POLISH MARITIME RESEARCH 4(76) 2012 Vol 19; pp. 54-61
10.2478/v10012-012-0040-4

Assessment of possible application of water-lubricated sintered brass slide bearing for marine propeller shaft

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ABSTRACT

Water-lubricated bearings have been applied to support ship screw shafts for over a hundered years. Development of plastic materials has caused that novel sliding polymers appeared available on the market and being approved by classification institutions, possible to be applied in shipbuilding. However in the subject – matter literature there is no reference to application of bearings fitted with metal or ceramic bushes in shipbuilding. Nevertheless they have important merits such as low resistance to motion, long service life or stability of form. But some doubts are evoked by their large value of shape elasticity module which can lead to local stress concentration at the bush edges in the case of non-axiality of bush and shaft axes. Hence this work has been aimed at the testing of a bearing made of bronze-graphite silter. In the work is presented a comparison of measurement results of resistance to motion , hydrodynamic pressure distributions in lubricating film as well as shaft axis trajectories of the bearing, with those made for a typical elastic polymer bearing. The measurement results have showed high quality of the tested material. In the opinion of these authors it could be applicable to bearing ship propeller shafts.

Keywords: bearing; bearing of ship propeller shafts; tribology; water-lubricated bearings

INTRODUCTION

As a result of becoming screw propulsion on ships widespread it was necessary to elaborate a reliable and durable bearing for ship propulsion shafts. Most of operational problems have been generated by the last section of the shaft on which screw propeller is installed – the so called screw propeller shaft. It results a.o. from a complex loading system in consequence of mass forces and harsh working conditions, especially exposure to salt sea water. From the very beginning of the application of mechanical propulsion to ships till now a slide bearing has been used in the classical propulsion system of floating units. In the past attempts have been made to use rolling bearings, however such solution can be met now rather rarely as it makes repair technology complicated, is less durable as well as cannot be used for large diameter shafts [1, 2].

Since many years ago engineers have faced troubles associated with propeller shaft bearing and in the past many failures of shafts, bearings and sealings, have happened. With the famous ship *Great Eastern* is connected an interesting story dealing with her propeller shaft bearing. This unusual ship propelled by screw propellers and side paddle wheels, of more than 200 m in length, built in 1859, was then several times greater than any other ship. After two cross - Atlantic trips it was revealed that her stern slide bearing fitted with white metal liner was completely worn out. Measurements of position of the shaft (shaft settling) showed that its wear reached the value of 63 mm. According to suggestion of John Penn, a British engineer, the white metal bearing was replaced by a new one made of guaiac wood (*lignum vitae*) [3]. After completing such modernization the problem of stern bearing did not recurred during the entire service period of the ship.

Already at the beginning of 20th century such wooden bearings have been in common use. However they have had some limitations. Among the most important can be numbered the sensitivity to lubrication with a contaminated lubricant e.g. in the conditions of sailing in shallow or inland waters.

Dynamical development of novel tribological materials including plastics has resulted in their successful application as a sliding material to water-lubricated bearings. Today there are a few worldwide companies which produce bearings with a polymer sliding bush approved by ship classification societies for application to ships.The tests performed in the past have demonstrated their high quality, hence they are widely applied today on both small and large ships.

In spite of the common application of polymer bearings, new research projects aimed at elaboration of better and better sliding materials, are still under way. Bronze-graphite sinters of all kinds can serve as an example of a material alternative to plastics. They have many merits, e.g. a high stability of form, higher resistance to elevated temperature and seizing-up.

ORIGIN

For recent years many types of water-lubricated slide bearings have been tested at Gdańsk University of Technology [4÷8]. Among other things, have been performed tests of a water-lubricated bearing fitted with the sliding bush made of BA1032 (CuAl10Fe3Mn2) Al-Fe-Mn bronze and journal of X10CrNi18-8 (AISI 301) stainless steel. The tests showed that such association cannot be lubricated with water, despite its perfect work in the conditions of lubrication with oil or plastic lubricant. The tested bearing has been seized up shortly after its starting (Fig. 1).

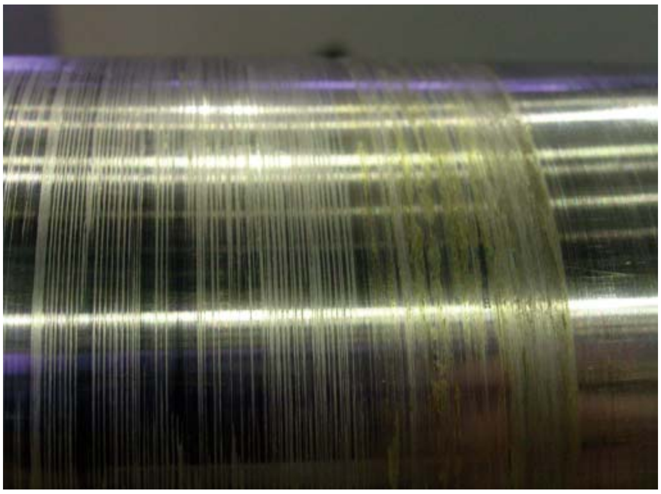

Fig. 1. *The damaged shaft made of X10CrNi18-8 stainless steel-fragments of the broze bush can be observed on its surface*

However in recent years novel, special sliding materials made on the basis of bronzes, iron or nickel, capable of operating in water, have appeared. One of such materials is bronze –graphite sinter. Its very good sliding qualities in the case of lubrication with water, advertised by its producer, result from its composite structure. Into bronze-tin matrix a lubricating medium in the form of graphite was inserted. Due to this, even after breaking lubricating film graphite particles effectively lower friction forces and prevent seizing. The material can be applied even in the temperature close to 800°C and is of a high resistance against corrosion. As novel sintered materials are successfully applied to water-lubricated bearings of vertical water turbines it was decided to test a material of the kind in the operational conditions of ship stern tube bearing. It should be stressed that screw propeller shaft which operates in horizontal position, produces much more harsh operational conditions for such bearing in comparison with those for a vertical shaft. It results from greater radial loads due to statical and dynamical mass forces occurring in the shaft - screw propeller system.

It was decided that such metal bearing will be compared with a typical, water-lubricated polymer bearing approved by classification institution. Such comparison will make it possible to determine drawbacks and merits of both the solutions.

For the tests was selected a bronze-graphite silter offered in the form of prefabricated bush elements pressed under high pressure and temperature. The bronze bush was pressed into a steel sleeve and then the internal hole was machined to its final dimensions. For the reason of a relatively high hardness of the implemented bronze (in comparison with that of polymer) the bush of the kind has been intended for interaction with the journal of the hardness of 35HRC at least. Therefore on the shaft made of X10CrNi18-8 steel a hardened-tempered sleeve made of X30Cr13 (AISI 420) steel was bonded by using thermo-compression method. The sleeve was next grinded and polished to ensure appropriate coaxiality with respect to journals of bearings which supported the shaft during the tests.

a)

b)

Fig. 2. *The tested bearing fitted with metal bush:* ***a)*** *the prefabricated sliding bush element made of siltered bronze,* ***b)*** *the complete bearing before its mounting at the test stand, in which the sliding bush and outer steel sleeve with circumferentially placed sockets for pressure transducers, are seen.*

PLAN OF THE TESTS

Data of the tested bearings are presented in the tab. 1.

As resulted from measurements, the polymer bearing had a non-ideal form (cylindricity error) as well as a worse state of sliding surface in comparison with that of the bearing made of silter. It results from difficulty in machining a very elastic polymer and its deformations resulting from absorption of water (soaking).

Many calculations performed in the past, based on hydrodynamic lubrication theory for EHL model [5, 6, 7] have shown that the bearing with rigid bush operated at a larger film thickness in comparison with the same bush but made of polymers. The different thickness of lubricating film in rigid

Tab. 1. Comparison of the tested bearings

Material	Shear modulus [MPa]	Shaft diameter [mm]/Bush length [mm]	Bearing clearance [mm]	Sliding surface roughness R_a [µm]	Shaft journal roughness R_a [µm]
Bronze-graphite silter	1.1x10^5	100/150	0.15	0.16	0.09
Uniform elastic polymer approved for marine applications	6x10^2		0.17-0.19 (shape errors)	0.35	0.09

bearing and elastic one results from elastic bush deformation under pressure, which changes lubricating gap form [9].

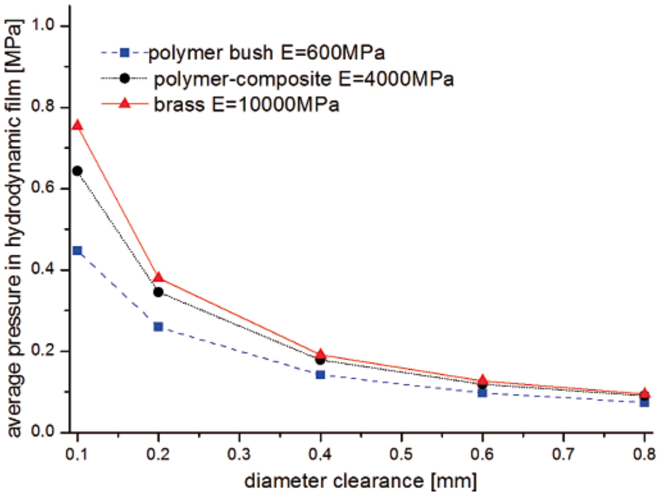

Fig. 3. Influence of shear modulus of bush material on load-carrying capacity of bearing, in function of bearing clearance value; journal diameter of 100 mm, shaft rotational speed of 11 rev/s

One of the main aims of experimental tests was to compare theoretical results with experimental ones as well as to determine drawbacks and merits of the bearings with metal bush.

EXPERIMENTAL TESTS

For the experimental tests was used the test stand already described in the subject-matter literature [6,10]. It has wide research possibilities. During the tests, resistance to motion, pressure in lubricating film as well as shaft axis trajectory was measured.

The tests were performed for working parameters typical for marine application of such bearings. In order to gain as much information about tribological qualities of both the bearings as possible,was planned a set of tests in different conditions realized by changing both load and rotational speed. The load was applied statically by hanging a weight. The tests were performed for three different load values giving the average pressures of 0.2, 0.4 and 0.6 MPa, respectively. Rotational speed was varied in the range from 0 to11 rev/s. The measurements were taken for the speed values of 1, 2, 3, 5, 7, 9 and 11 rev/s. In the case of determination of resistance to motion also statical friction coefficient was measured.

During the tests, water of 15°C temperature was pressed through the bearing. Its flow rate amounting at least to 6 l/min was sufficient to ensure proper lubrication of the bearing. All the measurements was performed for the initially run-in bearings (after 24h working period in the range of low rotational speeds, at mixed friction work).

RESULTS OF THE TESTS

Tests of starting resistance

A very important feature of water-lubricated slide bearings is their capability of starting under load. This is especially important in the case of horizontal ship propeller shafts which exert their weight onto bearings regardless of whether the shaft is in idle or motion state. In slide bearings, in contrast to rolling ones, the problem of appearance of very large values of resistance during start-ups under load, often happens. It results from a large, statical friction coefficient which occurs in conditions of lacking hydrodynamical lubrication. In the case of polymer bearings the effect can be heightened by stick-slip

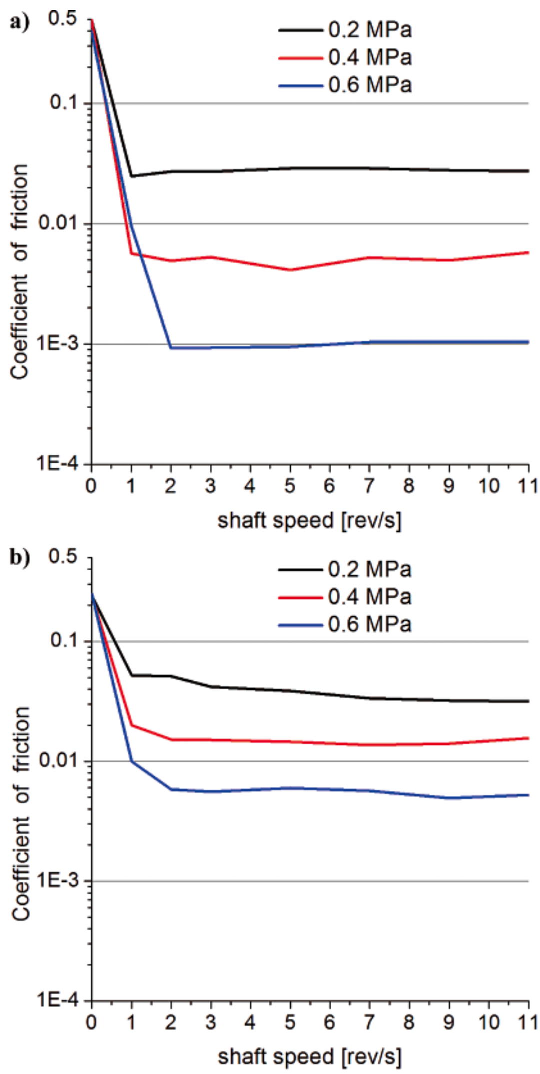

*Fig. 4. Measured friction coefficient in: **a)** bearing with metal bush, **b)** bearing with polymer bush*

phenomena. Usually, after reaching even a low rotational speed (of abt. a few rpm), a partial raise of hydrodynamical pressure occurs and in consequence friction coefficient drops suddenly. In Fig. 4, are presented the diagrams of measured friction coefficients in metal bearing and polymer one in function of rotational speed and average pressure values.

The recorded friction coefficient values indicate that in the bearing of metal bush a larger value of friction coefficient equal to abt. 0.5, occurs. In the same conditions in the polymer bearing the friction coefficient amounts to abt. 0.25. It means that the bearing with metal bush requires at least two times greater starting moment in comparison with the polymer bearing. It is also seen that in the metal bearing dropping resistance to motion is faster because after reaching the rotational speed of 0.5 rev/s the friction coefficient drops to the five times lower value equal to abt. 0.1. It is undoubtedly connected with forming hydrodynamic lubricating wedge in the rigid bearing faster than in the elastic (polymer) bearing. It should be mentioned that in the bearing with metal bush lower resistance to motion occurs in steady working conditions (regardless of pressure values).

Tests of journal displacement against bush

The trajectory diagrams are presented on the background of experimentally measured clearance circles (Fig. 5 ÷ 10). It makes a better interpretation of the results, possible. Moreover, shape of the clearance circles reflects shape of the bush, hence it makes it possible to assess bush shape errors. The shape error which appears in the bearing of polymer bush is not surprising

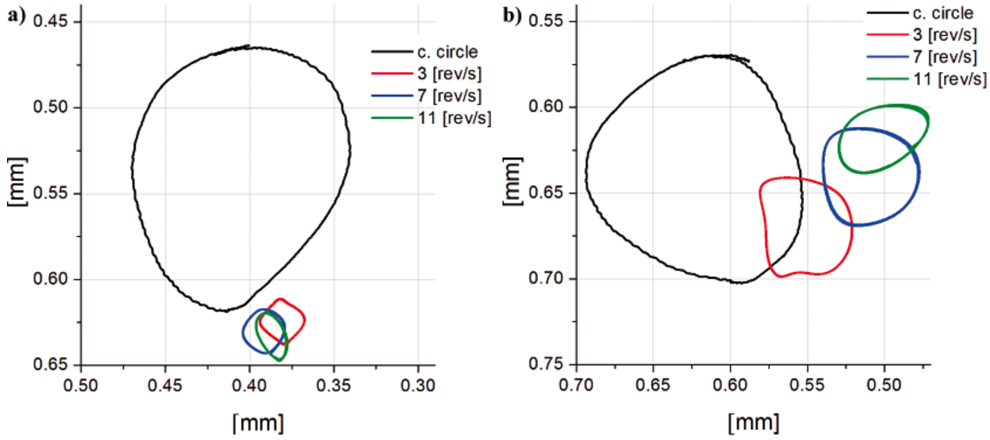

Fig. 5. *Shaft orbits measured on both sides of the bearing with metal bush, under load of 0.2 MPa;* ***a)*** *left side of the bush,* ***b)*** *right side of the bearing; orbits for three successive shaft revolutions*

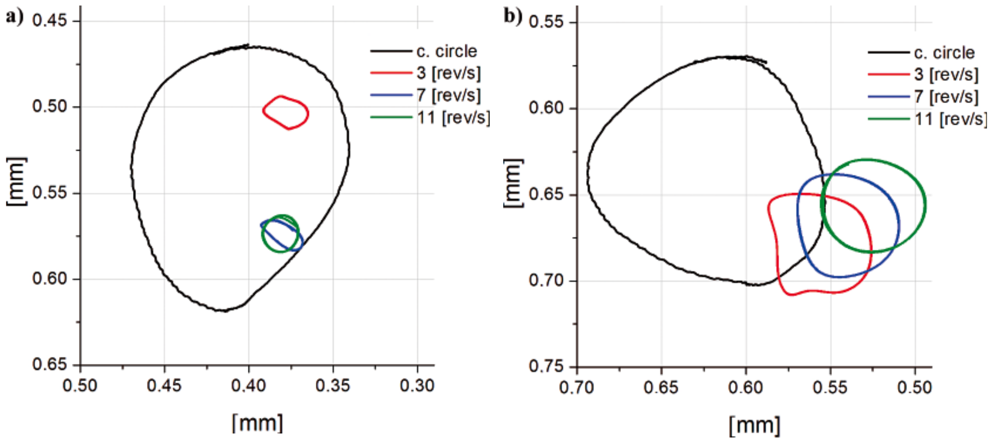

Fig. 6. *Shaft orbits measured on both sides of the bearing with metal bush, under load of 0.4 MPa;* ***a)*** *left side of the bush,* ***b)*** *right side of the bearing; orbits for three successive shaft revolutions*

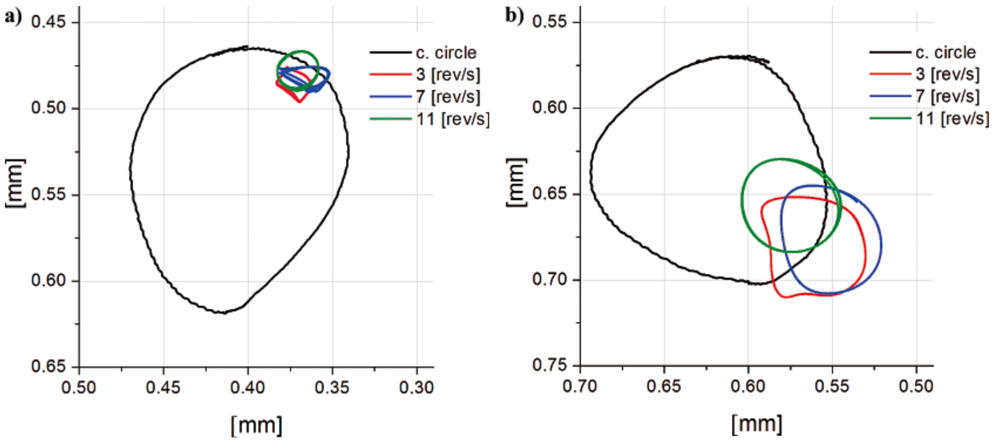

Fig. 7. *Shaft orbits measured on both sides of the bearing with metal bush, under load of 0.6 MPa;* ***a)*** *left side of the bush,* ***b)*** *right side of the bearing; orbits for three succesive shaft revolutions*

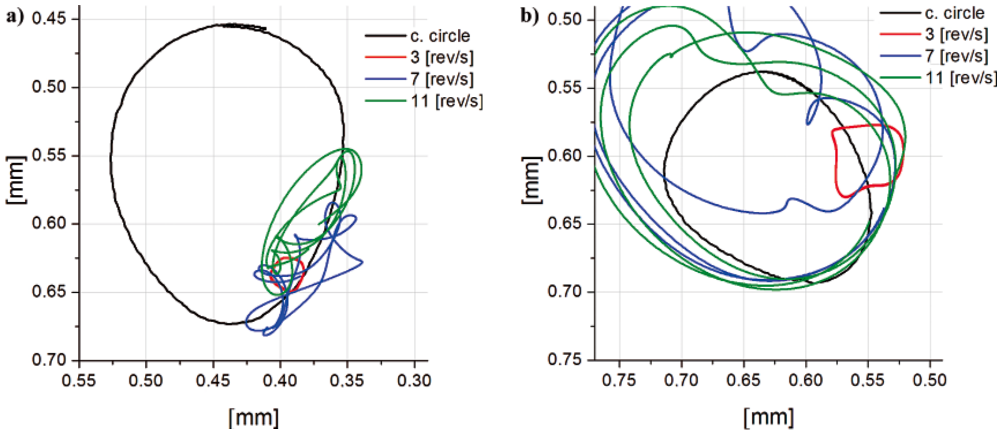

Fig. 8. Shaft orbits measured on both sides of the bearing with polymer bush, under load of 0.2 MPa;
***a)** left side of the bush, **b)** right side of the bearing; orbits for three succesive shaft revolutions*

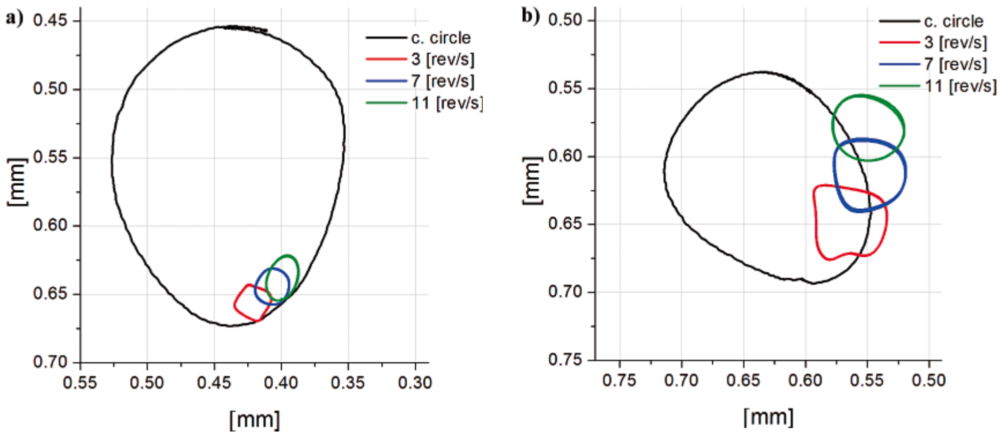

Fig. 9. Shaft orbits measured on both sides of the bearing with polymer bush, under load of 0.4 MPa;
***a)** left side of the bush, **b)** right side of the bearing; orbits for three successive shaft revolutions*

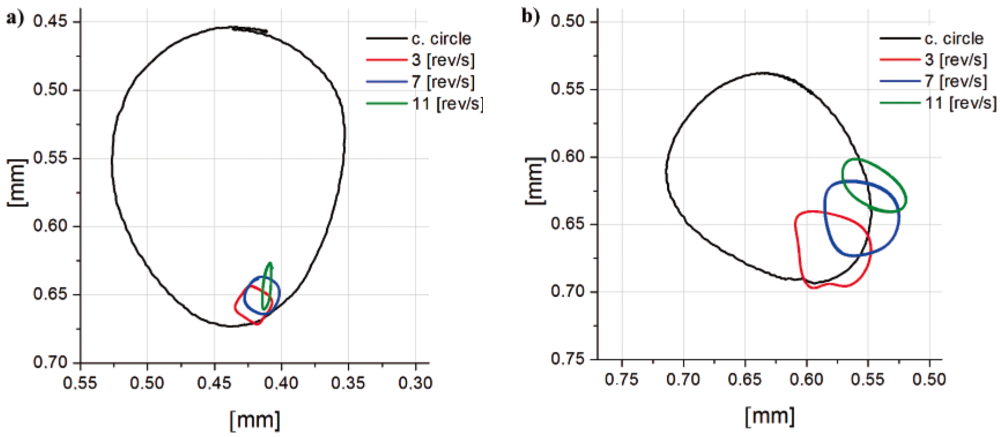

Fig. 10. Shaft orbits measured on both sides of the bearing with polymer bush, under load of 0.6 MPa;
***a)** left side of the bush, **b)** right side of the bearing; orbits for three successive shaft revolutions*

as it results from machining the elastic water-absorbing polymer. However, is puzzling the shape error of the bearing with metal bush machined to its final form after installation into the metal sleeve,. The tests of the bearing made of BA1032 (CuAl10Fe3Mn2) bronze, whose sliding surface was grinded, carried out in the past (as already mentioned), showed that it is possible to obtain its shape almost ideal (Fig. 11). In the discussed case the bush length was as large as 300 mm and in spite of that the high machining precision (H7 tolerance class) was achieved.

As results from the shaft axis trajectory measurements, the bearing operated at a significant non-coaxiality of shaft against bush (Fig. 5 ÷ 10). In spite of that it operated in the range of fluid friction. In the opinion of these authors it testifies to their high potential and big margin of load-carrying capacity. It should be added that the revealed non-coaxiality does not result from inappropriate assembling work but from free-support of the bearing, hence even a small asymmetry in hydrodynamic pressure distribution produces natural twisting of the bush.

Repeatability of trajectories draws attention. In the diagrams, orbits of three successive revolutions of shaft are presented. In most of the cases the oval orbits smaller than 10μm in size cover practically each other. Due to this the bearing practically does not produce vibrations, except of the situations of its unstable work (Fig. 8). Such phenomenon appeared only in the bearing with polymer bush under load of a small value.

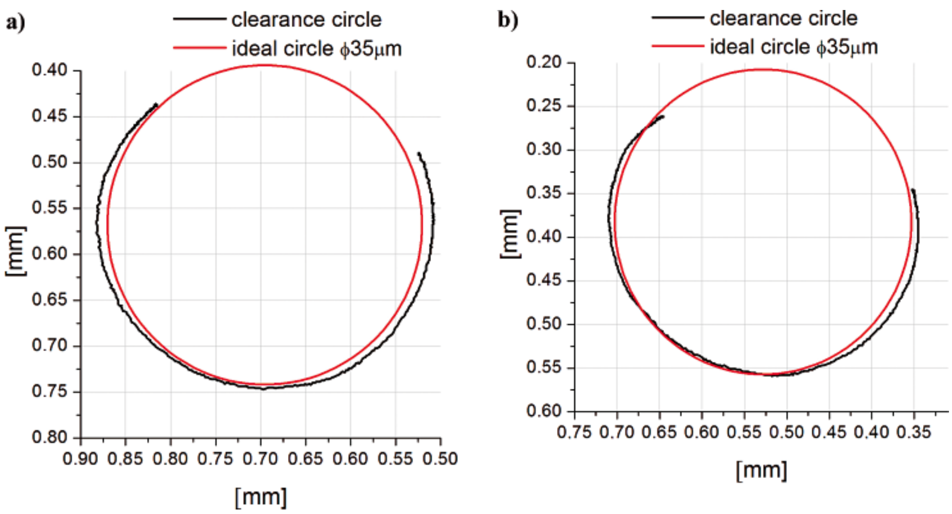

Fig. 11. Experimentally measured clearance circles of BA1032 bronze bearing whose sliding surface was grinded; a) left side of the bush, b) right side of the bearing

Tests of hydrodynamic pressure distribution

Below in Fig. 12 ÷ 14, are presented selected results of experimental tests of hydrodynamic pressure distribution in the metal (rigid) bearing and the polymer (elastic) one. By analyzing the obtained results the significant effect of bush rigidity on pressure distribution form can be observed. In the rigid (metal) bearing much greater maximum pressure which varies in function of values of average pressure and speed, occurs. The averaged values of the results achieved during the tests are given in Tab. 2 below. In the case of the rigid bush the maximum pressure is up to four times greater than the average nominal one, whereas for the flexible bush the maximum pressure does not exceed twofold value of the average pressure. It results from local change of lubricating gap form in the flexible bush, that leads to dropping maximum pressure at simultaneous increasing lubricating wedge lap angle.

During operation of the bearing in normal conditions pressure pulsations are rather small (Fig. 15a). As expected, the above described unstable work of the bearing (Fig. 8) generates significant pressure pulsations in the lubricating film (Fig. 15b).

Tab.2. Average values of pressure in lubricating film

	Shaft rotational speed [rev/s]	Nominal (average) pressure resulting from load [MPa]	Maximum pressure in lubricating film [MPa]
Rigid (metal) bush	3	0.2	1
	7	0.2	1.2
	11	0.2	0.8
	3	0.4	1.5
	7	0.4	1.7
	11	0.4	1.6
	3	0.6	2.1
	7	0.6	2.1
	11	0.6	2.1
Flexible (polymer) bush	3	0.2	0.38
	7	0.2	0.25
	11	0.2	0.28
	3	0.4	0.58
	7	0.4	0.5
	11	0.4	0.48
	3	0.6	0.75
	7	0.6	0.7
	11	0.6	0.68

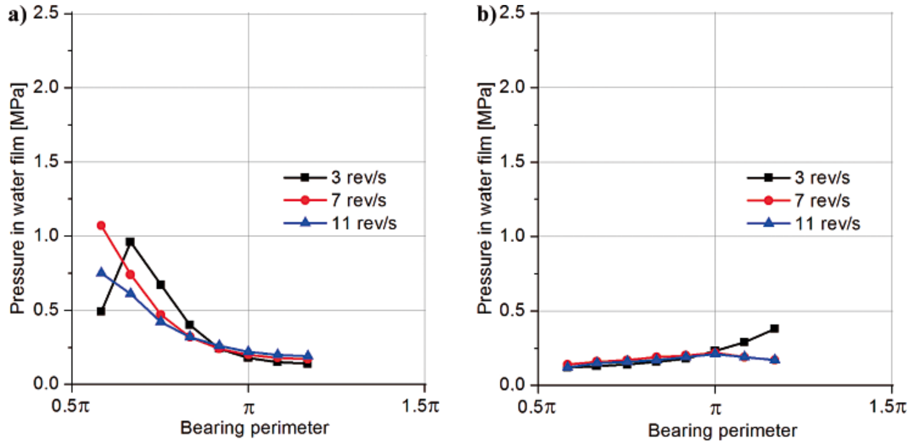

*Fig. 12. Measured distributions of pressure in lubricating film in lower half of the bush under load of 0.2 MPa; **a)** bearing with metal bush, **b)** bearing with polymer bush*

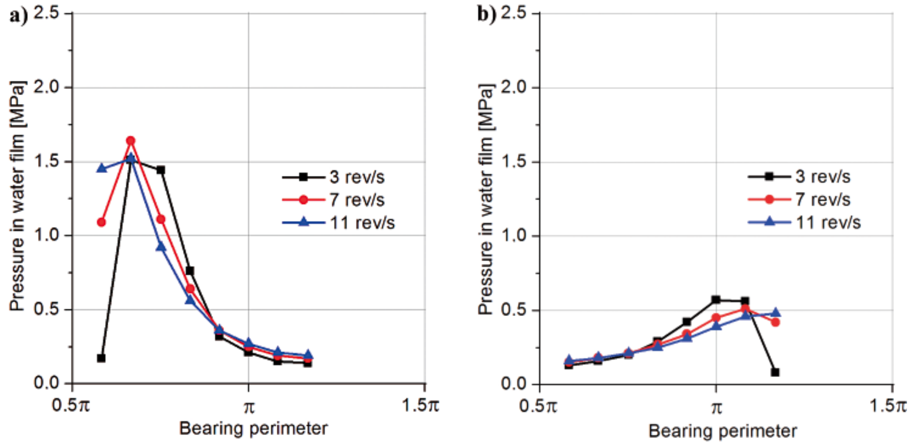

*Fig. 13. Measured distributions of pressure in lubricating film in lower half of the bush under load of 0.4 MPa; **a)** bearing with metal bush, **b)** bearing with polymer bush*

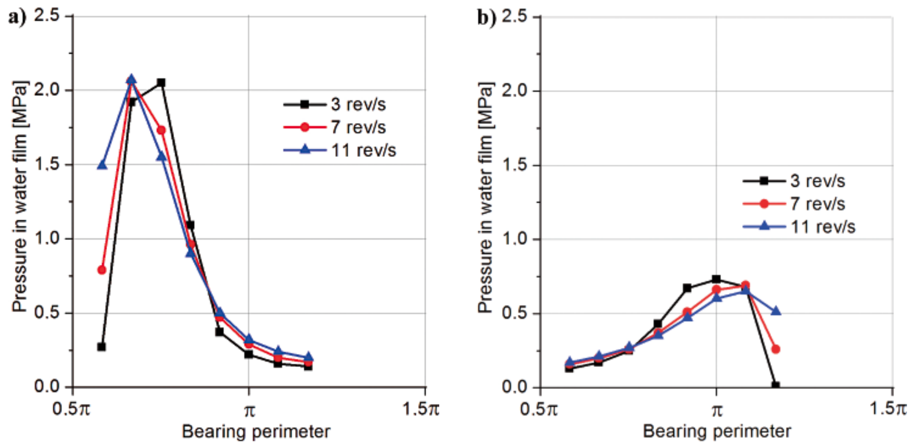

*Fig. 14. Measured distributions of pressure in lubricating film in lower half of the bush under load of 0.6 MPa; **a)** bearing with metal bush, **b)** bearing with polymer bush*

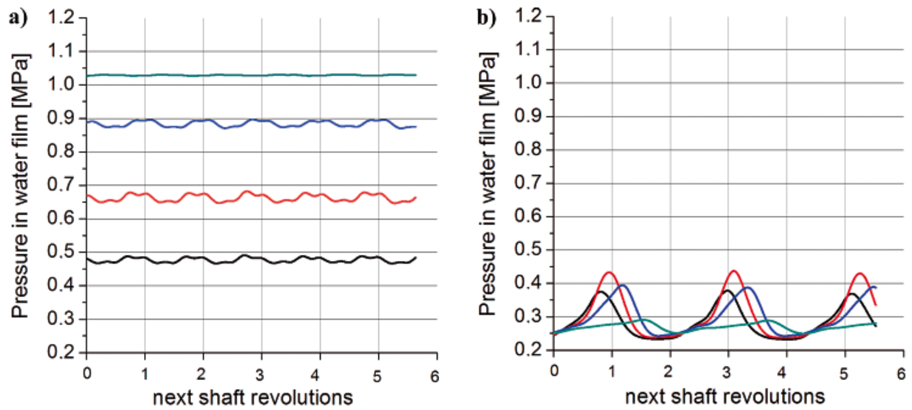

*Fig. 15. Comparison of the measured pressure pulsations in lubricating film; **a)** 0.6 MPa, 11rev/s, **b)** 0.2 MPa, 11rev/s*

CONCLUSIONS

- Analyzing the measurement results one can observe distinct difference between behaviour of the rigid (metal) bearing and flexible (polymer) one. Both the bearings generate hydrodynamic pressure distribution of different form. In the rigid bearing the maximum pressure is up to four times greater than the average nominal pressure, and in the flexible bearing – two times greater at the very most. In the metal bearing the starting moment over two times greater than that in the polymer bearing, occurs. Statical friction coefficients amount to 0.5 and 0.25, respectively. Worth adding that in the polymer bearing stick-slip phenomena were observed, that could detrimentally affect its friction coefficient value.
- Hydrodynamic pressure very fast develops and the friction coefficient value recorded during the tests for operation in the fluid friction range, amounted to 0.01, on average. It should be remembered however that due to limitations of the test stand the value is burdened with an error and evidently greatly overstated.
- The water-lubricated bearing with steel bush is of a greater load-carrying capacity in comparison with a typical polymer bearing. It results from lack of deformations of the rigid bush under hydrodynamic pressure as well as a high smoothness of its surface. This way the experimental tests confirmed the calculation results (Fig. 3).
- The bearings with steel bush show a lower sensitivity to contaminations which do not stick in sliding surface. Due to this fact wear occurs mostly on the side of bush, which is favourable from the point of view of repair cost.
- Apparently the necessity of application of hardened stainless steel journals was an obvious disadvantage of the metal bearings, which greatly rises cost of realization of such solution.

RECAPITULATION

The performed tests showed that in the assumed working conditions the water-lubricated bearings with metal bush are of a sufficient load-carrying capacity and can be applied also in the case of ship propeller shafts. Approval of the applied material by ship classification institutions remains to be a formal problem only.

The producer of the material indicates that one of the cardinal merits of such designs is their much greater resistance to wear in the conditions of lubrication with contaminated water since in the metal bearings, in contrast to polymer ones, the contaminations do not stick into bush sliding surface and, consequently, do not scratch the journal. As the referred opinion evokes some doubts, these authors have attempted to prepare tests of bearings of the kind in the conditions of lubrication with contaminated water. However the bearing with metal bush has an important disadvantage. It should be remembered that in the case of ship main propulsion shafts a problem is their appropriate assembling, and as a rule we then deal with a smaller or greater non-coaxiality between shaft and bush. It may also occur occasionally as a result of deformation of a flexible ship hull at its non-uniform weight distribution of cargo. The effect may also be strengthened due to hull response to waves. Therefore in some cases the so called edging, i.e. concentration of pressure between shaft journal and bush edge, may happen. The phenomenon can generate sudden wear process of the sliding pair. It will not occur in the case of applying elastic polymer because the bush will be deformed elastically, that leads to a greater contact zone.

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