The object of research is hydrostatic processes in sliding bearings with several layers of lubricant.

The task addressed was the influence of the structural and operational parameters of dual hydrostatic plain bearings on their static and dynamic characteristics. As a static characteristic, the bearing capacities of dual and conventional sleeve hydrostatic bearings were considered. When analyzing the dynamic characteristics, the amplitude-frequency characteristics were determined. They were obtained by calculation and as a result of experimental studies. When calculating the amplitude-frequency characteristics, the trajectory method was used. As external forces in the equations of motion of the rotor, hydrodynamic forces, the weight of the rotor, and its unbalance were considered.

Experimental determination of the trajectories of the rotor was carried out on a special bench.

It has been established that the bearing capacity of a double type bearing is approximately 1.75–1.85 times higher than the bearing capacity of a conventional sleeve bearing. The range of stable operation of the rotor on double-type bearings is approximately 1.4 times greater in comparison with the range of stable operation of the rotor on sleeve hydrostatic bearings. The vibration amplitudes in the resonance region for double bearings were approximately 1.5 times less than the rotor vibration amplitudes for sleeve bearings.

The results make it possible to recommend dual-type hydrostatic bearings for rotor bearings of nuclear power plants, in powerful aircraft engines with a gearbox, various types of pumps, and other power plant units. The derived theoretical dependences make it possible to carry out practical calculation of hydrostatic bearings of a double type

Keywords: double bearing, bearing capacity, dynamic characteristics, non-stationary loading, motion trajectory

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REVEALING THE EFFECT OF CHANGING THE OPERATING PARAMETERS OF A DOUBLE HYDROSTATIC BEARING ON ITS CHARACTERISTICS

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1. Introduction

When designing aircraft engines, powerful turbogenerators for nuclear power plants, various types of pumps and other units of power plants, it becomes necessary to solve a number of complex problems.

One of the important tasks is to ensure reliable operation of the rotor supports. The desire to get the dimensions of machines as small as possible is associated with an increase in the rotational speed of their rotors. The presence of residual unbalance of the rotors leads to an increase in the amplitudes of oscillations, and under these conditions, without a dynamic assessment, the machine cannot be recognized as fully operational. An important component in the dynamic evaluation of the machine is the reliable operation of its rotor bearings. The new operating conditions of modern high-speed machines require the improvement of existing or the development of new bearing designs. Given the need to dampen fluctuations in external loads, as well as the high speeds of rotation of the rotors of modern machines, fluid friction plain bearings are becoming more common. One of the directions for the development of gas turbine engines of civil aviation aircraft is to increase the bypass ratio. This gives a significant increase in engine efficiency. To further increase the bypass ratio of the engine, it is necessary to install a reduction gear in it. Such engines are developed by various engine building companies. Due to the large power transmitted by the gearbox, the internal gear bearings are subjected to large loads of the order of 100,000–200,000 N at a rotational speed of about 10,000 min^{-1}. With such high loads and high rotational speeds, rolling bearings cannot provide the required life of more than 10,000 hours. Plain bearings are an alternative to rolling bearings. Pratt Whitney develops geared motors. Plain bearings are used as internal gear bearings.

Very promising structures of plain bearings for modern operating conditions of machines and units of power plants are the designs of dual hydrostatic bearings. Compared to conventional sleeve bearings, they have an increased load capacity, by 1.75–1.85 times, and an extended range of stable operation by about 1.4 times. The oscillation amplitudes in the resonance region for double bearings are approximately 1.5 times less than the oscillation amplitudes of the rotor on sleeve bearings.

Due to the novelty and insufficient knowledge, as well as the prospects for the use of dual-type hydrostatic bearings, studies aimed at analyzing their performance are relevant.

2. Literature review and problem statement

Work [1] considers the static and dynamic characteristics of impact gas-film bearings. Using the theory of elasticity, a model of deformation of a foil with a ledge is built. On the basis of the developed procedure, the pressure distributions of the bearing gas film and the influence of the film structure parameters on the static and dynamic characteristics of the bearing were studied. However, in this study, no attention was paid to ensuring the fluid friction mode, which excludes wear of the bearing working surfaces. When studying the dynamic characteristics, the deformability of the bearing working surface was not taken into account. In [2], journal bearings operating on magnetic fluids are considered. Magnetized bearings can provide liquid support through magnetostatic force. The support provided by LPG helps reduce friction. Such a design will be of great importance especially in precision sliding machines. However, this study does not take into account the influence of dynamic phenomena in the bearing on the sliding accuracy. In [3], transient interactions between sliding wear and liquid-solid-temperature characteristics of a plain bearing were revealed. Support bearing characteristics including wear rate, wear depth, fluid pressure, and maximum temperature were calculated numerically. Numerical results have shown that the worn area forms a surface profile that can be useful for improving the hydrodynamic effect. However, in this study, the hydrodynamic effect will change over time, which will adversely affect the operation of the bearing and create a complex dynamic system. In [4], adjustable plain bearings are considered. The most common adjustment techniques are discharge pressure control; plain bearing clearance control, and lubricant viscosity control. The main tasks to be solved for the industrial application of adjustable plain bearings are considered. However, the study did not pay attention to the automatic control of the bearing. The creation of mechatronic systems is a promising direction in the development of adjustable plain bearings.

In [5], the characteristics and dynamic stability of a three-lobed plain bearing with micro protrusions are considered. Some useful recommendations for future scientific work in this area of research are also given in the work. However, the work does not consider the equations of motion of the rotor and provides an approximate estimate of the dynamic stability of the plain bearing. Work [6] reports experimental studies of sliding support systems. Two procedures of laboratory testing are presented and it is emphasized that they can be used to visualize certain bearing performance parameters. However, the work does not pay sufficient attention to the experimental study of the dynamic characteristics of sliding support systems. Paper [7] briefly describes the version of the original software for analyzing the hydrodynamics of cylindrical plain bearings. A two-dimensional problem of lubricating fluid flow in a bearing gap is considered. The design of the bearing, in which several zones of hydrodynamic friction are formed, is considered. However, the paper does not consider the hydrostatic modes of operation of the bearing, which make it possible to ensure fluid friction in the bearing. In [8], the influence of the structure of the base oil on the elastic-hydrodynamic friction is considered. It has been shown that the pure ester fluids used, with a rather small difference in molecular structure, can have a significant effect on bearing friction. The work does not pay attention to the fact that ethers are simple and complex, as well as to the analysis of other types of working fluid in the bearing. Paper [9] describes a recently developed base isolation system that can significantly reduce the lateral forces transmitted to buildings, bridges, and other structures. The system works on the basis of hydrostatic plain bearings, which minimize the friction between the bearings and the base plates. However, the paper does not take into account that when the bearing is operating in the lubricant layer, in addition to hydrostatic effects, hydrodynamic effects arise, and the operation of the bearing is described by a complex hydromechanical theory. In [10], to improve the reliability of operation, hybrid bearings are proposed, which are a combination of rolling bearings and film bearings. The results of theoretical and numerical studies are presented. The conditions for the occurrence of the minimum friction effect are substantiated. However, the work does not pay attention to the use of balls in rolling bearings made of metal-ceramic materials, which more significantly increase the reliability of the support assembly. In [11], an experimental study of a support bearing with inclined pads is reported. Optimized vortex grooves are made on the working surfaces of the bearing. The results of the study showed that the decrease in the surface temperature of the bearing due to the use of vortex grooves led to a significant increase in the bearing capacity. However, the work does not pay attention to the influence of different types of liquid on the temperature of the bearing surface. Studies are limited and it is impossible to assert a significant increase in the bearing capacity. In [12], plain bearings operating under the hydrodynamic friction mode are considered. On ships and submarines, propellers are often mounted on water-lubricated bearings. The results of an experimental study of the effect of rotation speed on the working eccentricity, friction coefficient and lubrication regime are presented. However, the paper does not consider the turbulent mode of operation of the bearing, which will inevitably occur due to the use of a low-viscosity water working fluid in the bearing. The turbulent regime affects the amount of friction in the bearing. It is noted in [13] that the total cost of friction and wear is 250 billion euros per year worldwide. The results of tests of plain bearings operating on a special mesogenic liquid are presented. This fluid reduces the coefficient of friction by 60 %. However, the work does not pay attention to the stability of mesogenic liquids. They reduce the coefficient of friction but do not make it minimal as in hydrostatic bearings. Work [14] shows the effectiveness of micro texturing on the tribological characteristics of a bearing. That made it possible to reduce the coefficient of friction and increase the contact load under conditions of insufficient lubrication. However, the paper does not pay attention to the consideration of other modes of friction in the bearing, which more significantly reduce the friction coefficient. Insufficient lubrication also leads to reduced bearing reliability. In [15], studies of a plain bearing for a hydraulic pump operating on sea water are reported. The bearing operates under conditions of semi-dry or boundary friction, which reduces its durability. The paper does not consider the possibility of using hydrostatic bearings, which use the working fluids of the pump under high pressure for their work. In [16], a study is performed into the influence of the main parameters on the friction resistance of plain bearings. It is shown that the main influence on friction is exerted by the speed of rotation of the pin and the pressure at the inlet and outlet of the bearing. However, other lubricants are not considered in the work. In connection with the trend of increasing the speed of rotation of the rotors of modern high-speed machines, low-viscosity lubricants are becoming more common as the working bodies of bearings.

Works [17, 18] are closest to the problem considered in the current paper. In [17], the theory of a dual radial hydrostatic bearing under non-stationary loading is presented. In [18], a description of the bench, a pilot plant, and a methodology for experimental studies of dual-type hydrostatic bearings is given. In works [1–18], there is no information on a comparative analysis of theoretical and experimental studies of dual-type hydrostatic bearings. A comparative analysis of experimental studies of the static and dynamic characteristics of double bearings and conventional hydrostatic sleeve bearings is also not considered.

3. The aim and objectives of the study

The purpose of this work is to identify the effect of changing the operating parameters of a dual hydrostatic bearing on its static and dynamic characteristics. This makes it possible to set the allowable limits for the structural and operational parameters of the bearing that ensure its performance.

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

– to determine the magnitude of the increase in the bearing capacity of a dual-type hydrostatic bearing in comparison with a conventional hydrostatic sleeve bearing;

– to determine the magnitude of the increase in the range of stable operation of a double-type hydrostatic bearing in comparison with a conventional hydrostatic sleeve bearing, and also to compare the oscillation amplitudes in the resonance region.

4. The study materials and methods

The object of research is hydrostatic processes in sliding bearings with several layers of lubricant. It is planned to build a mathematical model and devise a procedure for conducting the experiment, making it possible to study the static and dynamic characteristics of dual hydrostatic bearings and conventional hydrostatic sleeve bearings. It was assumed that the pressure gradient across the thickness of the working fluid is small compared to pressure gradients in other directions and the inertial terms of the Navier-Stokes equations are small compared to the viscous ones. The external load acting on the bearing was assumed to be non-stationary.

When deriving theoretical dependences for calculating the characteristics of a dual-type hydrostatic bearing, methods of hydromechanics were used. The characteristics of the bearing were determined on the basis of the pressure distribution function in the working fluid layer. The method of experimental research involved the analysis of the shaft rising on the layer of the working fluid, the registration of the amplitudes of the rotor oscillations and the boundaries of its stable movement. The Reynolds and flow balance equations were generalized to the case of a turbulent flow of the working fluid. The dynamic characteristics of the bearing were determined using the equations of motion of the rotor inside the bearing. When determining the pressures in the chambers, an iterative method was used until the specified accuracy was obtained. The pressure on the interchamber bridges was determined using the Reynolds equation, which was solved numerically by the finite difference method. The equations of motion of the rotor inside the bearing were solved numerically using the multi-step Adams method. The initial information for the Adams method was determined using the Euler method. When constructing the amplitude-frequency characteristics of the movement of the rotor inside the bearing, the trajectory method was used.

The numerical implementation of the derived mathematical dependences was carried out in the Excel software (developed by Microsoft Corp., USA). The drawings of the results were made in the Compass graphic editor.

The dual-type hydrostatic bearing under consideration consists of disk 1 (Fig. 1) fixedly mounted on shaft 2, outer part 3, and two inner parts 4 and 5. The outer rim of disk 1 is made expanded along the outer part, which allows the working fluid to be supplied both to its outer part and to the inner parts. This allows the formation of a double hydrostatic bearing with several layers of lubrication.

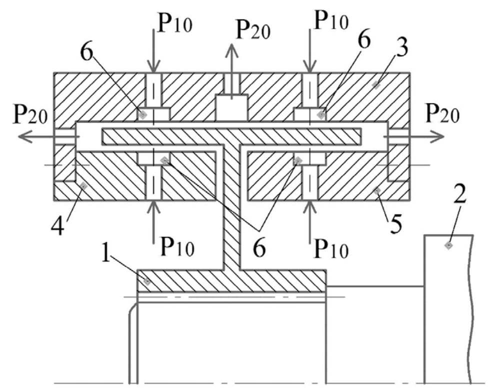

Fig. 1. Scheme of a double-type radial hydrostatic bearing: 1 – disk; 2 – shaft; 3 – the outer part of the bearing; 4, 5 – internal parts of the bearing

The necessary clearance is made between the working surfaces of the disk and the bearing. On the working surfaces of the bearing (on the outer and inner), bearing chambers 6 are made, into which the working fluid is supplied under high pressure P_{10}. At the entrance to the chambers, inlet pressure compensators – jets are installed. Having passed through the slotted path of the bearing, the working fluid enters the drain with a pressure of P_{20} (Fig. 1). Under the influence of the weight of the shaft with the disk, its center is displaced relative to the center of the bearing. The distance between the centers of the shaft and the bearing is called the eccentricity. Due to the concentricity of the position of the shaft with the disk in the bearing in the outer part of the bearing in the lower chambers, the pressure increases since the gap has decreased here, and in the upper chambers the pressure decreases due to the increase in the gap. This results in a load-bearing capacity in the outer part of the bearing. In the inner parts of the bearing, on the contrary, in the upper chambers the pressure increases since the gap has decreased there, and in the lower chambers the pressure decreases due to the increase in the gap. This creates a load-bearing capacity in the internal parts of the bearing. The bearing capacities of the outer and inner parts of the bearing are summed up since they are directed in the same direction. Therefore, the bearing capacity of a double bearing is significantly greater than that of a single sleeve bearing with one lubricating layer.

The theory of a dual radial hydrostatic bearing under a non-stationary external load is described in detail in [17]. The theoretical determination of the amplitude-frequency characteristics is associated with the joint solution of the system of equations of hydromechanics and the equations of motion of the shaft with the disk:

$$Q_{i1}=Q_{i11}+Q_{i12}, \quad (1)$$

$$Q_{i2}=Q_{i21}+Q_{i22}, \quad (2)$$

$$\frac{\partial}{\partial x_1}\left(\frac{h_1^3}{Kx_1}\frac{\partial P_1}{\partial x_1}\right)+\frac{\partial}{\partial z_1}\left(\frac{h_1^3}{Kz_1}\frac{\partial P_1}{\partial z_1}\right)=$$
$$=6\cdot\mu\cdot\frac{\partial(U_1\cdot h_1)}{\partial x_1}+12\cdot\mu\cdot V, \quad (3)$$

$$\frac{\partial}{\partial x_2}\left(\frac{h_2^3}{Kx_2}\frac{\partial P_2}{\partial x_2}\right)+\frac{\partial}{\partial z_2}\left(\frac{h_2^3}{Kz_2}\frac{\partial P_2}{\partial z_2}\right)=$$
$$=6\cdot\mu\cdot\frac{\partial(U_2\cdot h_2)}{\partial x_2}+12\cdot\mu\cdot V, \quad (4)$$

$$\frac{G}{g}\left[\frac{d^2e}{dt^2}-e\cdot\left(\frac{d\beta_0}{dt}\right)^2\right]=-2\cdot I+G\cdot\cos\beta_0+$$
$$+q\cdot\omega^2\cdot\cos(\omega\cdot t-\beta_0-\beta_H), \quad (5)$$

$$\frac{G}{g}\left[e\cdot\frac{d^2\beta_0}{dt^2}+2\cdot\frac{d\beta_0}{dt}\cdot\frac{de}{dt}\right]=-2\cdot J-G\cdot\sin\beta_0+$$
$$+q\cdot\omega^2\cdot\sin(\omega\cdot t-\beta_0-\beta_H), \quad (6)$$

where x_1, z_1 and x_2, z_2 are coordinate axes;

Q_{i1}, Q_{i2} – flow rates of the working fluid through the inlet compensating jets, respectively, for the outer and inner working surfaces of the bearing;

Q_{i11}, Q_{i21} – flow rates of the working fluid along the contour of the chambers for the outer and inner working surfaces of the bearing;

Q_{i12}, Q_{i22} – flow rates of the working fluid due to the movement of the shaft along the line of centers of the bearing and shaft;

P_1, P_2 – current values of pressure in the working fluid layer on the inter-chamber bridges for the outer and inner working surfaces of the bearing;

Kx_1, Kz_1 and Kx_2, Kz_2 – turbulence coefficients along the X and Z axes for the outer and inner working surfaces of the bearing;

h_1, h_2 are the current values of the clearances in the outer and inner parts of the bearing;

μ is the dynamic viscosity of the working fluid;

U_1, U_2 – circumferential speeds of the outer and inner working surfaces of the disk, which is fixedly mounted on the shaft;

V is the speed of approach of the disk to the shaft and the bearing;

G is the weight of the shaft with the disk;

g is the free fall acceleration;

e is the eccentricity of the disk in the bearing, which characterizes the distance between the centers of the disk and the bearing;

β_0 is the position angle of the disk in the bearing, which characterizes the position of the center line of the disk and the bearing;

t is the current time;

I, J – load capacity of the bearing in projections onto the line of centers of the disk and the bearing and the direction perpendicular to it;

q is the residual unbalance of the shaft;

ω is the angular velocity of rotation of the disk with the shaft;

β_H is the position of the angle reference axis.

A bench [18] was used to study the experimental characteristics of a dual-type hydrostatic bearing. It consists of an electric drive of the rotor of the experimental setup, a power supply system for double-type bearings, auxiliary systems, and a complex of measuring equipment. The drive consists of a DC motor and a gear multiplier.

An experimental comparison of the bearing capacity of double and sleeve hydrostatic bearings was carried out with a non-rotating shaft. When the supply pressure of the working fluid was zero, the shaft lay on the bearing. When a working fluid was supplied with different pressures at the inlet to the chambers, the shaft floated up. The value of the ascent was recorded by the indicator. After the experimental unit was sorted out, plugs were installed in the internal parts of the double–type bearing instead of jets and the bearing turned into a conventional sleeve. For this case, a working fluid was also supplied with different pressures at the inlet to the chambers, and the value of the shaft ascent was recorded.

A theoretical study of the dynamic characteristics of bearings was carried out on the basis of the trajectory method. Based on the calculated trajectories of the rotor inside the bearing, the amplitude-frequency characteristics were built, according to which the range of stable operation was established and the oscillation amplitudes in the resonance region were analyzed. The determination of the amplitude-frequency characteristics of the bearing was carried out using the developed calculation algorithm. The determination of the dynamic characteristics of the bearing is based on the pressure distribution function in the lubricant layer, which was determined from the joint solution of the Reynolds equations (3) and (4) and the balance of costs (1) and (2). Numerical implementation of the flow balance equations makes it possible to determine the pressures in the carrier chambers in an iterative way. The pressures in the chambers were determined taking into account the turbulence of the lubricant flow. Given the initial pressure values in the chambers, the following pressure values in the chambers were determined using the flow balance equations. The pressure values obtained in the chambers were used for the next iterations as the initial ones. The initial pressure values in the chambers were assigned from the experience of designing hydrostatic bearings. They must be less than the supply pressures. This is due to the fact that the working fluid passes through the inlet pressure compensator jet installed at the inlet to the carrier chambers. The more precisely the initial values of the pressures in the chambers were set, the smaller number of iterations was needed for the required result. It should be noted that in the numerical implementation of dynamic problems, it is necessary to increase the accuracy of obtaining results than in the implementation of static problems.

The pressure values obtained in the chambers and the pressure drop across the drain at the ends of the bearing, taken equal to zero, were used as boundary values to determine the pressures on the interchamber bridges. The pressures on the interchamber bridges were determined from the solution of the Reynolds equations (3) and (4). These equations do not have an exact analytical solution and were solved numerically using the finite difference method. The Reynolds equations were also generalized to the case of a turbulent flow of a working fluid using the most common method of V. N. Constantinescu. It is based on L. Prandtl's hypothesis about the mixing path.

To write the Reynolds equations in a finite difference form, the surface between the chambers was covered with a regular grid. The partial derivatives in equations (3) and (4) were written in finite difference form using a five-point template. Given the initial values of the pressures at the grid nodes and using the method of longitudinal-transverse sweep along the rows and along the columns, the pressures at the grid nodes were determined at the next step. The initial pressure values at the grid nodes were set to be the same and equal to half the average pressure value in adjacent chambers. The pressure values obtained at the grid nodes at the first iteration were used as initial values for the next iteration. The iteration process continued until the specified accuracy was obtained.

After determining the pressures in the chambers and on the inter-chamber bridges, the bearing capacity was calculated. It was determined as the sum of the bearing capacities of the chambers, inter-chamber and end jumpers of the bearing in projections onto the line of centers of the disk and the bearing I and the direction perpendicular to it J. The pressure on the surface of the chambers was taken constant, and after multiplying them by the area of the chambers, the bearing capacity of the chambers was determined. When determining the bearing capacity of inter-chamber jumpers, the Simpson method was used.

To build the amplitude–frequency characteristics, the method of shaft motion trajectories inside the bearing was used. The trajectories of the rotor movement inside the bearing were calculated using nonlinear equations of the rotor movement inside the bearing (5). The external loads in the equations of motion of the rotor inside the bearing were the hydrodynamic forces I and J, the weight of the rotor G, and its unbalance q. To solve the equations of motion (5), the multi-step Adams method was used since the error introduced into the Adams method at any step does not tend to grow exponentially. To calculate the coordinates and velocities of the disk center, the Adams method of the fourth order of accuracy was used. To implement the multi-step Adams method, it is necessary to have information about the four previous points. This information was determined using the one-step Euler method. The calculation using the described methods continued until repeating trajectories were obtained.

The amplitude-frequency characteristic is the dependence of the amplitude of the oscillations of the rotor on the frequency of its rotation and is built point by point. Given different values of the rotor speed, the trajectories of the rotor inside the bearing were calculated, which gave one point on the amplitude-frequency characteristic. The resulting amplitude-frequency characteristic makes it possible to analyze the magnitudes of the oscillation amplitudes, the resonance zone, and the boundaries of the stability of the rotor rotation.

The experimental determination of the amplitude-frequency characteristics of a shaft on hydrostatic bearings of a double type was carried out using two non-contact inductive sensors D-20 and amplifying equipment ID-2I. Inductive sensors were installed in two mutually perpendicular planes. They measured the vertical and horizontal movement of the shaft. The value of the radial clearance between the sensor and the shaft surface was approximately 0.2–0.4 mm. This value was obtained experimentally and allows obtaining the required sensitivity of the measuring equipment and the linear characteristic of the sensor. The signal from the displacement sensors was transmitted to amplifying equipment and, after amplification, the signal was output to a two-channel oscilloscope. The trajectory of the shaft movement appeared on the screen of a two-channel oscilloscope. The trajectory of the shaft movement inside the bearing was recorded using a video camera. The procedure of experimental studies of the amplitude-frequency characteristics of the shaft included the calibration of displacement sensors.

During the experiments, the measurements were carried out at a steady temperature regime. The readings of the measuring instruments were recorded simultaneously. The pressure of the supply of the working fluid, the working revolutions, and the movement of the shaft were measured. The shaft speed was adjusted by the generator rheostat.

Experimental studies of the operation of the shaft in an unstable region are associated with the danger of an accident of the study unit. Usually, when studying the dynamics of a shaft on hydrostatic bearings, they limit themselves to determining the instability boundary, that is, the revolutions at which self-excited oscillations (self-oscillations) begin to appear.

Experimental studies were carried out at various values of the working fluid supply pressure and in the range of shaft angular velocities from zero to values at which self-oscillations of the shaft appeared.

5. Results of theoretical and experimental studies of the characteristics of dual-type hydrostatic bearings

5. 1. Experimental comparative analysis of the bearing capacity of dual and conventional sleeve hydrostatic bearings

The double type bearing under test had the following geometrical and operating parameters:

1. The outer diameter of double bearing D_p=0.091 m.
2. The inner diameter of the double bearing D_n=0.083 m.
3. The outer diameter of the disk D_1=0.09088 m.
4. The inner diameter of the disc D_2=0.08912 m.
5. External and internal clearances between the disk and the bearing δ=0.00015 m.
6. Number of chambers in one bearing K=16.
7. Bearing length L_p=0.05 m.
8. Chamber length L_k=0.015 m.
9. Diameter of jets d=0.0012 m.
10. The weight of the shaft with disks G=15 kg.
11. Working fluid – water.

The test results for a non-rotating shaft are shown in Fig. 2.

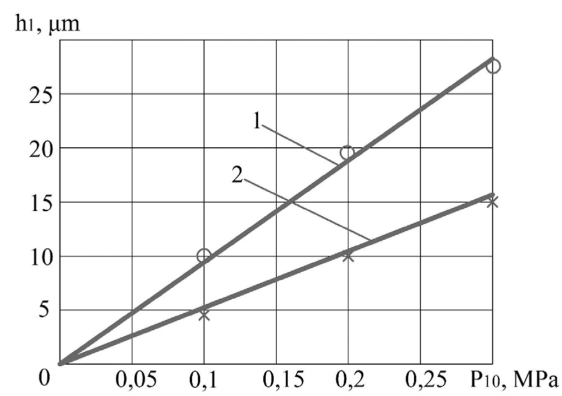

Fig. 2. Dependence of the minimum clearance h_1 between the shaft and the bearing on the supply pressure of the working fluid P_{10}: 1 — double hydrostatic bearing; 2 — single sleeve hydrostatic bearing

Fig. 2 shows that at all considered pressures of the supply of the working fluid P_{10}, the ascent of the shaft in double-type bearings is much larger (approximately by 1.8–1.85 times) in comparison with a single sleeve bearing. This suggests that the double hydrostatic bearing has a greater bearing capacity compared to a single sleeve hydrostatic bearing by about 1.8–1.85 times.

5. 2. Theoretical and experimental comparison of the ranges of stable operation of double and sleeve hydrostatic bearings

The results of an experimental study of the amplitude-frequency characteristics of the shaft on dual hydrostatic bearings are shown in Fig. 3.

An analysis of the results obtained shows that at both values of the pressure of the supply of the working fluid, there is a resonance and loss of stability of the movement of the shaft mounted on dual-type hydrostatic bearings. With an increase in supply pressure from 2 atm to 3 atm, the rotation frequency at which resonance is observed increases by about 1.5 times. The oscillation amplitude "A" in the resonance region also increases by about 1.2 times with an increase in the working fluid supply pressure.

Comparison of the calculated and experimental data at both supply pressures showed that the difference in oscillation amplitudes does not exceed 8 μm, and the stability limit in comparison with the experimental values is somewhat overestimated within 10–15 %. In the stable region, the trajectories of the shaft center had the form of closed curves close to a circle or an ellipse (Fig. 4). At the stability boundary, the motion trajectories were more complex due to the superposition of self-excited oscillations (self-oscillations) on the forced oscillations (Fig. 5).

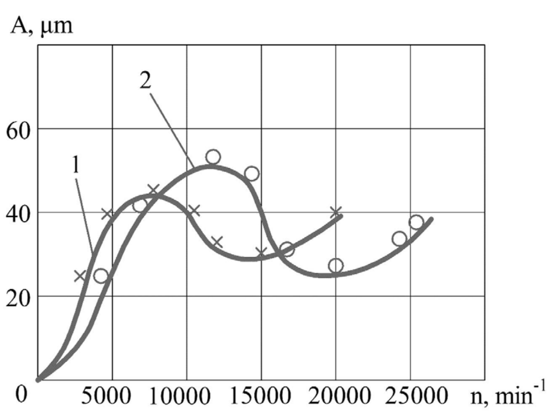

Fig. 3. Calculated and experimental values of the amplitude-frequency characteristics of the shaft on hydrostatic bearings of the double type: 1 – at the pressure of the working fluid supply P_{10}=2 atm; 2 – at the pressure of the supply of the working fluid P_{10}=3 atm

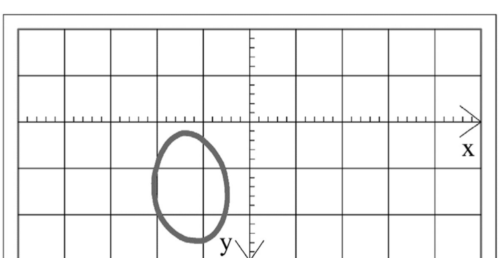

Fig. 4. The trajectory of the rotor center in a stable area

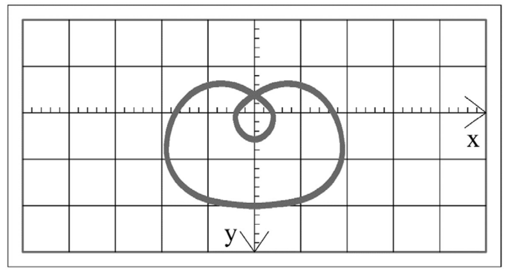

Fig. 5. The trajectory of the shaft center at the stability boundary

The oscillation amplitude in the resonance region of the sleeve bearing (Fig. 6) increases by 1.2 times with an increase in supply pressure from 2 atm to 3 atm. The range of stable operation of the shaft with an increase in the supply pressure of the working fluid from 2 atm to 3 atm increases by about 1.3 times.

A comparison of the calculated and experimental data at both supply pressures showed that the difference in oscillation amplitudes did not exceed approximately 8 μm, and the stability limit was somewhat overestimated in comparison with the experimental one within 10–15 %.

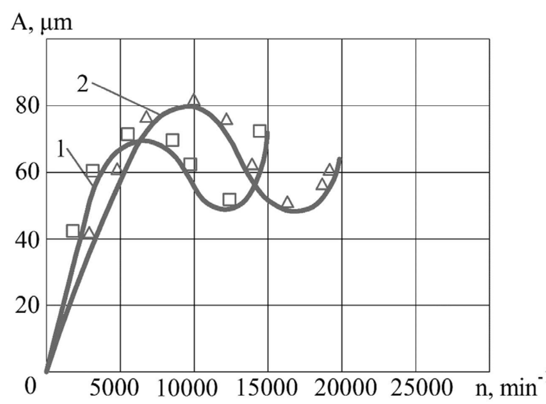

Fig. 6. Calculated and experimental values of the amplitude-frequency characteristics of the shaft on sleeve-type hydrostatic bearings: 1 – at the pressure of the working fluid supply P_{10}=2 atm; 2 – at the pressure of the supply of the working fluid P_{10}=3 atm

The range of stable operation of the shaft on double hydrostatic bearings is approximately 1.4 times greater in comparison with the range of stable operation of the shaft on sleeve hydrostatic bearings.

6. Discussion of results of the study of static and dynamic characteristics of double and sleeve hydrostatic bearings

The peculiarity of the proposed method is associated with the proposed type of double bearing with several layers of lubricant. In a dual hydrostatic bearing, it becomes necessary to jointly solve the Reynolds equations and balance the costs in the outer and two inner parts of the bearing. The Reynolds equation and the flow balance take into account the turbulence of the flow of the working fluid. In this statement, the calculation procedure was not considered in [1–16]. The results make it possible to reveal the magnitude of the increase in the bearing capacity and the expansion of the range of stable operation of a dual hydrostatic bearing in comparison with a conventional hydrostatic sleeve bearing.

The results of the calculation of the bearing capacity and the range of stable operation of double and sleeve bearings are shown in Fig. 2–6.

Fig. 2 shows that at all considered working fluid supply pressures, the shaft floating in double-type bearings is approximately 1.8–1.85 times higher in comparison with a single sleeve bearing. This means that the double bearing has a greater load capacity than a single sleeve bearing, by about 1.8–1.85 times. This is due to the fact that in a double bearing there is an outer and two inner parts, the bearing capacities of which are summed up. The results obtained make it possible to reduce the diametral dimensions of the bearing to ensure a given bearing capacity.

Fig. 3 shows that the amplitude of oscillations in the resonance region with an increase in the supply pressure of the working fluid from 2 atm to 3 atm increases by about 1.2 times, the range of stable operation also increases by about 1.3 times.

Fig. 4 demonstrates that in the stable region, the trajectory of the shaft center had the form of closed curves close to a circle or an ellipse, that is, the shaft performed harmonic oscillations.

Fig. 5 shows that at the stability boundary, the trajectory of the shaft movement had a more complex character due to the superposition of self-excited oscillations (self-oscillations) on the forced oscillations.

Fig. 6 demonstrates that the sleeve hydrostatic bearing has a clearly defined resonant zone and the oscillation amplitude is approximately 1.5 times greater in comparison with the double hydrostatic bearing. The range of stable operation of a sleeve bearing is approximately 1.4 times less than that of a double bearing. The improved performance of a double bearing compared to a single sleeve bearing is due to the presence of several lubricating layers and increased damping capacity.

The proposed solutions allow solving the problem of determining the static and dynamic characteristics of bearings with several lubricating layers.

The advantage of this study is an integrated approach associated with solving a complex hydrodynamic problem and nonlinear equations of rotor motion inside the bearing. The proposed method makes it possible to analyze the resonance zones and the boundaries of the stable operation of the shaft inside the bearing.

The results make it possible to recommend dual hydrostatic bearings in the rotor bearings of nuclear power plants, in high-power aircraft engines with gearboxes, and in various power plant units. The developed algorithm for calculating double hydrostatic bearings makes it possible to perform their practical calculation.

The limitations inherent in this study are related to the capabilities of the experimental bench, which do not allow studies to be carried out in a wider range of supply pressures.

The disadvantage of this study is that it does not take into account thermal phenomena that can take place in the supports of powerful turbogenerators.

The development of this research may consist in the improvement of the mathematical model, which makes it possible to solve jointly the dynamics of the bearing and shaft.

7. Conclusions

1. It has been experimentally established that the bearing capacity of a double hydrostatic bearing is approximately 1.8–1.85 times higher than the bearing capacity of a conventional hydrostatic sleeve bearing.

2. It has been established theoretically and experimentally that the range of stable operation of the rotor on dual hydrostatic bearings is approximately 1.4 times wider than the range of stable operation of the shaft on sleeve bearings. It was found that in the region of resonance, the amplitude of rotor oscillations on double bearings is approximately 1.5 times less in comparison with the amplitudes of rotor oscillations on single sleeve bearings. Comparison of the amplitude-frequency characteristics of the rotor, obtained by calculation and experimentally, shows good agreement.

Conflicts of interest

The author declares that he has 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.

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