Experimental research on water-lubricated marine stern tube bearings in conditions of improper lubrication and cooling causing rapid bush wear

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Abstract

Water-lubricated polymer bushes are increasingly popular in sliding bearings of marine propellers, water turbines, and pump shafts. This environmentally friendly solution is relatively simple and usually proves to be durable.

However, in practical experience one may also observe cases of rapid wear or even malfunctions of this type of bearing, some of which may be caused by insufficient flow of lubricant. The main purpose of the conducted research was to identify how different bearing types operate under conditions of no lubricant flow.

The conducted research determined that certain bearings continue to work properly despite a lack of lubricant flow and cooling. This is due to low frictional resistance levels, resulting in sufficiently low heat being generated in the friction zone of such bearings and dispersed into the surroundings once the bearing's temperature has risen and stabilized at a safe level.

1. Introduction

Water-lubricated polymer bush bearings are finding increasingly wide use in shafts of marine propellers, water turbines, and pumps. This fact stems partially from their simple construction, which also means that this type of solution has a relatively low price ([1], Lagersmit [2]). Three main construction systems of this bearing type have dominated in practical applications.

The first, which is also the simplest, is an open system with the bearing lubricated by surrounding water and freely flowing through it, for example ship stave bearings (Daugherty [3]) and certain water turbine (Nisaka [4]) and water pump bearings ([5], Laskey [6, 7]).

In the second, somewhat more complex system, the surrounding water is pumped, filtered, and then forced through the bearing to the outside. This solution offers significant advantages. First of all, the flow of the lubricating agent through the bearing may be controlled by regulating the work of the pump. In the event of lack of flow caused by pump malfunction or clogging of the filter, a flow sensor should detect and indicate the system failure. Such indication is very important as in most cases it makes it possible to avoid more serious consequences of a potential breakdown. The filtered water is usually free of large-sized impurities – hard particles – thanks to which the risk of rapid bearing damage is limited (You-Qiang [8]). The pump solution is readily used on ships in marine propeller shaft units (Harish [9]).

The third solution is the most advantageous but also the most expensive. It consists of a system with enclosed water circulation. The bearing unit is closed on both sides by seals which effectively protect the friction couple from being polluted by outside impurities. Clean, usually salt-free, filtered freshwater of appropriate temperature is supplied to the bearing unit by a pump. The chemically neutral water, free of impurities and with appropriately selected flow parameters, with ongoing control of potential changes to these parameters on the outflow, provides highly advantageous working conditions for the bearing. Such systems are particularly sought after and therefore frequently used in inland waterway ships, river water turbines, and dewatering pump systems (land reclamation, open-cast excavation, mines, etc.).

Based on extensive practical experience, it may be stated that in most cases water-lubricated bearings have a durability of at least a decade or so. After this period, the bushes are replaced with new ones and the shaft undergoes regeneration, which is usually limited to regrinding its sliding surface. In the case of more extensive surface damage, it may be necessary to apply overlay welding and grinding. Occasionally, other innovative regeneration methods may be applied, such as remachining the shaft and placing a thin-walled stainless steel or bronze sleeve on it (Grudziński [10]).

Thanks to research efforts carried out by R&D departments of major bearing manufacturers and scientific centres all over the world, both the durability and the reliability of this bearing type have improved. The research included important theoretical works aimed at pinpointing the conditions under which fluid film lubrication takes place. It was determined that excessive radial clearance and incorrect location of lubrication grooves may play a particularly strong role in reducing the bearing's hydrodynamic load-carrying capacity (Litwin [11, 12], Wang N. [13], Xijin [14]). In their research, the manufacturers focused on producing a material of low frictional resistance, high durability, and minimum water soaking, resulting in a negligible swelling characteristic (Ford [15], Laskey [6], Odendorff [16], Yamayo [17]).

Unfortunately, as it stems from practical experience, breakdowns of this bearing type do occur. Some of them may be caused by insufficient flow of the lubricating agent through the bearing, resulting in its insufficient cooling. It should be kept in mind that a polymer bush does not conduct heat well and under normal conditions most of the heat generated in the friction zone is absorbed by the flowing lubricating agent.

Friction is the decisive factor that determines whether overheating occurs in the friction zone. In the case of a cooling system breakdown, the heat must be transferred directly to the surroundings. With increasing friction, the amount of heat generated in the bearing rises. In the event of a cooling system malfunction, the heat must be transferred directly to the surroundings. Since heat exchange with the surroundings is problematic, due mostly to the bush's low thermal conductivity coefficient, the temperature of the bush increases, as does the temperature of the shaft, to a smaller degree. If the working conditions do not change, then after some time these temperatures should stabilize. In designing a bearing, the designer attempts to ensure that the temperature which establishes itself in the friction zone is within an acceptable range.

Overheating of the friction zone usually results in polymers undergoing a very rapid adhesive wear process. In the case of composites, delamination (separation of material layers) frequently takes place. Therefore, ensuring adequate cooling is of vital importance.

The ability to work under conditions of increased temperature is very important for a polymer bush and may turn out to constitute a significant advantage of a particular material. It is for this reason that certain sliding material manufacturers pay such close attention to this issue in efforts aimed at improving their products.

2. Origin and purpose

Existing specialist literature does not contain many descriptions of incidents of excessive wear or breakdowns in marine propeller shafts or water turbine bearings (Harish [9]). This stems from, among other reasons, the fact that the authors of expert opinions ordered by courts or ship classification societies are bound by confidentiality clauses. The manufacturers, if presenting the results of their research at all, naturally tend to present their products in the best possible light. However, it may be objectively stated that in recent years most of the reputable manufacturers have invested substantially in R&D, which has resulted in new, modern products with improved tribological properties appearing on the market (Ford [15], Laskey [6], Yamjo [17]).

Lack of information about bearing operation under limited lubrication and cooling conditions was the main motive behind the conducted research. How intensively can the bearings be loaded when no flow conditions appear? Will overheating appear in all the bearing types? Finding the answers to these questions was the purpose of the conducted research.

In real-life operations, breakdowns of stern tube bearings in propeller shafts do take place and frequently lead to very serious consequences, resulting in ships being taken out of service. Such breakdowns may be caused by malfunction of the pumping system (Fig. 1). Classification societies which validate ships for service require that flow and temperature sensors be placed in the cooling system, which in many cases has made it possible to prevent serious breakdowns.

Fig. 1. Melted polymer – the result of an overheated stern tube bearing caused by a cooling system breakdown; shaft diameter 350 mm.

On occasion, breakdowns caused by an overheated bearing may result from completely unexpected blocking of flow. There is, for example, a known case of a fishing line with hooks which wound itself up on a propeller shaft of a fishing vessel. The high temperature brought about by the resulting friction melted the nylon line into a monolithic ring which effectively blocked the outflow of water from the bearing (Fig. 2).

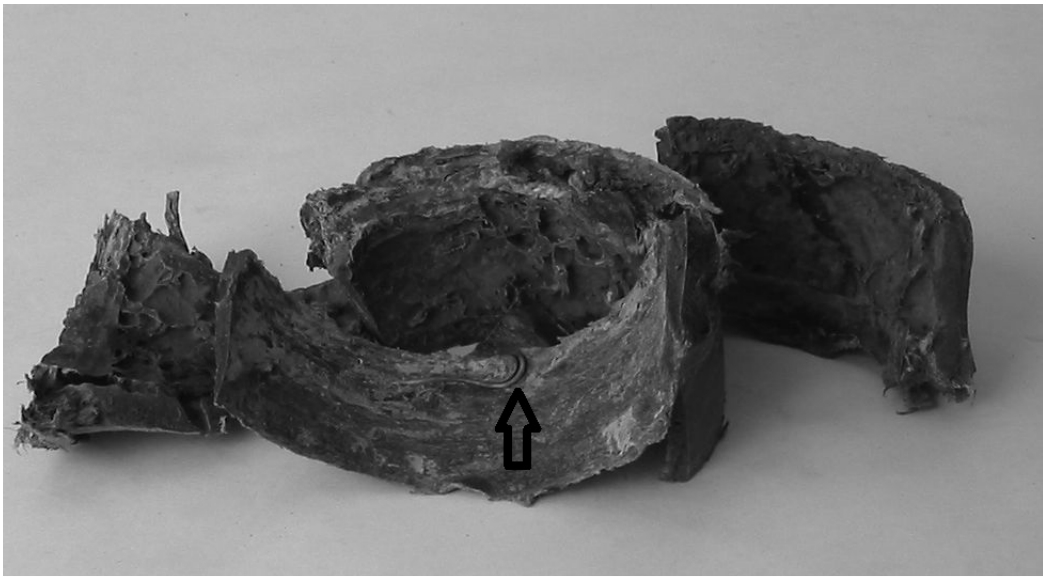

Fig. 2. Ring made of polymer hooking line which blocked water flow through fishing vessel propeller shaft bearings. The arrow indicates one of the fish hooks.

Experiment-based research into real-scale water-lubricated bearings is currently conducted in scientific centres all over the world and its results may be found in the scientific literature. One of the fundamental problems that are frequently brought up is the issue of hydrodynamic lubrication by low-viscosity lubricating agent such as water. It is for this reason that such aspects as the type of bush material, bush shape, location of lubrication grooves, and condition of the sliding surface are analysed (Cui [18], Dong [19], Ford [15], Gao [20], Hryniewicz [21], Litwin [22,23,24], Łubiński [24], Orndorff [16], Wang Y. [25], Yamjo[17]).

3. Scope of the tests and description of tested bearings

A group of six typical marine-industry water-lubricated bearings were chosen for the tests (Table 1). All of the bearings have been accepted by classification societies for use as stern tubes of propeller shafts. The bearings were of differing bush materials and geometries according the manufacturers' suggestions.

Table 1. Bearings subjected to experimental tests

No.	Code	Bush type and geometry / bush thickness [mm] / modulus of elasticity (E) if available	Approximate diameter clearance [μm] V: vertical, H: horizontal)	
			Left side of bush	Right side of bush
1	A	Rubber – NBR – six grooves around entire circumference / 12 mm	H = 900 V = 800	H = 800 V = 1000
2	B	Rubber – NBR – partial arc, grooves in upper part / 12 mm	H = 500 V = 600	H = 600 V = 600

3	C	Three layers – six grooves along entire circumference / PTFE 5 mm, NBR 5 mm, brass 3 mm	H = 450 V = 500	H = 500 V = 500
4	D	Three layers – partial arc, grooves in upper part / PTFE 5 mm, NBR 5 mm, brass 3 mm	H = 480 V = 600	H = 500 V = 480
5	E	Elastic polymer – partial arc, grooves in upper part / 12 mm / E = 600 MPa	H = 300 V = 400	H = 180 V = 200
6	F	Stiff composite – partial arc, grooves in upper part / 12 mm / E = 4000 MPa	H = 300 V = 350	H = 250 V = 300

The bushes made of rubber (NBR) were purchased as elements ready for assembly without requiring any finishing work (A and B). The two 3-layer bushes and the polymer bush (C, D, E) were provided, ready for assembly, by reputable manufacturers. The composite bush (F) was machined according to the manufacturer's guidelines.

All the bearings went through a run-in process which lasted dozens of hours. The running-in procedure was suggested by one of the manufacturers and was the same for all the tested bearings. The first stage of the process – loading of 0.25 MPa at a speed of 3 rev/s (1 m/s) – lasted for six hours. In the second stage, the load was increased to 0.5 MPa and the sliding speed remained the same. In case of the three-layer bearing which has lower frictional resistance, the final load was 1 MPa. It was impossible to conduct the same running-in process for different bushes because of certain limitations. Deformation of NBR bushes was too large to be sealed and caused a leak. Test rig propulsion was not strong enough to provide constant fluent motion because of very high frictional resistance in the polymer and composite bearings.

All of the tested bearings had a certain fault in the shape (non-cylindrical bush shape). Its magnitude may be estimated based on the bearing clearance value presented in Table 1. Such faults are typical and stem from difficulties during the process of machine-cutting of elastic bush material.

Bearings of two types of bush shapes were investigated. The bearings with grooves around the entire circumference (A, C) represent a classic solution which is frequently encountered when radial loads are not high or when the lubricating agent carries with it hard, solid particles. Due to its geometry, this type of bearing has very limited hydrodynamic capacity (Litwin [12]). The bearings with bushes in which the grooves were machined only in the upper, load-free half (B, D, E, F) have substantially better hydrodynamic properties. This has an impact on both their durability and the resistance of motion levels. As a result, it also influences the amount of heat generated in the friction zone, which may be of crucial importance when the bearing is working under near-breakdown conditions of severely limiting or even stopping the flow of lubricant.

4. Research methodology

A special test rig for investigating water-lubricated sliding bearings of the authors’ own design and construction was employed in the research work. The key fragment of the rig is presented below (Fig. 3.). The rig allows work to be conducted on sliding bearings at life-sized scale and under conditions approximating those which are found in ship propeller shafts and water turbine bearings.

The main shaft, with a diameter of 100 mm, was made of marine-grade stainless steel X10CrNi18-8, which is frequently used in this type of solution, especially those working in salty sea water. The drive was provided by an electric motor equipped with a reduction gear. Such a solution allowed a maximum starting torque of over 160 Nm to be reached. A static radial load was applied by suspending a weight on a special lever. The measurements were made for three load values of 4, 8, and 12 kN, which corresponded with calculated pressure values of 0.2, 0.4, and 0.6 MPa. Pressure values of 0.2 MPa may be considered as low, 0.4 MPa as normal, and 0.6 MPa as high. Past research indicated that certain materials are capable of working properly even at pressures as high as 0.8 MPa (Litwin [23]).

Using the test rig offers wide research possibilities. It allows the friction moment of the tested bearing, its clearance circle (set as the outmost limiting position of the journal in the bush), and pressure distributions at selected points in the space between journal and bush to be recorded. However, there are also certain limitations which result from the way in which radial force is applied on the tested sliding bearing. The load is applied through two rollers with ball bearings. It is for this reason that the recorded value of the friction coefficient is decreased by the motion resistance of the load-applying rollers as well as increased by the resistance of two sealing rings which allow water to be forced through the enclosed area of the bearing unit. This is why measuring the moment of friction, especially in the range of lower values, is marked by certain error. The measurement accuracy increases when recording higher motion-resistance levels, for instance during start-up. The majority of test rigs of this type used in scientific research centres all over the world have similar limitations (Gao [20], [Łubiński [24], Wang N. [13], Wang Y. [25], Yamjo [17]).

The tested bearing assembly was equipped with three temperature sensors, which allowed measurements to be conducted during the experiment. Two of the sensors recorded the water temperature in the proximity of the bearing's edge (Fig. 3, elements 8), while the third measured the temperature of the steel sleeve, that is, the bearing casing.

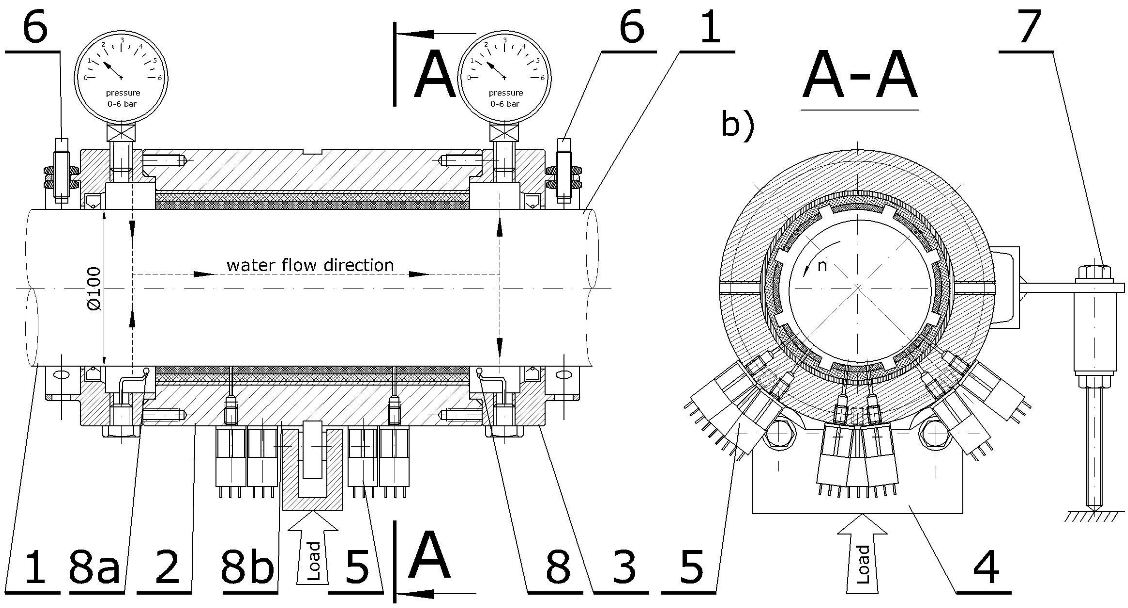

Fig. 3. Tested bearing assembly – test rig. 1. stainless steel main shaft, 2. steel sleeve – bearing casing, 3. covers with sealing rings, 4. radial force application unit, 5. sensors measuring pressure between journal and bush, 6. sensors measuring distance between journal and bush, 7. arm and sensor for measuring moment of friction, 8a. sensors of water temperature flowing through the bearing, 8b. sensor measuring steel bush temperature

The investigation of the impact of a lack of flow of lubricating agent through the bearing was carried out in the following way. During work in stable conditions at a shaft revolution speed of 11 rev/s, the flow of water through the bearing was stopped using valves on the inlet and outlet sides. The bearing assembly presented in Fig. 3 was filled with water but the entire heat generated in the friction zone was transferred through the casing walls and drive shaft to the external surroundings. The room temperature was 20 ˚C.

The measurements were conducted when there was no flow of lubricant and were continued until:

- The surrounding water temperature recorded on one side of the bearing reached approximately 80 ˚C. Once this value was reached, the valves were opened and water circulation was restored in order to cool the bearing. Thanks to this, bush damage was avoided.
- The surrounding water temperature stabilized: there was no temperature change for a certain, assumed time period.

The working conditions simulated on the test-rig were worse than those which are usually present in real-life situations, particularly in the case of heat exchange with the surrounding environment. On an actual ship, the heat generated in the friction zone is transferred through the bush to the steel protecting tube of the propeller shaft and then further on through the hull into the water, which is frequently below 10 °C. In the case of the test-rig, the heat from the sleeve was transferred to the surrounding air, and therefore the temperature during tests reached levels in excess of 50 °C. In order to cope with this problem, Yamjo and Kikkawa [17] kept the entire bearing assembly submerged in water during tests. However, this solution makes it difficult to conduct certain measurements as some of the apparatus has to work under water.

5. Results

The experimental results presented below are the product of nearly two years of working effort. During that time, a number of bearings were investigated and other measurements were taken, mostly aimed at defining their hydrodynamic load capacity and resistance of motion levels (Litwin [23, 26]).

The graphs present increases of average water temperature in the proximity of the bush as well as the temperature of the steel sleeve as a function of time, with measurements starting at the moment of blocking of lubricant flow (Figs. 4–10).

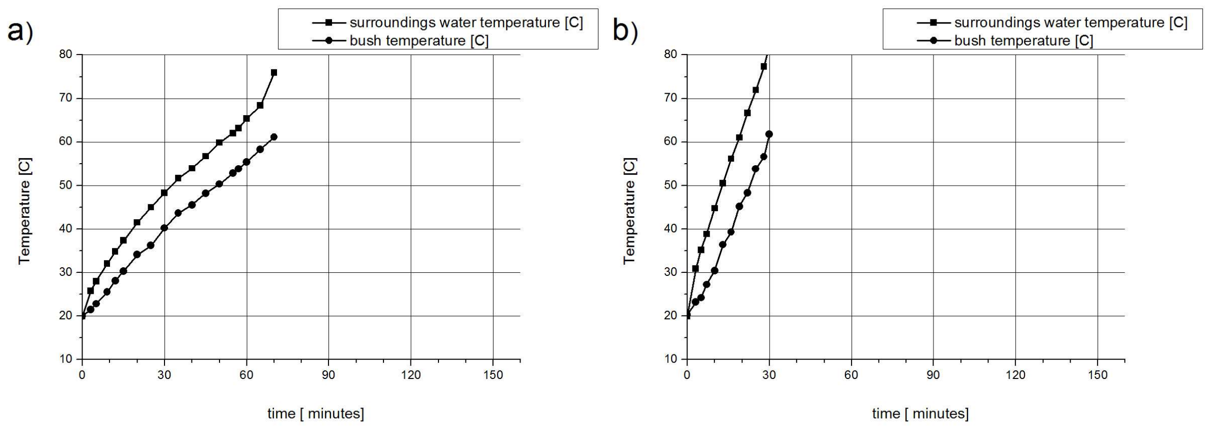

Fig. 4. Temperature increase diagram for bearing A, shaft revolution speed = 11 rev./s,
a) pressure = 0.2 MPa, b) pressure = 0.4 MPa

b)

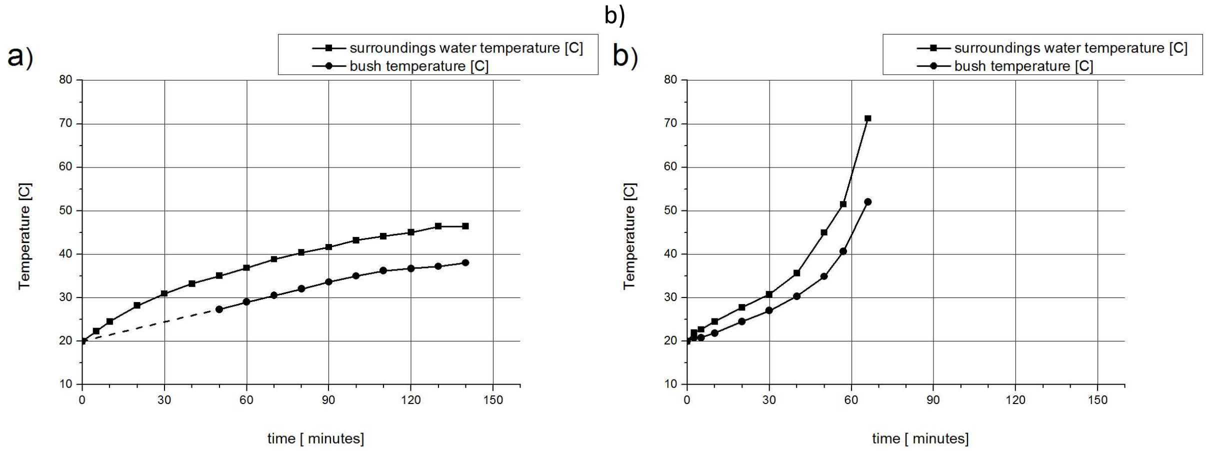

Fig. 5. Temperature increase diagram for bearing B, shaft revolution speed = 11 rev/s,
a) pressure = 0.2 MPa, b) pressure = 0.4 MPa

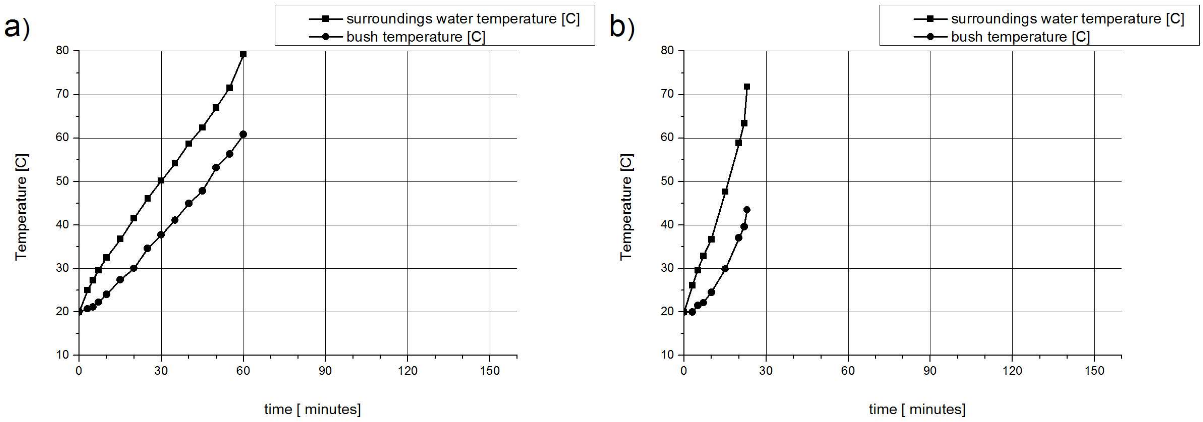

Fig. 6. Temperature increase diagram for bearing C, shaft revolution speed = 11 rev/s,
a) pressure = 0.2 MPa, b) pressure = 0.4 MPa

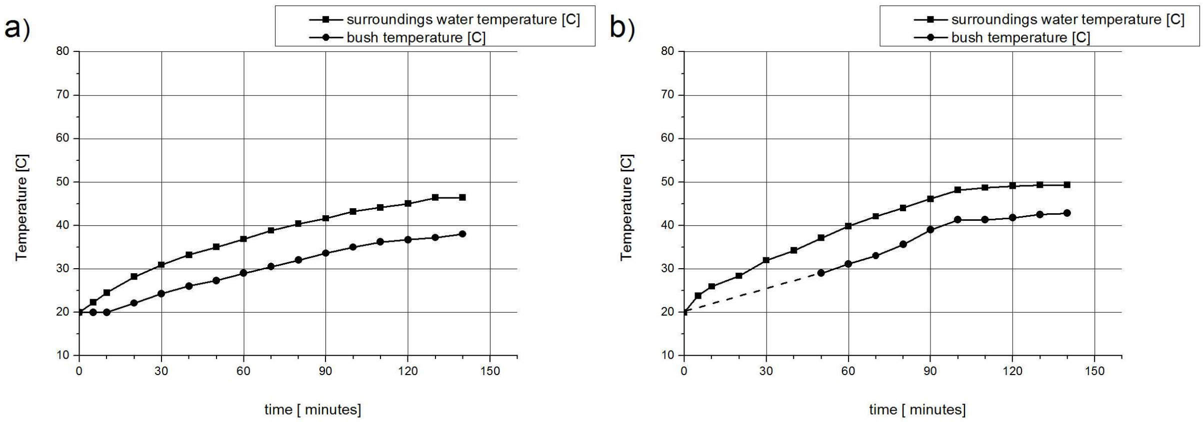

Fig. 7. Temperature increase diagram for bearing D, shaft revolution speed = 11 rev/s,
a) pressure = 0.2 MPa, b) pressure = 0.4 MPa

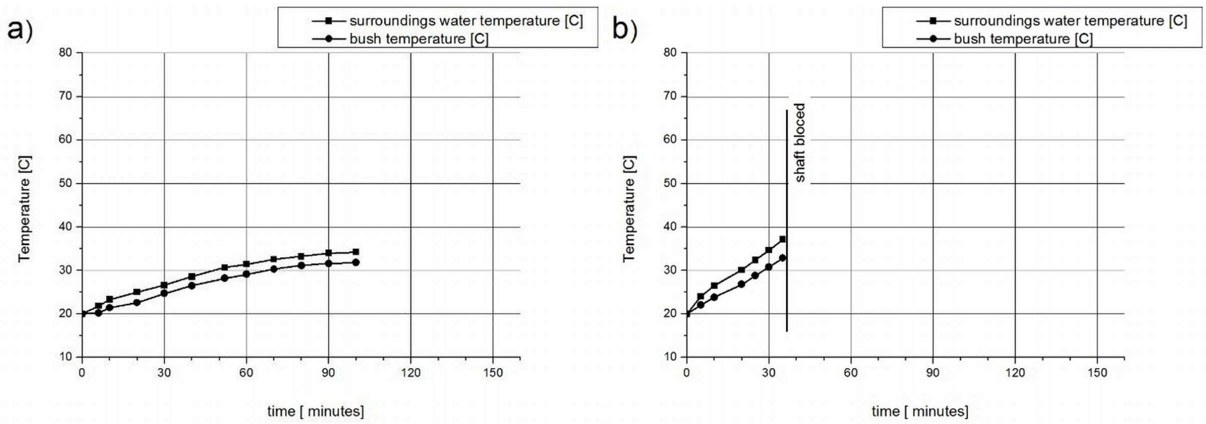

Fig. 8. Temperature increase diagram for bearing E, shaft revolution speed = 11 rev/s,
a) pressure = 0.2 MPa, b) pressure = 0.4 MPa

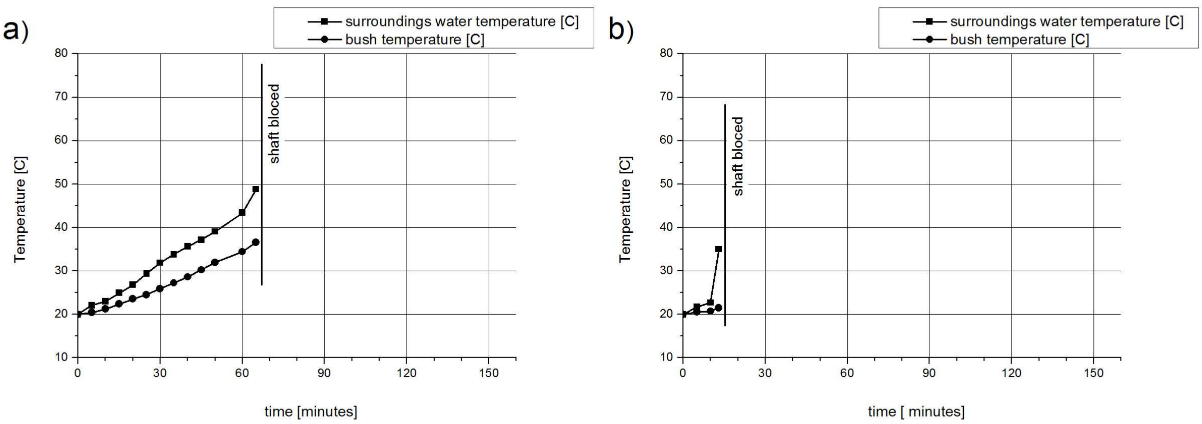

Fig. 9. Temperature increase diagram for bearing F, shaft revolution speed = 11 rev/s, a) pressure = 0.2 MPa, b) pressure = 0.4 MPa

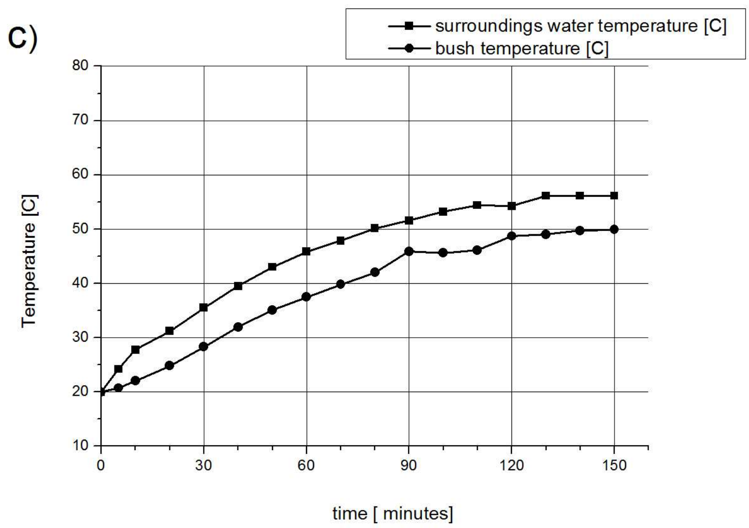

Fig. 10. Temperature increase diagram for bearing D, shaft revolution speed = 11 rev/s, pressure = 0.6 MPa

6. Discussion

In analysing the measurement results of the temperature increase over time, one may observe that the best working behaviour in a simulated breakdown was displayed by the three-layer bush bearing with a PTFE sliding surface and lubrication grooves placed in the upper part of the bush (D). Following approximately two hours of work, the bearing's surrounding water temperature became stable. The water temperature inside the bearing unit was below 50 °C and the temperature of the steel bush was about 40 °C. Such positive results indicated the low level of friction loss and provided encouragement for conducting tests with greater loads of 1.2 kN (pressure = 0.6 MPa, Fig. 10). These tests also proved that the bearing temperature stabilizes itself, as was the case in earlier trials. However, due to the greater loads and therefore also the increased friction, the recorded temperatures were a few degrees higher. Earlier research in which frictional characteristics, shaft orbits, and pressure distribution were acquired (Litwin [23,26]) had demonstrated that this bearing (D) has very good tribological properties: low resistance of motion levels during start-up and work in a mixed lubrication regime. The bearing also exhibits high hydrodynamic capacity as far as water-lubricated bearings are concerned. The undoubtedly excellent result is not, however, due only to material properties. It is also the result of work in the hydrodynamic lubrication regime, which is supported by the optimum geometry of the bush. This is attested by the fact that a bearing with a bush made of the same material (C) reached a temperature of approximately 80 °C (Fig. 6.) in the zone around the bearing after about 20 minutes of work under a pressure of 0.4 MPa and approximately an hour of work under a pressure of 0.2 MPa.

Earlier comparison-based research into bearing groups of classical geometry with grooves around the entire circumference (A, C) had demonstrated that their hydrodynamic capacity is low (Litwin [26]). The load-carrying capacity may be further limited by sliding surface roughness and there are significant faults in cylindricity (A). As a result of the conducted tests, it was determined that the bearings with classical geometry (A, C) overheated most rapidly. This is because they are working in mixed lubrication conditions, when the power of friction is substantial. In such a scenario, the temperature increases rapidly (Figs. 4, 6).

The executed tests demonstrated that a rubber bearing of classical geometry overheats very rapidly. However, it should be kept in mind that the investigated bearings were characterized by bush length equal to two journal diameters. In shipbuilding, such materials (A and B) are allowed to be used as propeller shaft bearings under the condition that their length is equal to as many as four journal diameters. As a result, the loads are decreased by half. Therefore, one may suppose that, especially in the case of the rubber bush bearing with grooves in the upper half (B), the bearing will work without failing in the conditions of no lubricating agent flow.

In analysing the working conditions of the investigated bearings, one may put forward the thesis that if hydrodynamic lubrication takes place in the bearing (so the power of friction is low) then it may be reasonably assumed that the bearing is going to work reliably despite a lack of cooling by flowing lubricating agent. However, it turns out that in fact this is not always the case. The rubber bearing (B) worked without overheating only at low loads (Fig. 5a.). At greater load values, the bearing overheated and the measurements had to be stopped. This demonstrates the fact that bearings with highly elastic bushes undergo significant deformation as a result of pressure in the lubricating film. The effect is such that the bearing may work in rather specific conditions. The middle of the bush, which is highly deformed by lubricating film, has no contact with the shaft, whereas mixed lubrication, which is a source of heat, takes place at the edges (Litwin [27]).

The measurements recorded for the polymer and composite bearing (E, F) were quite interesting. Both bearings have optimum geometry for facilitating hydrodynamic lubrication, despite having substantial faults in shape (Table 1). In the case of the polymer bearing (E) at low pressure (0.2 MPa), the working conditions stabilized and no overheating took place (Fig. 8a). After increasing the pressure to 0.4 MPa, the measured temperatures increased significantly more rapidly. However, most worrying was the fact that after approximately 35 minutes of work the level of motion resistance suddenly began to increase. The inverter which controlled the engine was not able to maintain the set revolution speed and would turn the engine off, signalling torque overload. The experiment was repeated a number of times and the differences in the time period after which the shaft stopped were minimal (33 to 38 minutes). It was concluded that the material with a significant thermal expansion coefficient increases its volume and locks the shaft (clamps itself on the shaft, since the bearing clearance on one of the sides is too small; Table 1). It is with this reason in mind that measurements of the outmost locations of the journal in the bush were taken when the shaft locked following one of the experiments. It turned out that the bush did increase in size, causing a decrease in the clearance, but the shaft was not in fact locked.

The most surprising results were achieved for the bearing with the composite bush. For both simulated working conditions, the measured temperatures increased rapidly and finally the shaft was blocked.

In the authors' opinion, on passing a certain temperature value, an intensive slick-slip phenomenon occurs, causing increasing friction.

7. Conclusions

The completed experiment-based research confirmed that a three-layer bearing with grooves only in the upper bush part is a safe solution in the event of a lubrication-system failure under the condition where it is filled with water.

Due to the fact that heat being generated in the friction zone was transferred through the bush and steel sleeve and then into surrounding air but not into the water around the casing, as is usually the case for real units, one may predict that under actual conditions there will be no dangerous temperature increase in some of the researched bearings (especially B and C).

The point of modifying polymer bearing materials in order to increase their thermal resistance is highly justified. On the other hand, it should be kept in mind that a key role is played by the bearing's low level of friction, and therefore special emphasis should be placed on the design process, as well as on precise manufacturing and proper assembly of bearings, so that they are capable of working under the conditions of fluid film lubrication.

The opinion that in a water-lubricated bearing the majority of heat generated in the friction zone is carried away by the lubricating agent and that the heat exchange through the bush is minimal is in fact correct. However, the situation changes completely in the case of a cooling system breakdown. Therefore, in designing a bearing assembly it is worthwhile to do so in a way which creates the best possible conditions for heat exchange between the friction zone and surroundings.

During work under conditions of no lubricant flowing through the bearing, an important role is played by the thermal conduction coefficient of the bush material. Differences in value of more than 20% may have a significant impact on temperature increase.

Bearing overheating can appear in a simple open, as well as in a complicated closed system. Installing of flow sensors is necessary in all the previously described solutions.

In carrying out the experimental research, attempts were made to destroy the bearing through overheating. It was hoped that this might lead to the melting of bush material similarly to the situation illustrated in Fig. 1. However, it turned out that along with the increase in temperature there are also increases in friction levels, for which the torque provided by the drive unit of the test-rig was too low.

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