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IMPROVING THE RELIABILITY AND STABILITY OF THE SLEEVE BEARING UNITS

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Keywords: sleeve bearing unit, ship shaftline, permanent magnetic field, magnet, centering.
Abstract. The problem of the reliability of the sleeve bearing units, including the ship ones, is described. A completely new design of the sleeve bearing unit for ship equipment based on the use of a powerful permanent magnetic field is presented. The technology of imposing a magnetic field on a sleeve bearing unit is described. Positive test results are presented: the increase in equipment efficiency, the decrease in the coefficient of friction and shaft vibration. The advantages of the invention and possible applications are enumerated.

ПОВЫШЕНИЕ НАДЕЖНОСТИ И СТАБИЛЬНОСТИ РАБОТЫ ПОДШИПНИКОВЫХ УЗЛОВ СКОЛЬЖЕНИЯ

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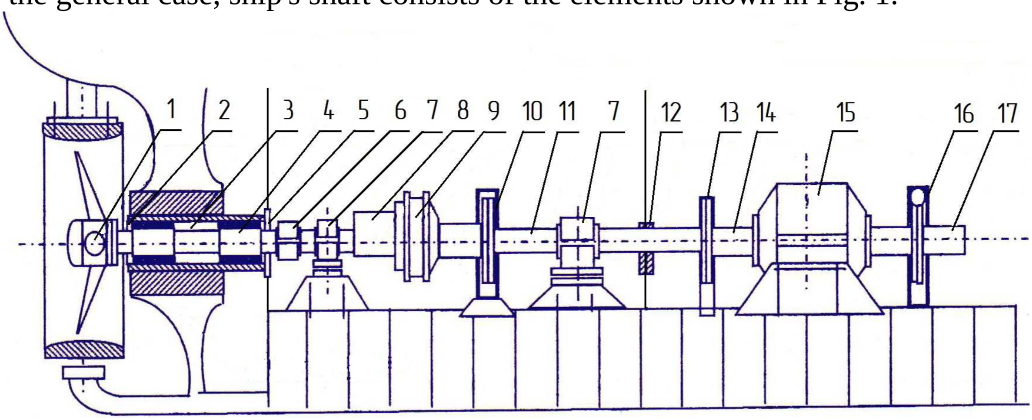
Ключевые слова: подшипниковый узел скольжения, судовой валопровод, постоянное магнитное поле, магнит, центровка.
Аннотация. Изложена проблема надежности подшипниковых узлов скольжения, в том числе и судовых. Представлена совершенно новая конструкция подшипникового узла скольжения для судовой техники на основе применения мощного постоянного магнитного поля. Описана технология наложения магнитного поля на подшипниковый узел скольжения. Представлены положительные результаты испытаний: увеличение КПД оборудования, снижение коэффициента трения и вибрации вала. Перечислены преимущества изобретения и возможные области применения.

Bearing units (BU) are the most important structural elements of machines and devices and constitute the main part of friction units. As a rule failures of techniques occur due to failures of BU (along with failures of other friction units), which, thus, limit the durability of machines and devices. Even with a sufficiently high-quality production of BU components, for example, an bushing, the characteristics of BU may be unsatisfactory, and a sudden failure will occur. In this case, failure should not be understood as the destruction of rubbing (working) surfaces, but the exit of one of the characteristics of BU beyond the permissible limits. Thus, for BUs used in control systems, the failure criteria can be: an increase in instability of the moment of resistance to rotor rotation, an excessively large and non-permanent displacement of the center of mass of the rotor, instability of the frequency and amplitude of the radial and axial beats of the rotor at some frequencies. For spindle BUs, typical failure criteria are low accuracy of rotation, increased vibration, low rigidity, high moment of resistance to rotation, and thermal jamming.

The most common criterion for failure of a general purpose BU is fatigue failure. At the same time, other characteristics, such as stiffness, level and spectrum of vibration, the moment of resistance to rotation, durability, are important for special use BU.

Thus, firstly, the durability of BU does not always coincide with the cyclic durability of the working surfaces of the bearing, and often turns out to be much smaller and even zero, as, for example, for BUs that do not have the required characteristics when installed in the product. Secondly, sufficient durability of individual parts of BU, tested on the stands, does not guarantee sufficient durability of BU. The latter circumstance is due to the fact that the loads acting in the node, as well as the temperatures may differ significantly from the bench ones. In addition, the assembly and installation change the gaps, tension and shape of the working surfaces of BU [1].

The BUs used in technique is complex mechanical systems. Among the most complex bearing assemblies are the ship shaft bearing assemblies. As a rule, the ship's shaft includes several shafts connected in series and several bearing supports. In the general case, ship's shaft consists of the elements shown in Fig. 1.



1 – propeller screw; 2 – stern tube gasket; 3 – stern tube device; 4 – rowing shaft; 5 – nose stern gasket; 6 – oil axle-bearing; 7 – basic bearing; 8 – half-coupling a rower shaft; 9 – step change mechanism; 10 – braking device; 11 – intermediate shaft; 12 – bulkhead gasket; 13 – current collector; 14 – thrust shaft; 15 – thrust bearing; 16 – shaft rotation device; 17 – shaft of the main engine

Fig. 1. Ship shaft structure

In the static state, the shaft is subjected to the action of distributed and concentrated loads from the gravity of the shafts and hung components and parts (flanges, pulleys, etc.) The loading of the shafts and supports also depends on the installation curvatures of the shafting during alignment due to the elimination of misalignment.

Under operating conditions, the shafting is additionally affected by a system of loads associated with the operation of the main engine, propeller and hull deformation due to changes in the vessel’s loading and meteorological environmental conditions.

Operational loadings include:

- torque from the main engine;
- the force of an emphasis created by the rowing screw;
- forces and moments of hydrodynamic character on the propeller from the oncoming flow of water;

- reactions on the supports and bending moments in the shafts, arising in connection with the general bending of the hull of the vessel from changes in its loading, air and water temperature, and sea excitement;

- reactions on supports and bending moments in shafts, associated with wear of rubbing bearing pairs;

- periodic loadings arising from static unbalance of rotating parts and assemblies;

- inertial forces transmitted from shafts and hung parts and assemblies during transverse and longitudinal pitching of the vessel;

- dynamic loadings transmitted from shafts and hung parts and assemblies in a collision of a vessel with any obstacles (during mooring, grounding, ice collisions, etc.);

- periodic loadings arising from inaccuracies in the assembly of shaft flanges (i.e., misalignment of the assembled shaft) or compensating misalignment on flexible (elastic) couplings [2].

All of the above loadings in the end negatively affect the alignment of the shafting, which in turn reduces the reliability and stability of the sleeve bearing units.

Sleeve bearings that use the oil wedge effect are called hydrodynamic. The fundamentals of hydrodynamic theory were developed by eminent scientists: N.P. Petrov, O. Reynolds, N.E. Zhukovsky, S.A. Chaplygin.

Fig. 2 shows a diagram of the formation of an oil wedge under the conditions of hydrodynamic lubrication of a bearing assembly. Between the inner surface of the bearing housing (bushing) with a diameter *D* and a trunnion with a diameter *d*, there is a gap filled with a lubricant. When the shaft is rotation from the state of rest, an oil wedge is formed, which, due to the increased pressure in it, has a bearing capacity and pushes the loaded rotating shaft trunnion upward and somewhat in the direction of its rotation.

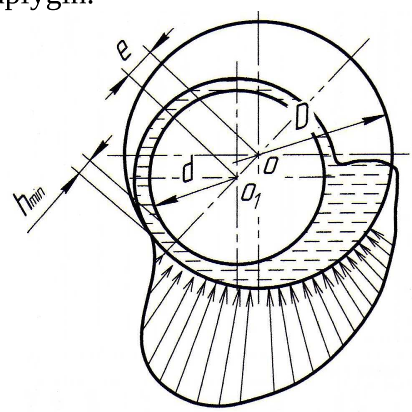
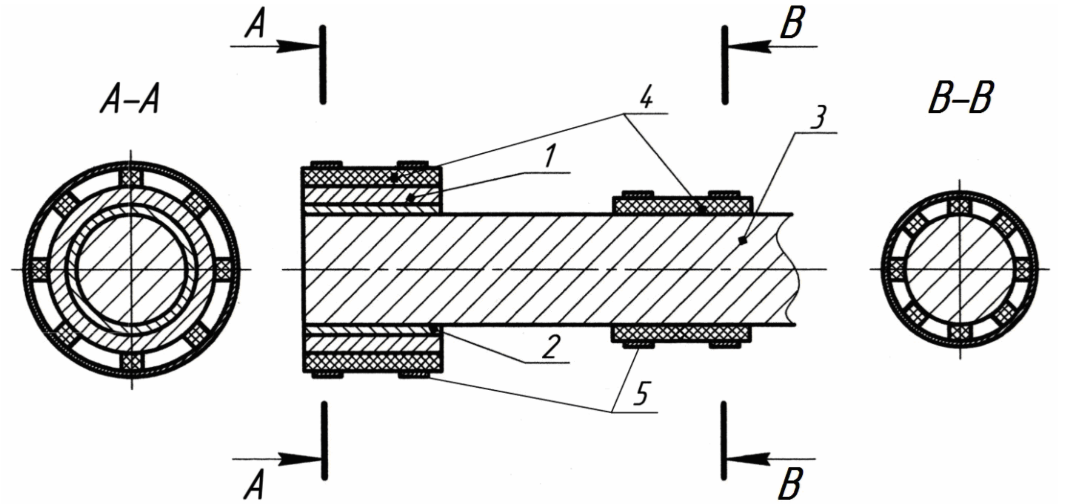


Fig. 2. Scheme of formation of the oil wedge and the distribution of hydrodynamic pressure in the sleeve bearing unit

The center of the shaft trunnion is displaced to the point O_1 at a distance *e* from the center of the bearing *O*. As a result, a minimum clearance h_{min} is formed, which ensures constant lubrication of the entire surface of the shaft trunnion. The movement of the center of the trunnion continues until the resultant of the pressure forces and the internal friction forces balance the internal loading. At high rotational frequencies and low loadings, the center of the trunnion O_1 can theoretically coincide with the center of the bearing *O*.

For a quicker and more accurate installation of the shaft trunnion in the center of the bearing, the authors have developed a completely new design of the sleeve bearing unit for the ship shaft [3], shown in Fig. 3.



1 – bearing housing; 2 – annular insert of non-magnetic materials; 3 – shaft; 4 – permanent linear magnets; 5 – clamps

Fig. 3. Sleeve bearing unit structure

Sleeve bearing unit consists of a bearing housing, a shaft and located between them an annular insert of non-magnetic materials. On the outer surface of the bearing housing and on the shaft are installed in the longitudinal direction and fixed by band clamps powerful permanent linear magnets with the same poles.

The invention works as follows: the rowing shaft 3 is driven by an electric motor (not shown in Fig. 3). When the rowing shaft 3 rotates in the bearing housing 1, the magnetic field created by the powerful permanent magnets 4 corrects the coaxiality of the position of the shaft 3. When the rowing shaft 3 increases in rotation under the action of the hydrodynamic pressure of the lubricant, a hydrodynamic wedge arises, promoting the ascent of the shaft 3 of which directs to the centre the bearing housing 1.

The new invention was tested on test equipment in the laboratory of Astrakhan State Technical University. During the tests, the work of the ship propulsion system was simulated. A general view of the equipment with installed magnets is shown in Fig. 4. Schematic diagram of the test facility is shown in Fig. 5.

This equipment can be tested at various speeds of rotation of the shaft in the range of 0-1200 rpm. The equipment is equipped with a gearbox and a loading device, allowing to apply various loads on the shaft in the range of 0-35 N·m.

In the experimental equipment there is the possibility of displaying some performance indicators on a personal computer screen, such as:

- indication of current readings of torque and load moments (M_{in} and M_{out}, respectively);
- the value of the current speed;
- thrust coefficient;
- slip coefficient and efficiency.

Also features graphing and data storage in tabular form are foreseen.

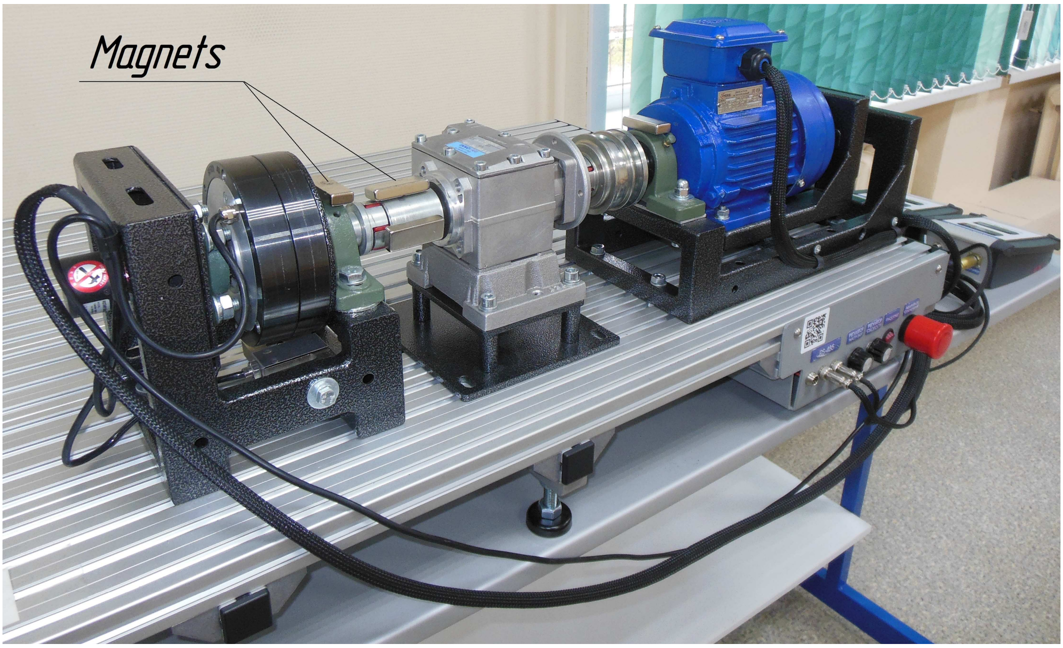
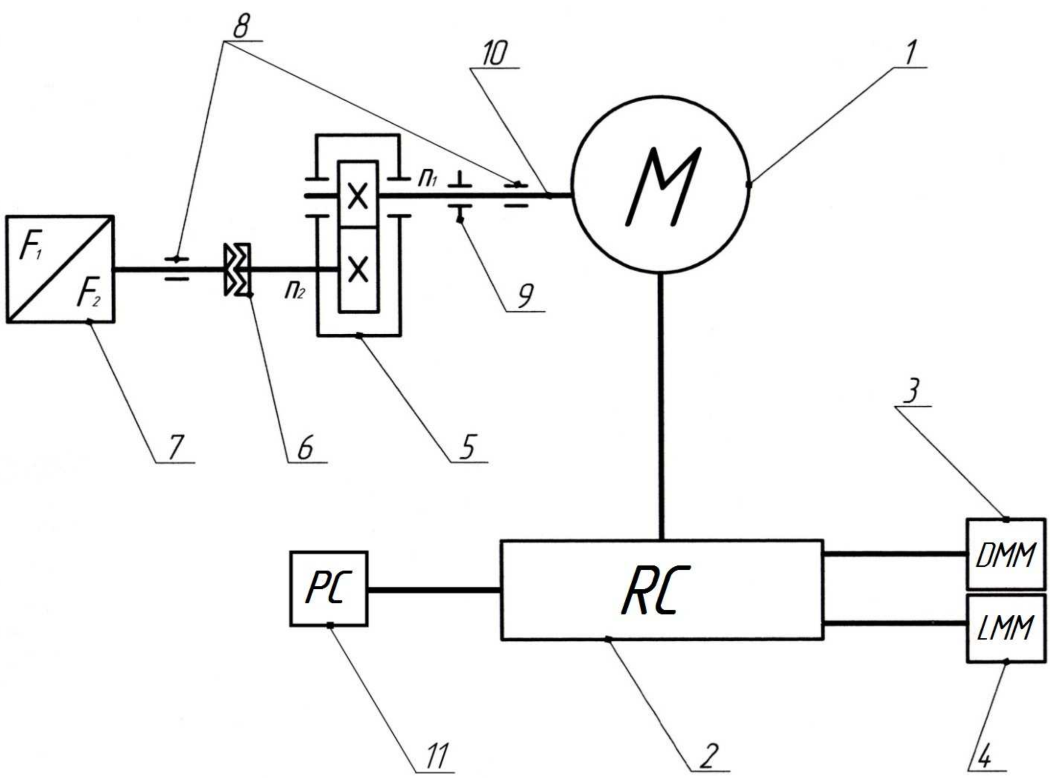


Fig. 4. General view of the test equipment with installed magnets



1 – electric motor; 2 – control panel; 3 – drive measurement module; 4 – load measurement module; 5 – reducer; 6 – coupling; 7 – load device; 8 – sleeve bearings; 9 – pulley; 10 – shaft; 11 – personal computer

Fig. 5. Schematic diagram of the test equipment

The drive is carried out by the electric motor of the model AIR63V4 y2. The main technical characteristics of the engine of this brand can be found in [4].

In the course of the experiments, powerful linear neodymium magnets of the brand N50 were used. The image of these magnets is shown in Fig. 6. Such magnets are the most powerful ones and have high coercive force (more than 850 kA/m) and magnetic energy (about 400 kJ/m^3). Characteristics of existing brands of neodymium magnets are presented in table. 1 [5].

Fig. 6. Permanent neodymium magnets N50

Tab. 1. Characteristics of neodymium magnets

Brand/ Class	**Residual magnetic induction, milliTesla (kiloGauss)**	**Coercive force, kiloAmper/meter (kiloErsted)**	**Magnetic energy, kiloJoule/m^3 (MegaGauss-Oersted)**	**Operating temperature, degree Celsius**
N35	1170-1220 (11,7-12,2)	≥955 (≥12)	263-287 (33-36)	80
N38	1220-1250 (12,2-12,5)	≥955 (≥12)	287-310 (36-39)	80
N40	1250-1280 (12,5-12,8)	≥955 (≥12)	302-326 (38-41)	80
N42	1280-1320 (12,8-13,2)	≥955 (≥12)	318-342 (40-43)	80
N45	1320-1380 (13,2-13,8)	≥955 (≥12)	342-366 (43-46)	80
N48	1380-1420 (13,8-14,2)	≥876 (≥12)	366-390 (46-49)	80
N50	1400-1450 (14,0-14,5)	≥876 (≥12)	382-406 (48-51)	80
N52	1430-1480 (14,3-14,8)	≥876 (≥12)	398-422 (50-53)	80
33M	1130-1170 (11,3-11,7)	≥1114 (≥14)	247-263 (31-33)	100
35M	1170-1220 (11,7-12,2)	≥1114 (≥14)	263-287 (33-36)	100
38M	1220-1250 (12,2-12,5)	≥1114 (≥14)	287-310 (36-39)	100
40M	1250-1280 (12,5-12,8)	≥1114 (≥14)	302-326 (38-41)	100
42M	1280-1320 (12,8-13,2)	≥1114 (≥14)	318-342 (40-43)	100
45M	1320-1380 (13,2-13,8)	≥1114 (≥14)	342-366 (43-46)	100
48M	1380-1420 (13,8-14,3)	≥1114 (≥14)	366-390 (46-49)	100

Brand/ Class	Residual magnetic induction, milliTesla (kiloGauss)	Coercive force, kiloAmper/meter (kiloErsted)	Magnetic energy, kiloJoule/m^3 (MegaGauss-Oersted)	Operating temperature, degree Celsius
50M	1400-1450 (14,0-14,5)	≥1114 (≥14)	382-406 (48-51)	100
30H	1080-1130 (10,8-11,3)	≥1353 (≥17)	223-247 (28-31)	120
33H	1130-1170 (11,3-11,7)	≥1353 (≥17)	247-271 (31-34)	120
35H	1170-1220 (11,7-12,2)	≥1353 (≥17)	263-287 (33-36)	120
38H	1220-1250 (12,2-12,5)	≥1353 (≥17)	287-310 (36-39)	120
40H	1250-1280 (12,5-12,8)	≥1353 (≥17)	302-326 (38-41)	120
42H	1280-1320 (12,8-13,2)	≥1353 (≥17)	318-342 (40-43)	120
45H	1320-1380 (13,2-13,8)	≥1353 (≥17)	326-358 (43-46)	120
48H	1380-1420 (13,8-14,3)	≥1353 (≥17)	366-390 (46-49)	120
30SH	1080-1130 (10,8-11,3)	≥1592 (≥20)	233-247 (28-31)	150
33SH	1130-1170 (11,3-11,7)	≥1592 (≥20)	247-271 (31-34)	150
35SH	1170-1220 (11,7-12,2)	≥1592 (≥20)	263-287 (33-36)	150
38SH	1220-1250 (12,2-12,5)	≥1592 (≥20)	287-310 (36-39)	150
40SH	1240-1280 (12,4-12,8)	≥1592 (≥20)	302-326 (38-41)	150
42SH	1280-1320 (12,8-13,2)	≥1592 (≥20)	318-342 (40-43)	150
45SH	1320-1380 (13,2-13,8)	≥1592 (≥20)	342-366 (43-46)	150
28UH	1020-1080 (10,2-10,8)	≥1990 (≥25)	207-231 (26-29)	180
30UH	1080-1130 (10,8-11,3)	≥1990 (≥25)	223-247 (28-31)	180
33UH	1130-1170 (11,3-11,7)	≥1990 (≥25)	247-271 (31-34)	180
35UH	1180-1220 (11,7-12,2)	≥1990 (≥25)	263-287 (33-36)	180
38UH	1220-1250 (12,2-12,5)	≥1990 (≥25)	287-310 (36-39)	180
40UH	1240-1280 (12,4-12,8)	≥1990 (≥25)	302-326 (38-41)	180
28EH	1040-1090 (10,4-10,9)	≥2388 (≥30)	207-231 (26-29)	200
30EH	1080-1130 (10,8-11,3)	≥2388 (≥30)	233-247 (28-31)	200
33EH	1130-1170 (11,3-11,7)	≥2388 (≥30)	247-271 (31-34)	200
35EH	1170-1220 (11,7-12,2)	≥2388 (≥30)	263-287 (33-36)	200
38EH	1220-1250 (12,2-12,5)	≥2388 (≥30)	287-310 (36-39)	200

The magnetization (direction of the magnetic field) is axial, i.e. the magnetization vector passes through the thickness of the magnets. The direction of magnetization is shown in Fig. 7.

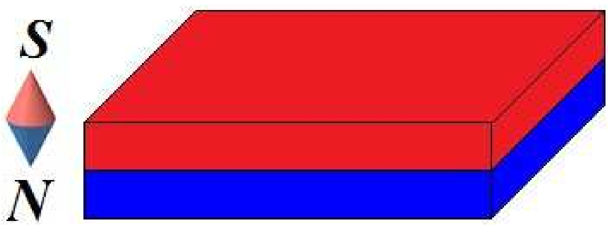


Fig. 7. Direction of magnetization of magnets

To install the magnets on the sleeve bearing unit in the correct (identical) direction of the poles, as well as measuring the residual magnetic induction force in different parts of the bearing unit, an ATE-6702 magnetic induction meter (magnetometer) was used (Fig. 8).

Measuring sensor of a magnetometer based on a Hall sensor with automatic temperature compensation. The ATE8702 magnetic induction meter allows measurements of the parameters of constant and variable (frequency 50 Hz/60 Hz) magnetic fields up to 30000 Gs or 3000 mTs. It is possible to determine the polarity using the built-in N-field and S-field indicator. The functional of the magnetometer is presented in Fig. 9. A more detailed description of it can be found in [6].

Fig. 8. Digital magnetic induction meter (magnetometer) ATE-8702

As a result of the experiments, the authors obtained positive results - an increase in the efficiency of the installation by several percent (an average gain of 2-3%), a decrease in the coefficient of friction and shaft vibration. Tab. 2 presents the numerical results of experiments with the modes of rotation of the shaft 500 and 800 rpm. Fig. 10, 11 show the graphs of the installation efficiency change at shaft speeds of 500 and 800 rpm, respectively.

Fig. 9. Functional magnetometer ATE-8702

Tab. 2. The results of measurements of efficiency at different speeds of rotation of the shaft

M_{out}, N·m	η, %			
	at 500 rpm		at 800 rpm	
	without magnetic field	with magnetic field	without magnetic field	with magnetic field
0,5	6,11	14,37	10,11	12,39
1,5	52,65	52,03	53,98	56,79
2,5	64,43	69,08	64,48	64,9
3,5	72,14	72,18	66,51	66,66
4,5	73,13	75,13	72	75,2
5,5	77,42	81,49	76,5	76,49
6,5	80,66	81,68	74,94	79,29
7,5	79,88	81,81	83,82	82,12
8,5	83,79	82,73	77,89	81,63
9,5	83,12	84,59	83,48	83,44
10,5	83,56	84,51	82,69	85,76
11,5	84,36	85,85	85,35	85,91
12,5	85,2	84,15	83,04	86,02
13,5	85,18	87,78	84,73	84,61
14,5	84,91	88,78	85,43	85,95
15,5	87,36	90,83	87,36	85,84
16,5	88,89		86,45	86,74
17,5			88,53	86,69
18,5			85,12	87,56
19,5			87,8	85,5
20,5			87,27	87,49
21,5			87,99	90,68
22,5			90,89	

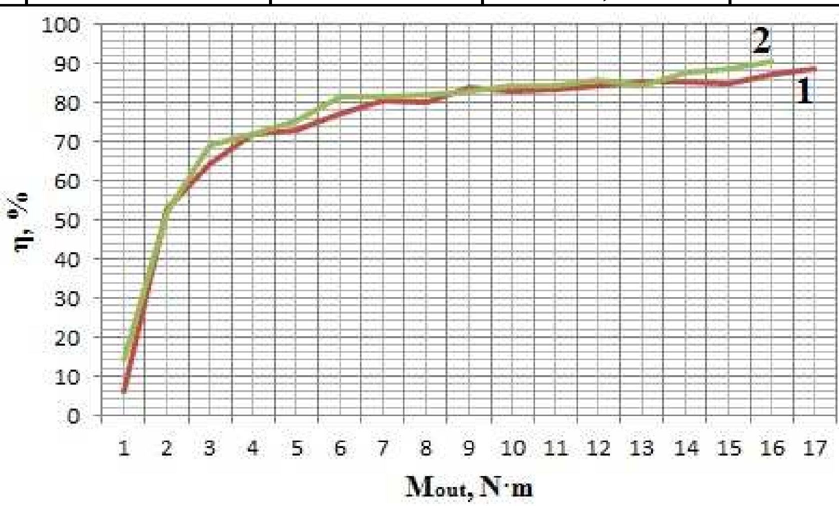


Fig. 10. Efficiency graph at 500 rpm:
1 – without magnetic field; 2 – with magnetic field

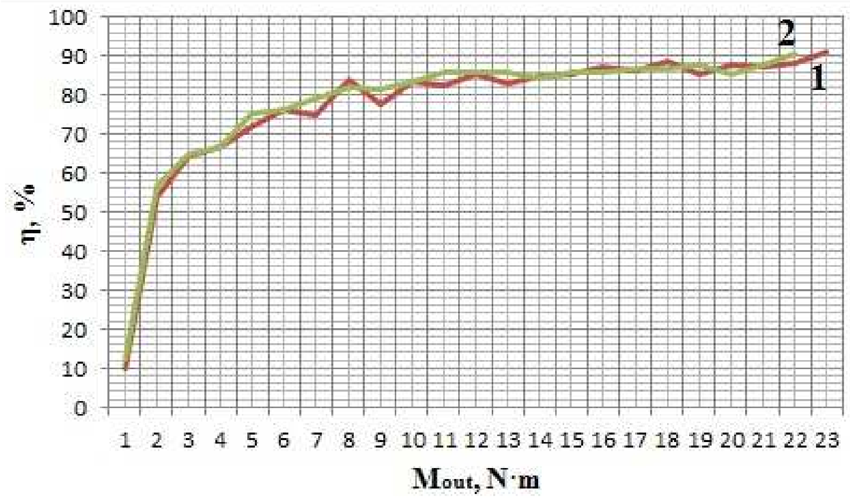


Fig. 11. Efficiency graph at 800 rpm:
1 – without magnetic field; 2 – with magnetic field

Key conclusions

1. A completely new structure of the sleeve bearing unit has been developed for ship equipment.

2. This invention has high operational technical and economic indicators, such as:

- reliability, durability and stability of the equipment;
- simplicity of design, ease of maintenance and high maintainability;
- relatively low cost;
- faster and more accurate shaft alignment;
- increase in equipment efficiency, reduction of friction coefficient and shaft vibration.

3. The invention can find application not only in ship engineering, but also in many other branches of mechanical engineering and national economy.

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