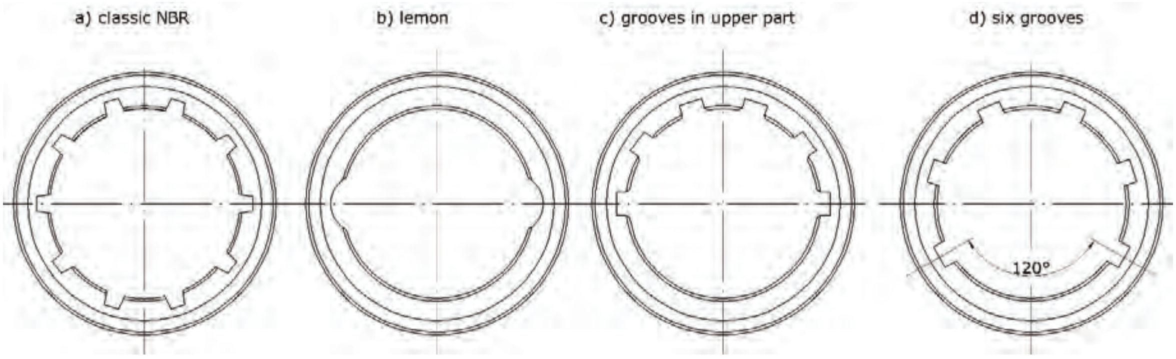

Fig. 3. Four popular bush geometries used in shipbuilding

Table 1. Properties of the tested materials

No./ geometry type/ diameter clearance [mm]	Material type	Modulus of elasticity (radial direction) [MPa]	Thermal expansion coefficient $\cdot 10^{-5}$ [1/K]	Thermal conductivity coefficient [W/K·m]	Water swell [%]	Maximum acceptable temperature of operation /melting point [°C]/[°C]
1/a/ 0.25–0.35	Nitrile rubber (NBR)	110	15–20	0.24	-	-
2/b, c, d/ 0.22–0.32	Nitrile rubber (NBR)	40	17	0.25	-	120/ -
3/b, c, d 0.2–0.3	Elastomeric polymer	35 MPa liner 440–600 shell	Nonlinear below 0°C – 10.2 0<T<30°C: 14.8 Over 30°C: 18.1	0.25	1.3	60/-
4/b, c, d/ 0.25–0.3	Three layers: polytetrafluoethylene (PTFE) / NBR / bronze	PTFE: 770 NBR: 40 Bronze: 103k	PTFE: 12.4 NBR: 17 Bronze: 2.2	PTFE: 0.19 NBR: 0.25 Bronze: 50	PTFE: NBR: Bronze: 0	PTFE: 120/200 (estimated)
5/b, c, d/ 0.25–0.35	Polymer	610	1	-	1.6	150
6/b, c, d/ 0.2–0.25	Composite with fabric	2800	0.9	-	0.5	-
7/a, b, c, d/ 0.25–0.35	Composite with fibres along sliding direction	2300	0.6	0.55	0.2	130/-

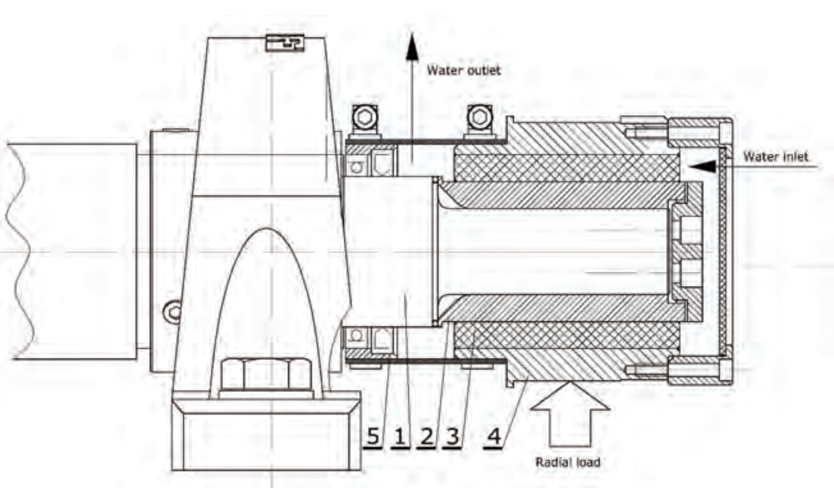

Fig. 4. Key parts of the test rig: tested bearing with seal module; 1 - main shaft, 2 - shaft sleeve, 3 - tested bush, 4 - bush housing, 5 - sealing ring guided by a ball bearing

The experiment was conducted using a specially designed test stand (Fig. 4) with a replaceable stainless steel shaft journal. The bearing assembly was sealed with a specially designed seal module (5). By guiding the sealing ring through the rolling bearing, it was possible to ensure tightness despite the progressive wear of the sliding couple and the gradually increasing displacement of the shaft and bushing axis. The lubricant was water contaminated with solid particles, which was kept in a tank with a conical bottom. To prevent the process of gravity causing sedimentation of the water contaminants, the mixture of water and contaminants was constantly stirred. The test rig enabled us to measure the friction force of the bearing assembly during operation and to measure the temperature of the water flowing through the bearing. The dimensions, the operating parameters of the test rig, and the measurement procedure are summarised in Table 2 below.

Table 2. Data on the components of the main test rig

No.	Parameter	Value
1	Journal diameter x journal length [mm] Shaft liner material Materials, geometries and diameter clearances of bearing bushes – table x	Φ 65 x 105 Steel X3CrNiMo13-4 (1.4313)
2	Specific pressure	0.6 MPa (a typical maximum value for marine applications)
3	Shaft speeds - running-in (10 h in clean water) - test conditions (60 h)	1st stage: 5h at 600 rpm 2nd stage: 5h at 1000 rpm 4 x 15 h at 1000 rpm, four lubricant replacements, total trial time 60 h
4	Lubricating liquid preparation	Pure, clean water 10 dm^3 5 cm^3 of solid particles of mineral origin

A mixture of fresh water with particles of mineral origin typical of the Baltic Sea catchment area was used as a lubricant [29][30]. For this purpose, natural sedimentary material was collected from the river. It was cleaned from biological contaminant like leaves parts. The high quality, modern digital microscope was used for making pictures (Fig. 5) The dedicated software provided by manufacturer was used for analysis of the particles size. After conducting a statistical analysis after many measurements of different samples percentages of particles of various sizes figure with particles size distribution was plotted (Fig. 6).

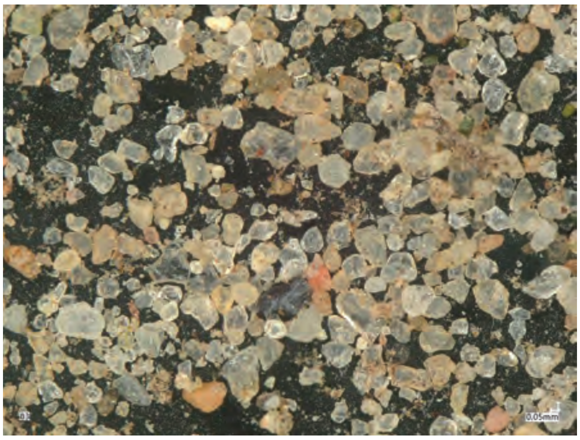

Fig. 5. Microscopic photo of solid particles of mineral origin used during the tests

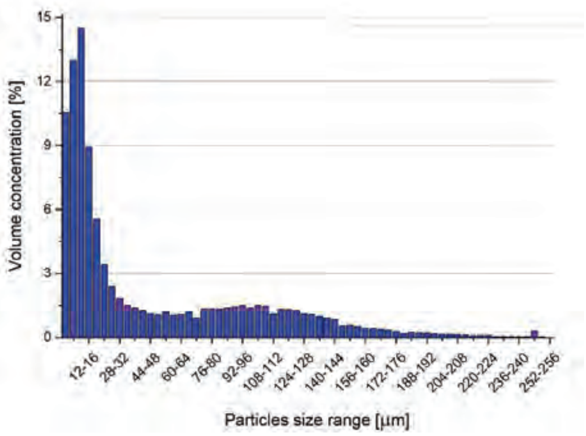

Fig. 6. Percentage size distribution of solid particles used during the tests.

In order to investigate the wear of the sliding pair, it was necessary to measure the wear of the shaft journal and bushings. The shaft journal wear was determined by analysing the measurement results with a profilographometer, which allowed us to calculate the consumption area and then the volume (Fig. 7).

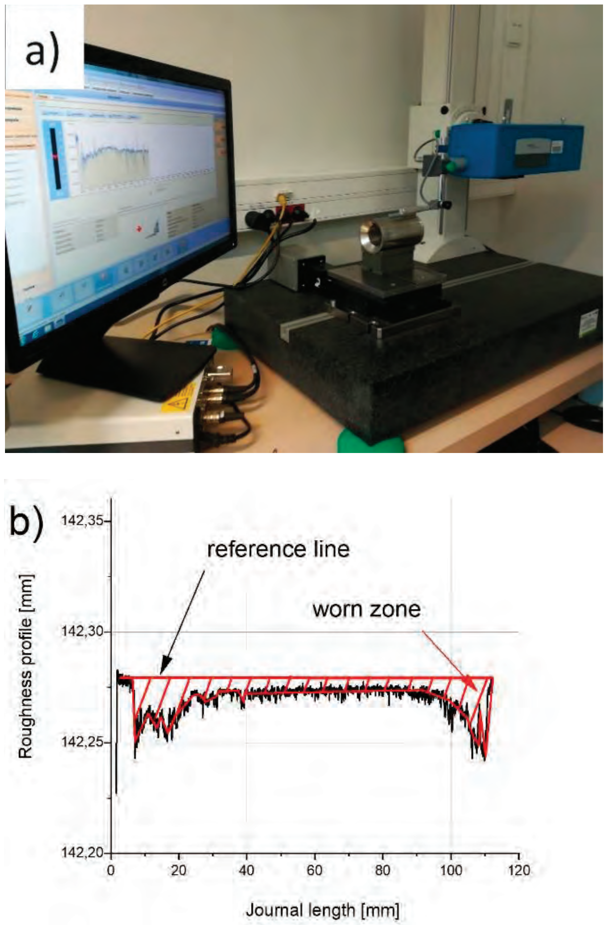

Fig. 7. (a) Profilometer measurements of the shaft journal surface profile; (b) method used to calculate the wear field based on the roughness profile.

The bush wear was determined based on the loss of bush wall thickness measured at three surfaces (Fig. 8), and the amount of consumption was then calculated using the method proposed by Archard [31]. The specific wear rate coefficient (k_i) was determined based on the wear volume (V_i), sliding distance (s) and applied force (F) as

$$k_i = \frac{V_i}{F \cdot s}$$

A well-known formula for the problem studied here involves replacing the wear volume (V_i) by the wear displacement (h_i), in this case the average value of the measurements of the worn depth on both sides of the bearing bush, i.e.:

$$h_i = \frac{V_i}{A}$$

where A is the area subjected to wear, and p is the specific pressure, giving

$$k_i = \frac{h_i}{p \cdot s}$$

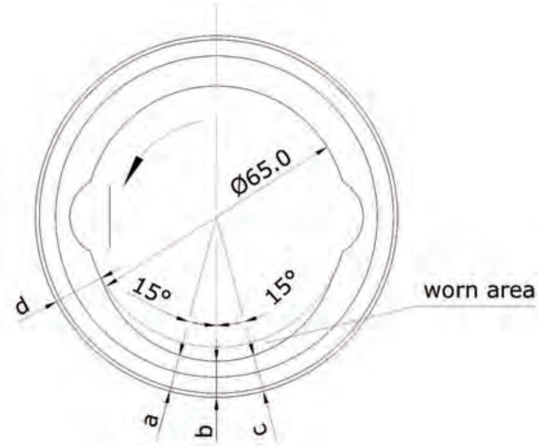

Fig. 8. Measurement of bush wear at three surfaces a, b, c, relative to the original bush thickness d.

RESULTS AND DISCUSSION

The results of testing 27 different sliding couples are presented below. The testing of each bearing took on average two weeks, and the full series of tests took almost a year. For each material pair, we began by testing the couple under clean water lubrication conditions with a typical geometry for a given solution (Fig. 3). During the bearing operation, which lasted 70 h, the friction force was measured and recalculated to give the coefficient of friction (COF) and the temperature of the water flowing through the bearing on the supply and outlet sides. Lubrication with a liquid containing hard solid particles of mineral origin in the case of a metal sliding pair is accompanied by a very intense process of crushing solid particles, causing a temperature rise [10]. In the case of bearings with non-metal, more flexible bushings, the wear process was different; after adding the contaminants, the resistance to movement in the bearing increased, but the particle crushing process was not intensive. The wear process was accompanied by a visible increase in temperature over the entire system, which was due, among other things, to the small volume of the lubricant system, which had a volume of only 10 l. The temperature and COF charts presented below were acquired for a classic, popular solution, a rubber bearing with grooves made around the entire circumference (Figs. 9 and 10), and the low intensity of the wear process is visible. A comparison of the temperature measurement results clearly shows an increase in the temperature of the lubricant flowing through the bearing. In the case of clean water, this increase is about 4°C, whereas for contaminated water it is about 6°C. This is undoubtedly due to the greater resistance to movement in the case of lubrication with a liquid containing solid particles compared to pure water.

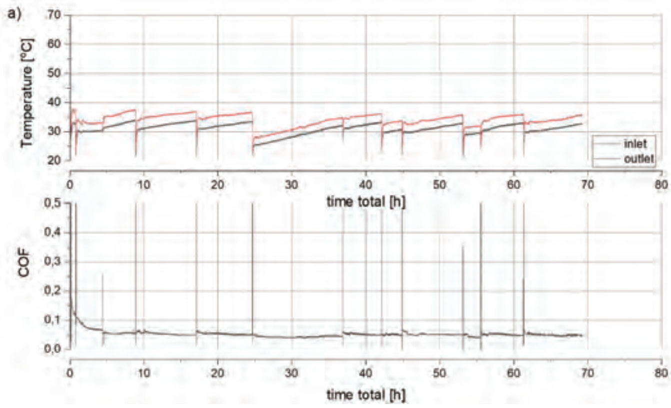

Fig. 9. Temperature of the water at the inlet and outlet sides of the bearing, and coefficient of friction (COF) for a classic bearing with a rubber bush with grooves around the entire circumference, under clean water lubrication (bearing 1a)*

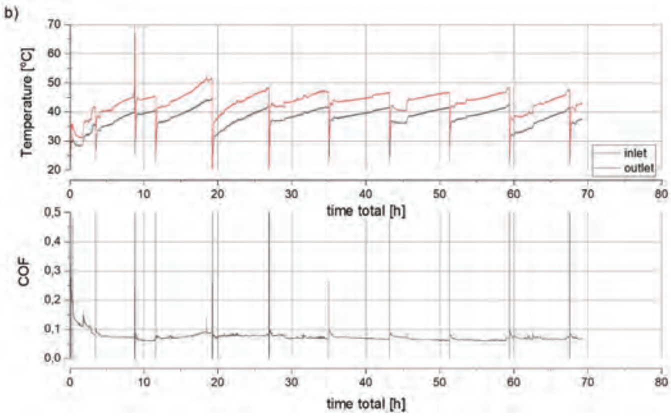

Fig. 10 Temperature of the water at the inlet and outlet sides of the bearing, and coefficient of friction (COF) for a classic bearing with a rubber bush with grooves around the entire circumference, under contaminated water lubrication (bearing 1a)

The reference line showing the maximum acceptable wear level is marked on the volumetric journal wear comparison chart above. This is conventionally used to show the level of wear recorded for the classic solution, i.e. a bearing with a rubber bushing with through lubrication grooves on the

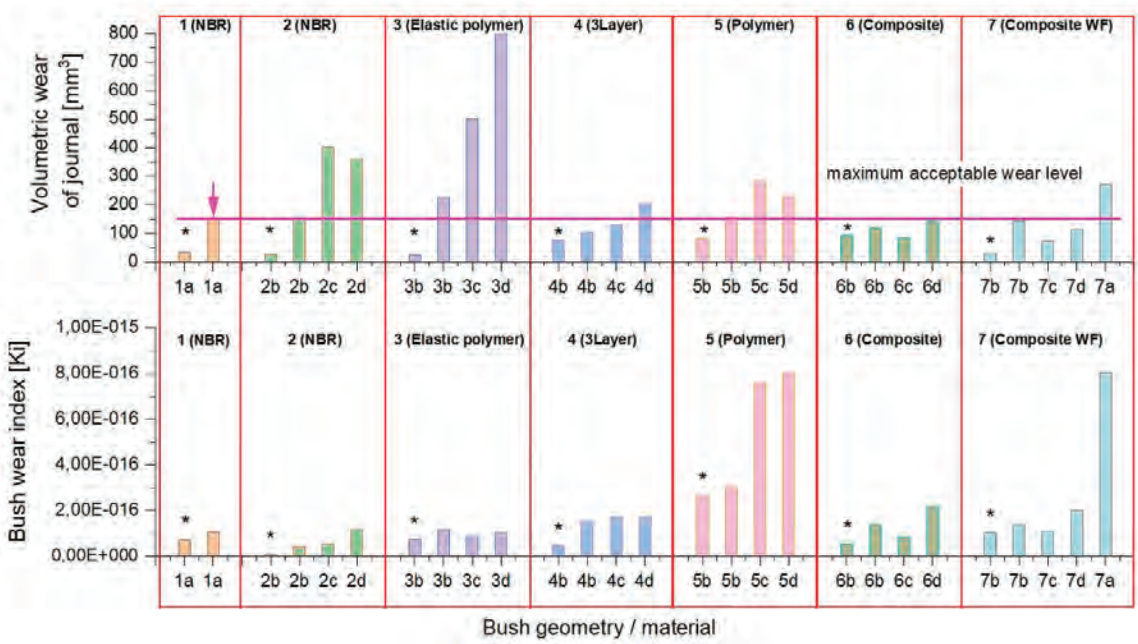

Fig. 11. Summary of wear results for shaft journals and bearing bushes, for varying sliding couples and geometries (indicates tests performed with clean water)*

circumference. It is known from experience that in many countries, a five-year service life for this type of bearing is considered satisfactory; after this period of time, the propeller shaft is dismounted and the bearing journals are regenerated by grinding or replaced with new ones. This is time-consuming and expensive, but it ensures reliable operation over the assumed five-year period. The overall cost of such an operational strategy is acceptable for the shipowner because the rubber bushing itself, especially if it is mounted in a composite sleeve, is not much more expensive than bronze or brass.

From the test results, it can be concluded that bearings with polymer bushes (material 5), which are popularly used in marine propulsion systems, are not the best option for sliding couples in cases where water contaminated with solid particles may be the lubricating fluid. The experimental tests carried out here show that it can be expected that the wear process of the shaft journal and bearing will be intensive. Although the geometries of bushings b and c favour the generation of hydrodynamic phenomena, thanks to the fact that the lubrication grooves are placed in the unloaded part, the lubrication film is often not thick enough to allow large solid particles to pass through the bearing without increasing its wear. The limited hydrodynamic load capacity often results from the relatively large bearing working clearance recommended for materials with a high coefficient of thermal expansion, such as polymers. This increase in clearance is intended to protect the bearing against jamming of the shaft, which results from the thermal expansion of the bushing as the bearing clearance gradually decreases as a result of the increase in volume. Previous research has proven that this may be the cause of sudden failures [32]. Some polymers increase in volume as a result of absorbing water (soaking of a polymer results in swelling), so manufacturers often also recommend precautions in the form of an increased bearing clearance. Due to the low permissible operating temperatures, which can be as much as 60°C, efforts are very often made to provide effective, failure-free cooling; a significant number of lubrication grooves may be placed in the bushing, which can limit the hydrodynamic load capacity or even cause the bearing to work in the region of mixed friction, meaning that the wear of the sliding couple may be very intense (3D bearing).

In the case of chemically cured resin reinforced with fibres (materials 6 and 7), the thermal expansion coefficient is usually lower, and the materials are also usually more resistant to higher temperatures (up to around 130°C). This makes it possible to safely reduce the bearing clearance, and has a positive effect on the hydrodynamic properties of the bearing. After testing the composite materials, an acceptable level of wear of the journal and bush was found for the sliding couple when no lubrication grooves were made in the bush in its lower, loaded part (geometries b and c). The addition of grooves to the lower part of the bushing probably limited the hydrodynamic load capacity of the bearing and, as a result, caused intensive wear of both the journal and the bushing (7a). The opinion, popular among engineers, that particles are washed out by the lubricant flowing through densely spaced grooves was not confirmed. However, one surprising result was the wear resistance of a classic bearing with a rubber bush, with lubrication grooves arranged around the entire circumference (1a). The geometry of this bush is not conducive to the formation of fluid friction, and yet the wear of both the journal and the bush was lower than for the other solutions. According to the author of the work, this is due to the local deformation of the flexible bearing surface.

It was found that the forces exerted on the particle were not large enough to crush it, and that it rolled between the shaft journal and the bushing. The local deformation of the bush is shown in the microscopic photo below of a specially prepared model (Fig. 12). This concept helps to explain why on elastic bearing bushes only large scratches probably made by larger hard particle are easy to notice (fig. 13, 2b and 4b). For harder bushes process of crushing a particles results intensive wear especially when hydrodynamic properties are poor because of lubrication groves located in lower half of the bearing bush (fig. 13, 6d and 7a).

Fig. 12. Contact surface of the steel-NBR pair, with solid particles of mineral origin between them, showing the local deformation of the flexible NBR.

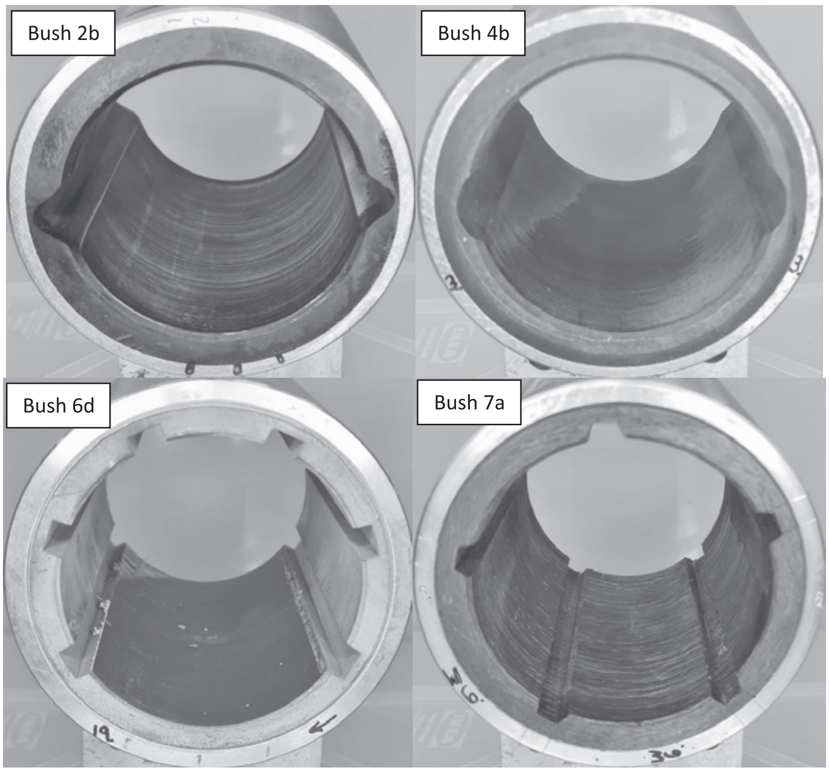

Fig. 13. Selected group o tested bearing bushes; 2b- NBR with scratches on surface, 4b- 3 layer bush with smoothed surface and visible contact zone, 6d – polymer bush with wear zone between grooves, 7a- composite with deep worn zone – visible contact zone (dark)

CONCLUSION

The cooling conditions in the strut bearing of a ship's propeller shaft are excellent, due to the very intense flow of surrounding water caused by the movement of the ship's propeller. However, due to the pressure differences caused by the rotation of the propeller and the flow disturbances caused by numerous factors, the flow rate of water through the sliding pair itself is difficult to determine. Accurate temperature measurements at the sliding surface of a non-metal bearing are practically impossible, making it difficult to monitor the operating conditions of the bearing. Our tests confirmed that the classic NBR material works well in bearings lubricated with surrounding water, where a lack of filtration means that contaminants easily find their way into the lubrication gap. These tests also confirmed that the composites often used for sliding bushings work well as long as the geometry of the lubrication gap favours the formation of a load caring lubricant film. Another interesting aspect is the gradual evolution of the bearing geometry, from grooves evenly spaced around the entire circumference, which was popular until recently, including for composite materials (geometry a), to grooves only in the upper, unloaded part (geometry c), and then to only two grooves in the horizontal plane (geometry b). This progression undoubtedly results from an understanding of the hydrodynamic phenomena that occur in the stern bearing of the propeller shaft. The upward shift of the two lubrication grooves (Fig. 2) is due to the increasing appreciation of the influence of the horizontal force acting on the propeller shaft resulting from the operation of the twin-propeller drive system [33][34][35].

One popular solution for ships with twin-propeller propulsion is to extend the stern tube outside the ship's hull, and to support it with brackets at the end and place sliding bearings inside, closing the entire system with seals. This solution has significant advantages. The lubricant (mineral oil, EAL or water) circulates in a closed system, meaning that the temperature and flow rate of the lubricant can be controlled, and the lubricant itself can be filtered to remove impurities and the products of wear and corrosion.

When assessing the results of research work, the scale effect should be borne in mind. Our tests were carried out on bearings with a journal diameter of 65 mm. In many typical applications of this type of solution, such as in propeller shafts supported in cantilever bearings, for example in the case of popular passenger and car ferries, the shaft diameters exceed 400 mm. This means that the obtained lubricant films will have a proportionally greater thickness, while the sizes of solid particles may be similar to those used during experimental tests. Hence, in reality, the working conditions of a ship will not be as difficult as during laboratory tests. It is also worth noting that in a real system, cooling will also be better due to the inflow of water at a relatively constant temperature, whereas during our laboratory tests, the water temperature in the system slowly increased. An important conclusion from this research is that the classic bearing material, NBR, works well in open system operating conditions when the lubricating water is not filtered. However, it should be remembered that in accordance with the regulations of classification societies, bearings with an NBR bush must have a length corresponding to four shaft journal diameters (L/D=4), while for most modern materials approved for use in shipbuilding, the bush can be half as long (L/D=2). A longer bearing imposes higher costs and technological difficulties at the production and assembly stages. It is also worth considering that modern materials such as a three-layer shell with a PTFE surface (material 4) or modern composites (materials 6 and 7) work very well as long as they have the right geometry and bearing clearance. Our research continues, and the research stand has been rebuilt and improved; future work will focus on the scale effect and the impact of particle size on the wear process in a bearing operating in the regions of fluid friction and mixed friction.

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AUTHORSHIP CONTRIBUTION STATEMENT

Wojciech Litwin: Idea, Inspiration, Methodology, Resources, Investigation, Data curation, Validation, Visualization, Writing – original draft, Writing – review & editing
Agnieszka Barszczewska, Ewa Wojtowicz, Izabela Szwoch, Leszek Matuszewski: Investigation, Data curation, Validation, Writing – original draft

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