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REVEALING THE INFLUENCE OF STRUCTURAL AND OPERATIONAL PARAMETERS OF A WOODEN HYDROSTATIC BEARING ON ITS PERFORMANCE

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Wooden bearings, lubricated with water and other low-viscosity lubricants, have been used for centuries. In steamship engines, turbines of hydroelectric power plants and propellers of submarines, it is proposed to use wooden hydrostatic bearings, which are guaranteed to provide liquid friction and have high reliability.

The object of this study is hydrostatic and thermal processes in sliding supports with wooden hydrostatic bearings. The influence of the design and operational parameters of a wooden hydrostatic bearing on its performance was tackled. Theoretical dependences are given, making it possible to determine the bearing capacity and temperature increase in the hydrostatic bearing. Quantitative values of specific pressures on the working surface of the bearing and an increase in the temperature of the working fluid due to friction and pumping losses were established.

It was found that the specific pressures of the liquid do not exceed the tensile strength of the bearing material. The reduction in load capacity due to an increase in the temperature of the working fluid was 10.95 %, owing to the small change in the temperature of the working fluid in hydrostatic bearings. The results of the study make it possible to establish the permissible boundaries for the purpose of the design and operational parameters of the hydrostatic bearing ensuring its operability.

For the effective use of hydrostatic wooden bearings at the listed facilities, it is necessary to have a system for supplying lubricant to the bearing under high pressure. The system should include filters to clean the lubricant from impurities. The theoretical dependences built allow designers of hydrostatic bearings to use them in practical calculations

Keywords: wooden bearing, specific pressure, thermal regime, bearing capacity, operational parameters

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1. Introduction

The performance of any machine is influenced by a large number of factors. One of them is the reliability and durability of the supports of the rotors of the machines. There are a large number of types of bearings that are used depending on the operating conditions, the nature of the load, and other factors. The work considers the analysis of the operability of wooden hydrostatic bearings. Wooden bearings have been used for centuries and are applied in machines in which it was desirable or inevitable to use water as a lubricant. They successfully compete with metal bearings. The durability of a wooden bearing is 4 to10 times that of a bronze bearing. A wooden part is 8 to 10 times lighter than an iron bearing. Wooden bearings made of lignum vitae emit natural grease for 20 years. Not every metal bearing will be able to have such a resource. On the world's first nuclear submarine, USS Nautilus (SSN-571) wooden assemblies were used in the bearings. The turbine bearings of the Conowingo hydroelectric power plant on the Susquehanna River were also made of lignum vitae.

The cost of lignum vitae is high from 30 to 80 thousand dollars per cubic meter. Therefore, compacted wood of poplar, aspen, alder is widely used, which, after pressing, has a strength of 8–10 times higher compared to natural. The manufacture of bushings and liners of plain bearings is one of the uses of wood modified by a seal. The liners of plain bearings made of modified wood work reliably in abrasive, aqueous, and aggressive environments. The cost of compacted wood as a material for the manufacture of wood bushings of friction units are 5 times lower than the cost of steel, bronze – 20 times. Oil-soaked compacted wood makes it difficult for moisture to penetrate it.

A number of advantages of wooden bearings, combined with the advantages of hydrostatics, indicate the need for their use as shaft supports of various machines.

The lack of information on the design of these bearings renders relevance to research into this area.

2. Literature review and problem statement

Paper [1] examines the wear resistance of a wooden plain bearing. To assess wear resistance, a test bench based on a microcontroller was designed and manufactured. Bearings manufactured along the tangential orientation of wood have been found to have better wear resistance than bearings from other orientations. The rate of wear was nonlinear. However, that study did not look at ways to more significantly reduce the wear resistance and improve the reliability and durability of a wooden plain bearing. The tests were limited. Widely used compacted woods of poplar, aspen, alder were not tested. Work [2] examines in historical terms the different types of bridge bearings. The article points out that wooden bushings were used in bridge bearings. In the historical analysis of bearing types, the designs of wearless hydrostatic wooden bearings, which can significantly increase the reliability and durability of machine support units, were not considered. The issues of force interaction of the working fluid and the bearing support surface were not considered. The various materials used in the construction of wooden bearings were not analyzed. Paper [3] discusses the dynamic characteristics of multilayer plain bearings lubricated with silicone lubricants. It is noted that the wooden house, which weighs 40 tons, is supported by 16 plain bearings. By August 2002, 500 homes had been built in Japan using this system. A limitation of the considered calculation methodology is the consideration of one of the varieties of silicone oils. Silicones include a whole class of organosilicon compounds. Silicone has good anti-wear properties and can be used in a wide range of temperatures. However, it is used for lightly loaded bearings, and the above study looks at wooden houses that weigh 40 tons. Paper [4] describes the design of a self-lubricating plain bearing made of modified wood for heavily loaded friction units and the technology of its manufacture. A method for determining the coefficient of friction of wood on steel has been developed and the MI-1M friction machine has been modernized. The sliding friction coefficient varies between 0.06 and 0.11 depending on the magnitude of the load. The above design of the self-lubricating bearing has a short service life and complex manufacturing technology. Most often self-lubricating sliding supports with porous bearings impregnated with oil are used. The amount of oil in the supports of such bearings, as a rule, is not replenished during the entire period of operation of the product. In existing designs of self-lubricating supports the efficiency of the lubrication system is not yet great. The paper does not consider more efficient hydrostatic bearings that provide guaranteed liquid friction. In work [5], plain bearings made of modified pressed wood were experimentally tested, showing that a high probability of their failure occurs when the surface hardness of the counter-body is less than 50 HRC. Rollers made of steel 45, 45 X, 18 X GT were used as a counter-body. However, in that study, metal-ceramic hard alloys with significant advantages were not considered as a counter-body material although they have advantages in strength and wear resistance. It considered only the contact of solids of wood and steel and did not take into account the influence of the lubricant on the friction process in the bearing. When the shaft rotates in the bearing, hydrodynamic phenomena also occur in the wedge space. In work [6] thermally softened wood was studied under the influence of combined stresses. The bearing was tested using a thermohygrostat test machine. Crumple strength and stiffness decreased with increasing temperature and thermal softening. The reduction in both strength and stiffness in the longitudinal direction was greater than in the transverse direction. The tests simulated operating conditions, such as the presence of abrasive dust. However, that study did not take into account the change in bearing dimensions with increasing temperature and the effect of changing bearing dimensions on bearing capacity. The change in the characteristics of the lubricant with increasing temperature and the impact of this change on the bearing characteristics were not taken into account. Work [7] analyzes the strength of wooden elements. The article presents the possibility of increasing the bearing capacity of wooden structures due to special methods of wood modification or layer-by-layer reinforcement. The amorphous part of the cellulose structure provides the plastic character of the compression deformation. The article discusses various ways to modify wood. The considered method of increasing the bearing capacity of a wooden bearing is ineffective. More effective is the use of hydrostatic wooden bearings, which use both hydrostatic and hydrodynamic effects to significantly increase the bearing capacity of the bearing. Work [8] considers the improvement of the work of wooden plain bearings. That pilot study examined a number of factors that influence the characteristics of wooden bearings, including wood properties, manufacturing methods, and operating conditions. A wooden bearing test procedure was developed, and a testing machine was manufactured. The results of the study showed that the characteristics of the lubricants, the speed of the workload, as well as the density and permeability of the wood can have a significant impact on the operation of the bearing. However, that study did not take into account the effect of shaft rotation speed on the appearance of hydrodynamic phenomena in the lubricant layer, which significantly affect the characteristics of the bearing. Operating conditions were selective and limited.

Work [9] considers the influence of lignum vitae anisotropy on its tribological properties. The wear process went through two stages due to the anisotropy of the lignum vitae cell fibers. The fibers of the cells were first subjected to the process of separation, and then to the processes of impact and bending. Due to the low mechanical properties of lignum vitae in the vertical direction of the fibers, the fibers and cells were easily destroyed, which led to severe wear, higher coefficients of friction and wear rate. In that study, a very limited number of factors affecting the wear rate and friction coefficient were considered. No attention is paid to the influence of different types of lubricant on the wear process. During the wear process, the temperature in the friction zone increases, the influence of which was also not taken into account when analyzing the operation of the bearing. Work [10] shows the use of wood as a material for automotive spare parts. The low strength of wood was a limitation of its application for plain bearings. In that work, the wood was impregnated with polyethylene glycol, lubricating oil, and epoxy to improve mechanical properties and tribological characteristics. Based on the results of the tests, it was found that polyethylene glycol is a promising material for improving the physical properties of wood for plain bearings. The above study does not pay attention to the automatic supply of lubricant from the pore channels and its effect on the tribological characteristics of the bearing. The polyethylene glycol recommended in operation as a promising material for plain bearings is questionable since its fluidity properties are significantly dependent on temperature. In work [11], the bearing malfunctions are analyzed. The vibration signal contains information about the state of rotating mechanisms and makes it possible to analyze the intensity of noise. An effective strategy for diagnosing bearing failures is proposed, involving three tests. The proposed strategy for detecting bearing faults proved to be more effective than existing techniques. However, the proposed technique for diagnosing bearing failures is very limited and can only be applied to certain types of plain bearings. When diagnosing the operation of the bearing, no attention is paid to the dynamic phenomena that occur during the operation of the bearing due to the presence of an imbalance in the rotor.

Without dynamic evaluation, modern high-speed machines cannot be recognized as fully operational. In work [12], on the basis of experimental results, the influence of the anatomical properties of wood on its tribological properties is analyzed in detail. The results showed that the tribological properties of wood are greatly influenced by its anatomical properties. A sufficient amount of extract substances can significantly improve the tribological properties of wood by reducing the coefficient of friction and improving the washing capacity. The above study does not pay attention to the different types of extract substances and their various lubricating properties. Friction between the lubrication layers during shaft rotation inside the bearing, which can be significant at high rotational speeds, is also not taken into account. The paper does not offer a cardinal way to solve the problem of wear and tear.

In works [1–12] there is no information on the design of wooden bearings of hydrostatic type and no determination of specific pressures of the working fluid on the surface of the bearing, and there is no thermal analysis of wooden hydrostatic bearings.

3. The aim and objectives of the study

The purpose of this work is to identify the impact of changes in the operating parameters of a wooden hydrostatic bearing on its performance. This makes it possible to establish permissible boundaries for the purpose of the structural and operational parameters of the bearing ensuring its operability.

To accomplish the aim, the following tasks have been set:

– to identify the magnitude of the effect of the supply pressure of the working fluid on the specific pressure of this liquid on the working surface of the wooden bearing;

– to establish the effect of increasing the temperature of the working fluid, as a result of friction and pumping losses, on the bearing capacity of a wooden hydrostatic bearing.

4. The study materials and methods

The object of this study is hydrostatic and thermal processes in sliding supports with wooden hydrostatic bearings. It is planned to construct a mathematical model that makes it possible to study the pressure of the working fluid on the surface of the bearing and its thermal regime. Assumptions were accepted that the pressure gradient in terms of the thickness of the working fluid is small in comparison with the pressure gradient in other directions and the inertial terms of the Navier-Stokes equations are small in comparison with viscous ones. The external load acting on the bearing was taken to be constant.

In the development of theoretical dependences, hydromechanics methods were used to calculate the characteristics of a wooden hydrostatic bearing. The characteristics of the bearing were determined on the basis of the function of pressure distribution in the working fluid layer, the loss was calculated from the joint solution of the Reynolds equations and the balance of working fluid flows. When determining the pressures in the chambers, an iterative method was used until the specified accuracy was obtained. The pressure of the working fluid on the inter-chamber lintels was determined using the Reynolds equation, which was solved numerically using the method of finite differences in combination with the method of longitudinal-transverse sweep. The carrying capacity of the bearing was determined using the trapezoidal method. When determining the specific pressures of the working fluid on the working surface of the bearing, material resistance methods were used. To assess the thermal mode of operation of the bearing, an increase in the lubrication temperature as a result of friction losses and pumping of the working fluid was determined.

The numerical implementation of the built mathematical dependences was performed in the software «Excel» (developer «Microsoft», USA). Drawings of the results obtained were performed in the graphic editor «Compass».

The diagram of the wooden hydrostatic bearing in question is shown in Fig. 1.

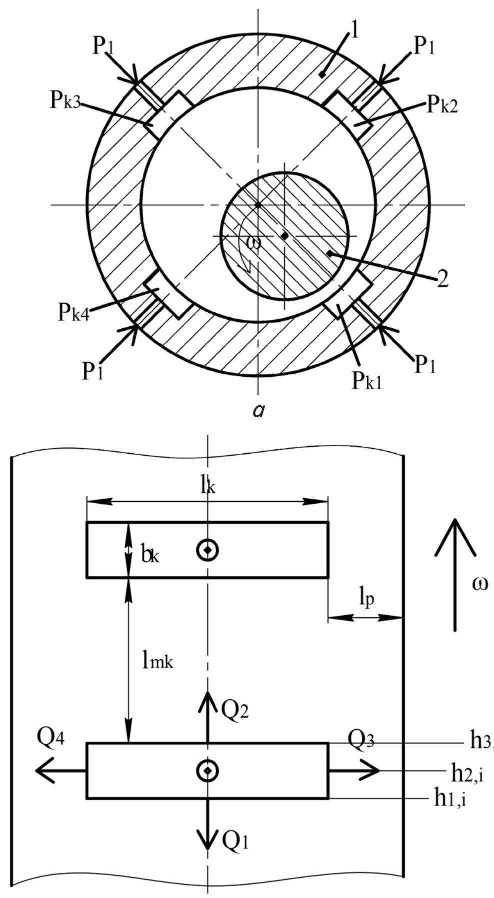

Fig. 1. Diagram of a wooden hydrostatic bearing: *a* – general view of the bearing; *b* – working surface with rectangular bearing chambers

On the working surface of bearing 1 there are four rectangular bearing chambers into which the working fluid is supplied under high pressure P_1. Shaft 2 rotates inside the bearing with an angular velocity ω. To ensure the required bearing capacity, the pressure value P_1 is determined by calculation. Hydrostatic pressure occurs between the bearing working surfaces and the shaft surface, which provides the necessary bearing capacity and causes specific pressure in the bearing material.

To determine the bearing capacity of a wooden hydrostatic bearing, it is necessary to have a pressure distribution function in the working fluid layer. Its determination is associated with the joint solution of the Reynolds equations and the balance of flow of the working fluid. The pressures in the chambers are determined from the solution of the flow balance equation of the working fluid. The balance of working fluid costs is recorded from the condition of equality of costs through input compensating devices (nozzles) and costs along the contour of the i-th chamber (Fig. 1).

Let's record the balance of the working fluid for the i-th chamber:

$$Q_{11} = Q_1 + Q_2 + Q_3 + Q_4, \tag{1}$$

where Q_{11} is the flow rate of the working fluid through the inlet compensating device; Q_1, Q_2, Q_3, Q_4 are the flow rates of the working fluid along the contour of the i-th chamber.

The flow rate through the input compensating device will be recorded using the well-known formula of hydraulics [13, 14]:

$$Q_{11} = \psi_1 \cdot \pi \cdot r_1^2 \sqrt{\frac{2}{\rho}\left(P_1 - P_{ki}\right)}, \tag{2}$$

where ψ_1 – an input coefficient of 0.62...0.82; r_1 – nozzle radius; ρ – density of the working fluid; P_1 is the supply pressure at the inlet to the nozzle; P_{ki} is the pressure in the i-th bearing chamber.

When recording flow rates along the contour of the i-th chamber, both the portable and gradient flow of the working fluid were taken into account.

Let's write down the expression for the balance of the working fluid in a dimensionless form:

$$\left(P_{k,i}\right)_{n+1} = a_{17,i}\sqrt{1-\left(P_{k,i}\right)_n} + a_{14,i} \cdot P_{k,i-1} + a_{15,i} \cdot P_{k,i+1} + a_{18,i}, \tag{3}$$

where

$$a_{14,i} = -\frac{a_{9,i}}{a_{8,i}},\ a_{13,i} = -\frac{1}{a_{8,i}},\ a_{12,i} = a_{7,i} + a_{11,i},$$

$$a_{11,i} = \frac{2 \cdot a_{6,i} \cdot P_2}{a_{1,i}},\ a_{10,i} = -\frac{a_{5,i}}{a_{1,i}},\ a_{9,i} = -\frac{a_{3,i}}{a_{1,i}},$$

$$a_{8,i} = \frac{a_{3,i} + a_{5,i} + 2 \cdot a_{6,i}}{a_{1,i}},\ a_{7,i} = \frac{-a_{2,i} + a_{4,i}}{a_{1,i}},$$

$$a_{6,i} = \frac{h_{2,i}^3 \cdot b_k}{12 \cdot \mu \cdot l_n},\ a_{5,i} = \frac{h_{1,i}^3 \cdot l_k}{12 \cdot \mu \cdot l_{tk}},$$

$$a_{4,i} = \frac{\omega \cdot R \cdot h_{3,i} \cdot l_k}{2}, a_{3,i} = \frac{h_{1,i}^3 \cdot l_k}{12 \cdot \mu \cdot l_{tk}},$$

$$a_{2,i} = \frac{\omega \cdot R \cdot h_{1,i} \cdot l_k}{2},\ a_{1,i} = \psi_1 \cdot \pi \cdot r_m^2 \sqrt{\frac{2}{\rho}},$$

where ω – the angular velocity of rotation of the shaft; R – radius of the working surface of the bearing; $h_{1,i}$, $h_{2,i}$, $h_{3,i}$ – clearances in the bearing; b_k – the width of the chambers; l_p – length of the end partitions of the chambers; l_{tk} – the length of the inter-chamber partitions of the bearing; μ – dynamic viscosity of the working fluid; l_k – length of chambers; P_2 is the pressure of the working fluid on the drain.

Setting the initial values of the pressures in the chambers using (3), I obtain the pressure in the chambers at n+1 step.

The iterative process continues until the specified accuracy is obtained.

To obtain the pressures on the inter-chamber partitions, the Reynolds equation was applied [15]:

$$\frac{\partial}{\partial x}\left(\frac{h^3 \cdot \partial P}{\partial x}\right) + \frac{\partial}{\partial z}\left(\frac{h^3 \cdot \partial P}{\partial z}\right) = 6 \cdot \mu \cdot \frac{\partial \cdot (u \cdot h)}{\partial x}, \tag{4}$$

where x, z are the circumferential and axial coordinate axes in the bearing; h – current gap on the inter-chamber partitions; u – the circumferential velocity of the shaft; P – current pressures on the inter-chamber partitions; μ is the dynamic viscosity of the working fluid.

The current clearance in the bearing h is determined by the dependence:

$$h = \delta_0 - e \cdot \cos(\varphi - \beta_0), \tag{5}$$

where $\delta_0 = R_1 - R_2$ is the radial clearance in the bearing; R_1 – bearing radius; R_2 – shaft radius; e – eccentricity characterizing the distance between the centers of the shaft and bearing; φ – current circumferential angular coordinate; β_0 is the angle of position of the shaft in the bearing.

Approximate numerical methods were used to solve the Reynolds equation. One such method is the finite difference method, which was used in conjunction with the longitudinal-transverse sweep method.

By setting the initial values of the pressures in the grid nodes, the longitudinal-transverse sweep was used to determine the pressure values in the grid nodes, in the next step. The iterative process continued until the specified accuracy was obtained.

The bearing capacity of the bearing was determined from the known pressure values in the chambers and inter-chamber partition assemblies.

Carrying capacity of a bearing was defined as the sum of the bearing capacity of the chambers, inter-chamber and end partitions of the bearing, in projections onto the line of action of the external force and the direction perpendicular to it.

Let's write the expressions for the vertical W_v and W_h horizontal components of the carrying capacity of the bearing as a whole:

$$W_V = W_{Vkam} + W_{Vtk} + W_{Vpkp} + W_{Vp}, \tag{6}$$

$$W_H = W_{Hkam} + W_{Htk} + W_{Hpkp} + W_{Hp},$$

where W_{Vkam} and W_{Hkam} are the bearing capacities of the chambers in vertical and horizontal axis projections; W_{Vtk} and W_{Htk} – bearing capacities of inter-chamber partitions in projections onto the vertical and horizontal axes; W_{Vpkp} and W_{Hpkp} – bearing capacities of the ends of inter-chamber partitions in projections onto the vertical and horizontal axe=s; W_{Vp} and W_{Hp} are the bearing capacities of the ends of chamber partitions in projections onto the vertical and horizontal axes.

According to the calculated carrying capacity of the bearing and its geometric size, the specific pressure of the liquid on the working surface of the bearing was determined:

$$P_p = \frac{W}{L_p \cdot D_p}, \tag{7}$$

where L_p is the length of the bearing; D_p – bearing diameter.

The calculation of the wooden hydrostatic bearing was carried out at the following values of geometric and operating parameters:

1. Bearing diameter D_p=0.33 m.
2. Shaft speed ω=314 s.

3. Bearing power pressure P_1=5; 20; 50 MPa.
4. Bearing length L_p=0.14 m.
5. Working fluid water at a temperature of 25 °C.
6. The radial clearance in the bearing δ_0=0.00009 m.
7. Chamber length l_k=0.1 m.
8. Chamber width b_k=0.02 m.
9. Number of chambers K=4.
10. Shaft eccentricity in bearing e=0.00009 m.

The thermal calculation of a wooden hydrostatic bearing makes it possible to quantify the increase in the temperature of the working fluid and assess its effect on the change in the carrying capacity of the bearing. The determination of the increase in the temperature of the working fluid was calculated from the following dependence [16]:

$$\Delta T = \frac{N_C}{Q \cdot C \cdot \rho}, \tag{8}$$

where Q is the flow rate of the working fluid; ρ is the density of the working fluid; C – heat capacity of the working fluid; N_C – total power loss for friction and lubrication pumping.

The flow rate of the working fluid was determined by the known pressures in the carrying bearing chambers:

$$Q = \psi_1 \cdot \pi \cdot r_1^2 \sqrt{\frac{2 \cdot P_1}{\rho}} \cdot \sum_{i=1}^{m} \sqrt{1 - P_{k,i}}, \tag{9}$$

where m is the number of chambers on the working surface of the bearing; ψ_1=0.62...0.82 – entry coefficient; r_1 is the radius of the nozzle; $P_{k,i}$ is the pressure of the working fluid in the i-th chamber; P_1 is the pressure of the working fluid into the bearing.

The power loss for pumping N_{pr} was determined by the flow rate of the working fluid Q and the pressure of the working fluid supply to the chambers P_1:

$$N_{pr} = Q \cdot P_1. \tag{10}$$

Friction power losses are estimated by the following dependence:

$$N_{tp} = \frac{\mu \cdot \omega^2 \cdot R_b^3 \cdot L_b \cdot 2 \cdot \pi}{\delta_0}, \tag{11}$$

where μ is the dynamic viscosity of the lubricant; ω is the angular velocity of the shaft; R_b – bearing radius; L_b – bearing length; $\delta_0=R_c-R_b$ – radial clearance in the bearing; R_b is the radius of the shaft.

Total power losses for friction and pumping:

$$N_C = N_{tp} + N_{pr}. \tag{12}$$

The author declares that he has no conflict of interest regarding this study, including financial, personal, authorship or other nature, which could affect the research and its results reported in this paper.

5. Investigation of pressures on the working surface of a hydrostatic bearing and analysis of its thermal state

5. 1. Research results and analysis of specific pressures of the working fluid on the working surface of a wooden hydrostatic bearing

The main characteristics of hydrostatic bearings are load capacity, lubricant consumption, and power loss for friction and pumping. The basis for determining these characteristics is the function of pressure distribution in the lubricant layer, which is determined from the joint solution of the Reynolds equations (4) and the flow rate balance (1). For the numerical realization of the flow rate balance, its dimensionless recording (3) was used, which makes it possible to determine the pressures in the bearing chambers by an iterative method. Previously, the gaps in the locations of the chambers were calculated by expression (5). Using the specified geometric dimensions of the bearing and the calculated clearances, the auxiliary coefficients a_i used in the flow balance equation (3) were determined. The process of determining the pressures in the chambers is iterative. Setting the initial values of the pressures in the chambers by expression (3) determined the following values of the pressures in the chambers, which were used for the next iteration as initial. The initial values of the pressures in the chambers were determined from the design experience of hydrostatic bearings. They should be less than the power pressures. This is due to the fact that the working fluid passes through the inlet pressure compensator of the liquid nozzle installed at the inlet to the load-bearing chambers. In the lower carrying chambers, large initial pressure values were set, and in the upper ones, smaller values were set. The more precisely the initial pressures were set, the fewer iterations produced the desired result. The right lower chamber was taken as the first chamber (Fig. 1), and the rest were numbered in a circumferential counterclockwise direction. At a supply pressure of 5 MPa, the initial dimensionless values of the pressures in the chambers were taken as follows: P_{K1}=0.9; P_{K2}=0.24; P_{K3}=0.23; P_{K4}=0.8. The iterative process continued until the specified accuracy was obtained $|P_{K,i+1}-P_{K,i}|\leq 0.0005$. To obtain a given accuracy, it was necessary to perform 7 iterations. After 7 iterations, the pressures in the bearing's chambers were obtained with the specified accuracy: P_{K1}=0.91792; P_{K2}=0.20384; P_{K3}=0.1883467; P_{K4}=0.84179.

The resulting chamber pressures and the discharge pressure drop at the bearing ends, taken to be zero, were used as boundary values to determine the pressures on the inter-chamber partitions. The pressures on the inter-chamber partitions were determined from the solution to the Reynolds equation (4). This equation does not have an exact analytical solution and was solved numerically using the finite difference method. To record the Reynolds equation in finite difference form, the surface between the chambers was covered with a regular grid with 5 mm steps in the circumferential and axial directions. Partial derivatives in equation (4) were written in finite difference form using a five-point pattern.

By setting the initial pressure values in the grid nodes, using the method of longitudinal-transverse sweep along the rows and along the columns, the pressures in the grid nodes were determined in the next step. The initial values of the pressures in the grid nodes were set the same and equal to half the average value of the pressures in the adjacent chambers. The resulting pressure values in the grid nodes in the first iteration were used as the initial ones for the next iteration. The iteration process continued until a predetermined accuracy of 0.0005 was obtained in all corners of the grid.

After determining the pressures in the chambers and on the inter-chamber partitions, the carrying capacity of the bearing was calculated using expression (6). The carrying capacity of a bearing was defined as the sum of the carrying capacity of the chambers, inter-chamber and end partitions of the bearing in projections onto the line of action of the external force and its direction perpendicular. On the surface of the chambers, the pressure was taken to be constant and, after multiplying them by the area of the chambers, the carrying capacity of the chambers was determined. When determining the carrying capacity of inter-chamber partitions, the trapezoid method was used. Knowing the carrying capacity of the bearing, the specific pressure of the lubricant on the working surface of the bearing was determined by expression (7).

Setting different values of power pressures and repeating the described calculation algorithm, specific pressures of the working fluid on the working surface of the bearing were obtained at other values. The results of the calculations are shown in Fig. 2.

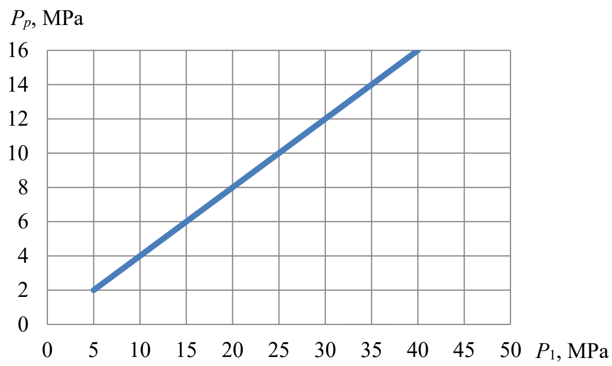

Fig. 2. Dependence of the specific pressure of the working fluid on the working surface of the bearing on the supply pressure P_1

The nature of the change in specific pressure on the working surface of the bearing from the liquid side is linear. With an increase in the supply pressure of the working fluid from 5 MPa to 50 MPa, the specific pressure will increase from 2.3 MPa to 16.65 MPa, that is, approximately 7.2 times.

Quantification of the specific pressure on the working surface of the bearing on the fluid side makes it possible to establish the permissible limits for choosing the geometric and operating parameters of the bearing and is a novelty.

5. 2. Determination of the effect of temperature increase on the carrying capacity of a wooden hydrostatic bearing

According to the accepted geometric and operating parameters, I evaluate the thermal regime of a wooden hydrostatic bearing.

To determine the increase in the temperature of the working fluid, expression (8) was used. The flow rate of the working fluid required for the calculation was calculated by expression (9). When calculating the flow rate of the working fluid, the pressure values in the chambers $P_{K,i}$, obtained according to the calculation algorithm already described, were used. The loss of friction and pumping power losses were calculated using formulas (10) and (11). Setting different values of the supply pressures and using the described calculation algorithm, the increase in the temperature of the working fluid in the bearing was determined. The results of the calculation are shown in Fig. 3.

At a supply pressure of the working fluid of 5 MPa, the increase in the temperature of the working fluid as a result of friction and pumping is 3 °C. With an increase in the supply pressure, the heating of the working fluid increases and at a supply pressure of 50 MPa, the temperature increase of the working fluid is 12.46 °C.

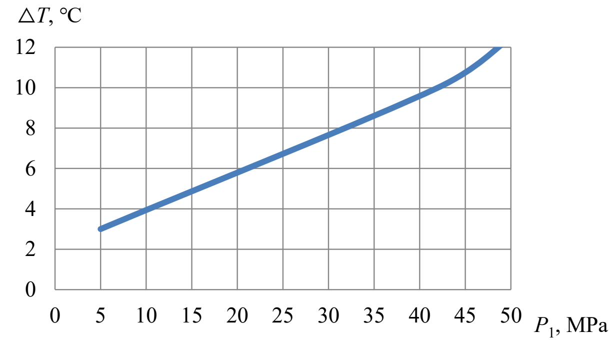

Fig. 3. Determination of the effect of power pressure on increasing the temperature of the working fluid in a wooden hydrostatic bearing

In order to assess the effect of increasing the temperature of the working fluid on the characteristics of the bearing, the carrying capacity of the bearing was calculated taking into account the increase in the temperature of the working fluid.

To identify the effect of increasing the temperature of the working fluid on the carrying capacity of the bearing, according to the described calculation algorithm, the equation of the flow balance (3) and Reynolds (4) was jointly solved. The resulting pressure diagram in the working fluid layer was numerically integrated over the working surface of the bearing and thus its carrying capacity was determined. To quantify the effect of increasing the temperature of the working fluid, two design options were considered. In the first embodiment, the carrying capacity was determined for different values of the supply pressure at a constant temperature of the working fluid. In the second embodiment, the carrying capacity for different values of power pressures was determined at a variable temperature of the working fluid using the results of the calculation shown in Fig. 3. The temperature variability of the working fluid was taken into account using different values of the viscosity of the working fluid. The results of the bearing's carrying capacity calculation are shown in Fig. 4: for the first option, curve 1; and for the second option, curve 2.

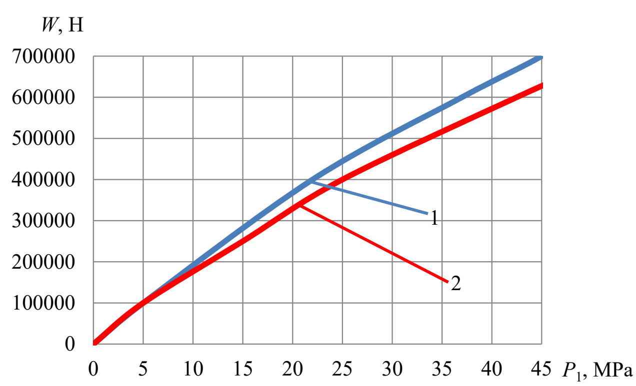

Fig. 4. Determination of the effect of temperature increase on the carrying capacity of a wooden hydrostatic bearing: 1 – at the temperature of the working fluid t=25 °C; 2 – taking into account the heating of the working fluid as a result of friction and pumping losses

With a bearing power pressure of 20 MPa, the bearing load capacity reduction was 4.19 %. With increasing power pressure, the reduction in carrying capacity due to the heating of the working fluid increases. At a supply pressure of 50 MPa, the carrying capacity reduction due to the heating of the working fluid was 10.95 %.

Analysis of the thermal mode of operation of a wooden hydrostatic bearing showed that the increase in the temperature of the working fluid in such bearings is insignificant.

6. Discussion of results of the study of specific pressures on the working surface of the bearing and its thermal mode of operation

The peculiarity of the proposed method is associated with the proposed type of wooden bearing, which provides guaranteed liquid friction. In a wooden hydrostatic bearing, both hydrostatic and hydrodynamic effects are used and there is a need to jointly solve the Reynolds equations and the flow rate balance. The flow rate balance equation takes into account both the gradient and portable flow of the working fluid. In this formulation, the calculation method in works [1–12] was not considered. The results obtained make it possible to estimate the specific pressures of the working fluid on the working surface of the bearing and to establish their effect on the strength of the bearing material. Analysis of the thermal mode of operation of the bearing makes it possible to assess the change in the bearing carrying capacity by increasing the temperature of the working fluid.

The obtained results of the calculation of specific pressures on the working surface of the bearing and its thermal mode of operation are given in Fig. 2–4.

Fig. 2 shows that as the bearing supply pressure increases, the specific pressures on its working surface increase. When the bearing supply pressure increases from 5 MPa to 50 MPa, the specific pressure on the working surface of the bearing increases by about 7.5 times. This is due to the fact that with an increase in power pressure, the pressures in the chambers and on the inter-chamber partitions increase. At the same time, the pressure in the lower chambers increases more significantly than in the upper chambers. The obtained results make it possible to establish permissible limits for the purpose of the structural and operational parameters of the bearing ensuring its operability.

Fig. 3 shows that as the bearing supply pressure increases, the heating of the working fluid due to friction losses and pumping increases. When the bearing supply pressure increases from 5 MPa to 50 MPa, the additional heating of the working fluid changes from 3 °C to 12.46 °C. This is due to the fact that friction between the layers of lubricant increases and power losses for pumping increase. Moreover, with an increase in power pressure, pumping losses increase significantly. If at a pressure of 5 MPa pumping losses amounted to 39 % of the total power loss, then at a pressure of 20 MPa they accounted for 85 % of the total power loss.

Fig. 4 quantifies the effect of increasing the temperature of the working fluid on the carrying capacity of a hydrostatic wooden bearing. Fig. 4 shows that at a bearing supply pressure of 20 MPa, the bearing load capacity reduction is 4.19 %. At a supply pressure of 50 MPa, the carrying capacity reduction due to the heating of the working fluid was 10.95 %. This is due to the fact that with an increase in power pressure, power losses increase, the temperature of the working fluid increases, and its viscosity decreases. Reducing the viscosity of the lubricant leads to a decrease in the carrying capacity of the bearing.

Analysis of the temperature regime of the hydrostatic bearing showed that the increase in the temperature of the working fluid in such bearings is significantly less in comparison with other types of plain bearings.

This is due to the fact that in hydrostatic bearings, the lubricant is supplied under high pressure. A large consumption of lubricant does not allow a significant increase in its temperature.

It is established that the value of the bearing supply pressure affects the specific pressures of the working fluid on the bearing surface.

Analysis of the specific pressures of the liquid on the working surface of the bearing showed that they do not exceed the tensile strength of the bearing material. In the calculations, the maximum value of the specific pressure on the bearing surface of 16 MPa was obtained while the strength limit of lignum vitae is 86 MPa.

Determination of specific pressures on the working surface of the bearing and analysis of its thermal mode of operation makes it possible to design hydrostatic bearings more rationally.

The proposed solutions make it possible to solve the problem of the influence of specific pressures of the working fluid on the strength of the working surface of the bearing in a calculated way that was absent in existing studies. Determination of the change in the temperature of the working fluid in the hydrostatic bearing showed its slight increase.

The advantage of this study is an integrated approach related to solving a complex hydrodynamic problem, addressing the strength issues of the working surface of the bearing and the analysis of the thermal mode of operation of the bearing.

The obtained results make it possible to recommend wooden hydrostatic bearings in the shaft supports of steamship engines, turbines of hydroelectric power plants, submarine shafts, and contribute to improving the reliability and durability of various machines. The derived theoretical dependences make it possible to design wooden hydrostatic bearings taking into account the pressures of the working fluid and the thermal mode of operation, as well as to use them in practical calculations.

The limitations inherent in this study are primarily due to the change in the geometric dimensions of the bearing due to changes in the temperature of the working fluid.

The disadvantage of this study is the lack of quantitative experimental confirmation of the results obtained.

The development of this study may involve improving the mathematical model that takes into account the turbulent flow of the working fluid. Complex experimental studies are required to confirm the results of the calculations.

7. Conclusions

1. It has been established that the magnitude of the bearing supply pressure significantly affects the specific pressures of the working fluid on the bearing surface. When the supply pressure changes from 5 MPa to 50 MPa, the specific pressure of the working fluid on the working surface of the bearing increases from 2.3 MPa to 16.65 MPa, that is, approximately 7.2 times. In the considered range of bearing power pressures, the specific pressures of the liquid do not exceed the strength limits of the bearing material.

2. It has been found that with an increase in the bearing supply pressure, the heating of the working fluid due to friction and pumping losses increases. When the bearing supply pressure is increased from 5 MPa to 50 MPa, the additional

heating of the working fluid increases from 3 °C to 12.46 °C. The reduction in carrying capacity due to an increase in the temperature of the working fluid is not significant. At a bearing power pressure of 20 MPa, the reduction in carrying capacity is 4.19 %, and at a power pressure of 50 MPa, the reduction in carrying capacity is 10.95 %.

Conflicts of interest

The authors declare that they have no conflicts of interest in relation to the current study, including financial, personal, authorship, or any other, that could affect the study and the results reported in this paper.

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Data availability

All data are available in the main text of the manuscript.

References

1. Chandra, P. N., Reddy, P. P. S., Vinayak, K., Shukla, S. R., Rao, A. K. P. (2022). Wear of Wooden Journal Bearings. Biotribology, 29, 100205. doi: https://doi.org/10.1016/j.biotri.2021.100205
2. Wetzk, V. (2006). Entwicklung der Brückenlager – Geschichtlicher berblick. Bautechnik, 83 (10), 721–732. doi: https://doi.org/10.1002/bate.200610063
3. Masaki, N., Takeuchi, S., Hirata, H. (2003). Dynamic Characteristics of a Sliding Bearing Using Lubricant Material for Base Isolation of Light Structures. Seismic Engineering. doi: https://doi.org/10.1115/pvp2003-2101
4. Shamaev, V. A., Parinov, D. A., Polilov, A. N. (2018). Study of the Plain Bearings of Modified Wood for Heavy-Loaded Friction Assemblies. Journal of Machinery Manufacture and Reliability, 47 (2), 168–172. doi: https://doi.org/10.3103/s1052618818020115
5. Anikeyeva, M. V., Vrublevskaya, V. I. (2020). Antifrictional Characteristics of Sliding Bearings from Modified Pressed Wood. Journal of Friction and Wear, 41 (5), 453–458. doi: https://doi.org/10.3103/s1068366620050025
6. Tsubata, S., Sakai, S., Tanaka, S., Morita, T., Nishiya, T., Yamanobe, K., Kanayama, K. (2022). Influence of Thermal Softening of Wood on the Fundamental Performance of Drift Pin Joints I. Mokuzai Gakkaishi, 68 (1), 8–16. doi: https://doi.org/10.2488/jwrs.68.8
7. Stoyanov, V. V., Zhgalli, S. (2016). Load Bearing Capacity of Wooden Bending Elements. Bulletin of Higher Educational Institutions. Lesnoi Zhurnal (Forestry Journal), 1, 115–121. doi: https://doi.org/10.17238/issn0536-1036.2016.1.115
8. Sathre, R., Gorman, T. (2005). Improving the performance of wooden journal bearings. Forest Products Journal, 55 (11), 41–47. Available at: https://rogersathre.com/Sathre&Gorman_2005_wooden_bearings.pdf
9. Dong, C., Yang, Y., Yuan, C., Bai, X., Guo, Z. (2022). Effects of anisotropy of lignum vitae wood on its tribological performances. Composites Part B: Engineering, 228, 109426. doi: https://doi.org/10.1016/j.compositesb.2021.109426
10. Kim, S. S., Yu, H. N., Hwang, I. U., Lee, D. G. (2008). Characteristics of wood–polymer composite for journal bearing materials. Composite Structures, 86 (1-3), 279–284. doi: https://doi.org/10.1016/j.compstruct.2008.03.011
11. Chi, K., Kang, J., Zhang, X., Xiao, S., Die, X. (2019). Experimental application of stochastic resonance based on Wood-Saxon potential on fault diagnosis of bearing and planetary gearbox. Journal of the Brazilian Society of Mechanical Sciences and Engineering, 41 (11). doi: https://doi.org/10.1007/s40430-019-1999-x
12. Wu, Z., Guo, Z., Yuan, C. (2022). Insight into the influence of the anatomical properties of wood on the tribological properties. Journal of Cleaner Production, 330, 129800. doi: https://doi.org/10.1016/j.jclepro.2021.129800
13. Emtsev, B. T. (1978). Tekhnicheskaya gidromekhanika. Moscow: Mashinostroenie, 463.
14. Uskov, M. S., Maksimov, V. A. (1985). Gidrodinamicheskaya teoriya smazki. Moscow: Nauka, 147.
15. Tipey, N., Konstantinesku, V. N., Nika, A., Bitse, O. (1964). Podshipniki skol'zheniya (raschet, proektirovanie, smazka). Buharest, 457.
16. Bogdanov, O. N., D'yachenko, S. K. (1966). Raschet opor skol'zheniya. Kyiv: Tekhnika, 242.