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REVEALING THE INFLUENCE OF STRUCTURAL AND OPERATIONAL PARAMETERS OF A HYDROSTATIC BEARING IN A GEAR-TYPE FUEL PUMP ON ITS MAIN CHARACTERISTICS

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The object of this study is hydrostatic processes in the sliding supports of gear-type fuel pumps.

The problem solved was the influence of the structural and operational parameters of hydrostatic bearings of the fuel pump on their static characteristics. The carrying capacity and consumption of the lubricant were considered as static characteristics. The characterization was based on the function of distributing the pressure in the lubricant layer. It was determined from the joint solution of the Reynolds equations and the balance of costs. The carrying capacity of the bearing was determined by the numerical integration of the pressure distribution function in the lubricant layer. The lubricant consumption was determined by the calculated pressures in the chambers. Variants of the working surface of the bearing with two and three carrying chambers were considered. Due to the fact that the load in the pump acts in one direction during operation, the scheme of the working surface of the bearing with two carrying chambers was adopted. The fluid consumption of such a bearing was less compared to a bearing with three carrying chambers. One of the parameters that significantly affect the carrying capacity of the bearing is the diameter of the nozzle installed at the inlet to the chambers.

It has been established that the dependence of the carrying capacity of a hydrostatic bearing on the diameter of the nozzle is nonlinear. With an increase in the diameter of the nozzle from 1 mm to 2.3 mm, the carrying capacity of the bearing increased by about 2.83 times. The extraction of fuel for the operation of the hydrostatic bearing was 1 % of the fuel pumped by the pump.

The results make it possible to recommend hydrostatic bearings as shaft supports for gear-type fuel pumps and can be used for practical calculations

Keywords: hydrostatic bearing, gear pump, carrying capacity, Reynolds equation, flow rate balance

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1. Introduction

The reliability of any machine is largely determined by the operability of the rotor supports of power plant units. One of the important units in the system of operation of an aircraft engine is a heat pump. Rolling bearings are most widely used as shaft supports for gear-type fuel pumps. However, the use of rolling bearings has some drawbacks. Due to the heavy loads acting on the rolling bearings, they have large radial dimensions, often exceeding the dimensions of gears. In addition, an additional lubrication system is required to lubricate the rolling bearings since it is not possible to use the working fluids of the machines as a lubricant.

One of the main advantages of hydrostatic bearings is the possibility of using kerosene as a lubricant of the working fluid of the machine. The kerosene in the pump is under high pressure, which also indicates the possibility of using hydrostatic bearings. Hydrostatic bearings refer to plain bearings that provide guaranteed fluid friction. The main criterion for the performance of these bearings is the minimum thickness of the working fluid layer separating the rubbing surfaces. The thickness of the lubricating film should exceed the total height of micro-irregularities and deviations from the shape of the circles of the working surfaces of the shaft and bearing. Thus, for one full revolution of the shaft, there is no contact of micro-irregularities of rubbing surfaces. Therefore, these bearings are practically wear-free if you do not take into account the initial moments of starting and the end of the shaft rotation stop.

The use of hydrostatic bearings also expands the range of materials used for their manufacture. The diametrical dimensions of hydrostatic bearings are also smaller than those of rolling bearings.

A number of advantages of hydrostatic bearings indicate the need for their use in gear-type aviation fuel pumps. The lack of information on the design of these bearings for fuel pumps makes research on this problem relevant.

2. Literature review and problem statement

Paper [1] considers a hybrid plain bearing, the radial load applied to the gears due to an increase in pressure in the pump is fully perceived by hybrid plain bearings. Low viscosity aviation fuel is used to lubricate them, which makes the design and analysis of plain bearings particularly challenging. A numerical model has been developed for the analysis of hybrid plain bearings for fuel pumps. As a result, a simple and reliable methodology for the characteristics of the pressure distribution in the lubricant was obtained. However, the cited study does not pay attention to assessing the impact of bearing operation on pump flow characteristics. Paper [2] reports a simulation and experimental verification of the movement of a floating bearing sleeve in a gear pump with external gearing. The model of connection graphs adopted in theory has been tested experimentally. However, the cited study does not pay attention to the use of hydrostatic effects that occur in the lubricant layer due to the use of load-carrying chambers on the bearing surface. Work [3] discusses the design strategy and the procedure for calculating a gear-type pump, which includes several stages. One of these steps is the design of bearings for the pump shafts. The design of the support bearings, in particular the radial clearance, is aimed at achieving a satisfactory minimum oil film value under given boundary conditions. At this stage of the actual calculation, an iterative bearing design procedure was implemented. However, the cited study did not pay attention to the use of high-pressure pump fluid for bearing operation. In [4], the reverse design with liquid lubrication of gear pump bearings is considered. Using the minimum thickness of the oil film and the deflection of the neck as the objective function, the model is built using reverse design thinking. If the carrying capacity coefficient is neglected, the probability of bearing tipping becomes less. However, the cited study does not consider the possibility of using carrying chambers in optimizing the bearing design. Paper [5] considers the solution of the system problem of a gear pump with plain bearings based on a new simulation model. The developed model consists of a system of modules: a hydrodynamic model of the pump, a model of bearing lubrication, a model for evaluating the movement of the shaft and a geometric model. The bearing transition lubrication model includes the effect of misalignment and surface roughness. Work [6] considers the mobility method proposed by Booker to predict the movement of the shaft inside the bearing. The cited article presents a hybrid method with the addition of a hydrostatic lubrication effect to the original mobility method. However, in the cited study, the problem of dynamics is considered in a simplified linear statement and does not take into account the whole complex of forces affecting the trajectory of the shaft inside the plain bearing. In [7], the experimental conditions of operation of plain bearings of aviation fuel pumps are considered: high temperature, high pressure, and media with low viscosity. A study of ultrasonic measurement of the dynamic thickness of the oil film of the fuel pump is reported. However, the cited study does not consider changes in bearing dimensions due to high temperature and that affect the thickness of the oil film. The viscosity of the lubricant also changes under the influence of high temperature. Paper [8] presents an approach to the calculation of a one-dimensional hybrid pressure distribution. The Reynolds equation with the corresponding boundary condition is solved to derive a two-dimensional distribution of hydrodynamic pressure using the finite difference method. Based on hydrostatic and hydrodynamic pressure, a one-dimensional total pressure profile can be obtained. However, the cited study does not consider a two-dimensional model of the oil film, and this can lead to significant errors in determining the characteristics of the bearing. Paper [9] proposes a strategy for optimizing the design of bearings, based on CFD modeling and data on lubricant characteristics. Based on the simulation data, the bearing design is optimized using a genetic algorithm. The results showed that the load capacity error was less than 1 %, with an average rise in fuel temperature of 40 °C. However, the cited study did not pay attention to dynamic phenomena that can significantly affect bearing optimization. In [10], the dynamic characteristics of dry screw pumps are simulated, taking into account the forces created by gas, gears, bearings, and rotor gravity, and the load on the bearing is also considered in detail. The throttle force, gear load, bearing load, and bearing power loss at different initial pressures were comprehensively discussed. However, the cited study does not pay attention to the effect of gear shaft speed on power loss in bearings, which can be significant. Paper [3] reports a methodology for designing multistage pumps with external gearing and a shaft supported by hydrodynamic plain bearings. The results underline the accuracy of the methodology for estimating the required feed rate. In addition to accuracy, this procedure is flexible and reliable. However, the cited study did not consider the use of a high-pressure pump fluid for the operation of a plain bearing. In [11], the use of active magnetic bearings in various devices is considered. The finite element method allows for numerical analysis of systems of this type. In the cited article, a two-dimensional software simulation of the AMP axial system was performed. However, the cited study does not consider the possibility of investigating the working fluid of the pump for the operation of bearings. In [12], the parts of the engine and transmission operating under high loads and conditions of dry and boundary lubrication are considered. It is shown that the coating of tungsten carbide carbide, obtained by chemical deposition from the gas phase, prevents abrasion and protects important parts from wear and corrosion. A coating with a thickness of 50 μm makes it possible to get a hardness close to 72 HRC. However, the cited study does not consider the possibility of completely eliminating wear through the use of hydrostatic bearings.

The closest to the problem considered in the current article is work [13]. Paper [13] discusses the theory and results of the calculation of a hydrostatic bearing, in which both hydrodynamic and hydrostatic effects are used. The flow of the working fluid is generalized to the case of turbulent flow of the lubricant.

In works [1–13], the specific design of the working surface of the bearing, which is most suitable for bearings of gear fuel pump pumps, is not considered. A comparative analysis of various design options for the working surface of the bearing in relation to gear-type fuel pumps is also not considered. There is no information on the carrying capacity and flow rate of the working fluid (fuel) for the adopted specific design of hydrostatic bearings.

3. The aim and objectives of the study

The purpose of this work is to identify the influence of the structural and operational parameters of the hydrostatic bearing of the fuel gear pump on its static characteristics. This makes it possible to solve the problem of assigning the design parameters of the bearing.

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

– to identify the effect of the number of carrying chambers on the working surface of a hydrostatic bearing on its carrying capacity and working fluid consumption;

– to establish the diameter of the nozzle that provides the predefined carrying capacity of the bearing and to identify the effect of fuel extraction for the operation of the hydrostatic bearing on the flow characteristics of the pump.

4. The study materials and methods

The object of the study is hydrostatic processes in the sliding supports of gear-type fuel pumps. It is planned to construct mathematical models of hydrostatic bearings with two and three carrying chambers on the working surfaces of the bearing. Conducting a comparative analysis of the carrying capacity and flow rate of the working fluid. It has been assumed that the pressure gradient over the thickness of the working fluid is small in comparison with the pressure gradients in other directions and the inertial terms of the Navier-Stokes equations are small in comparison with the viscous ones. The external load on the bearing was assumed to be constant.

When establishing theoretical dependences, fluid mechanics methods were used to calculate the characteristics of a hydrostatic bearing. The characteristics of the bearing were determined on the basis of the function of pressure distribution in the working fluid layer. The Reynolds equations and the cost balance equations were generalized to the case of turbulent flow of the working fluid. When determining the pressures in the chambers, an iterative method was used. The calculation continued until the specified accuracy was obtained. The pressures on the interchamber bridges were determined using the Reynolds equation, which was solved numerically by the finite difference method.

The numerical implementation of the derived mathematical dependences was carried out in the Excel software (developed by Microsoft, USA). Drawings of the results obtained were carried out in the graphic editor «Compass».

The layout of the supports of the fuel gear pump is shown in Fig. 1.

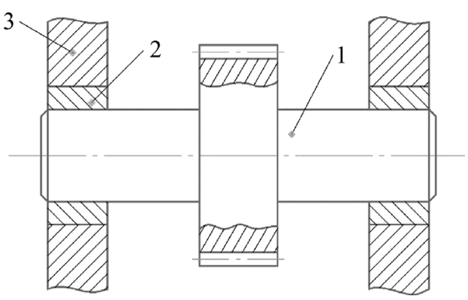

Fig. 1. The layout of the supports of the fuel gear pump: 1 – shaft; 2 – hydrostatic bearing; 3 – housing

When designing the structure of a hydrostatic bearing, two structural schemes were considered, taking into account the features of the gear pump. The traditional design of the sleeve hydrostatic bearing has been changed because in the gear pump the bearing load acts constantly in one direction. Fig. 2 shows diagrams of hydrostatic bearings with two carrying chambers (*a*) and with three carrying chambers (*b*).

In the above diagrams of hydrostatic bearings, carrying chambers are made on the working surfaces, the pressure in which is indicated as P_k. The working fluid is supplied to the carrying chambers under high pressure P_1. At the inlet to the chambers, inlet pressure compensators with a small diameter d_k are installed.

The main static characteristics of a hydrostatic bearing are the load-carrying capacity and flow rate of the working fluid. The determination of these characteristics is based on the function of pressure distribution in the working fluid layer, which was determined from the joint solution of the Reynolds equations and the flow balance of the working fluid. The pressures in the chambers were determined from the solution of the equation of the balance of flow rates of the working fluid. The balance of flow rates of the working fluid is recorded subject to the condition of equality of costs through the inlet compensating devices (nozzles) and flows along the contour of the *i*-th chamber (Fig. 3).

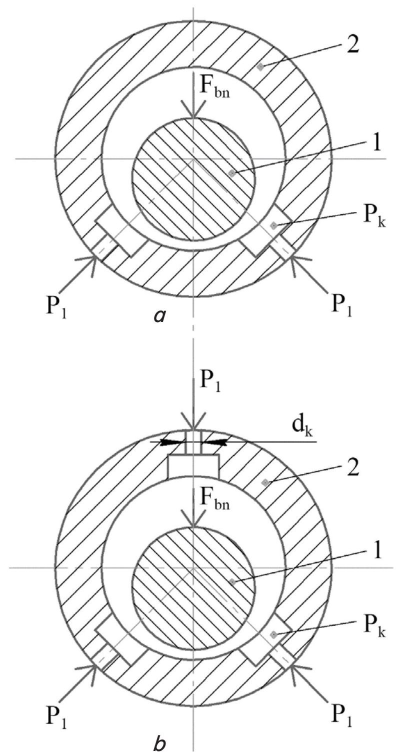

Fig. 2. Diagrams of hydrostatic bearings: 1 – shaft; 2 – hydrostatic bearing

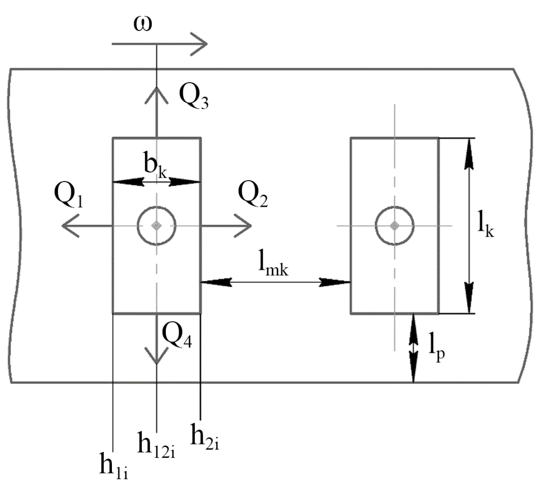

Fig. 3. Sweep of the working surface of a hydrostatic bearing with two chambers

Let's write down the balance of consumption of the working fluid for the *i*-th chamber:

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

where Q_{in} is the flow rate of the working fluid through the inlet compression device; Q_1, Q_2, Q_3, Q_4 are flow rates in the axial and tangential directions.

The flow rate through the inlet compression device (nozzle) is written using a known hydraulics formula [14] in the following form:

$$Q_{in} = \psi_{in} \cdot \pi \cdot r_1^2 \cdot \sqrt{\frac{2}{\rho}\left(P_1 - P_{ki}\right)}, \qquad (2)$$

where ψ_{in} is the inlet coefficient equal to 0.62–0.82 [15]; r_1 is the radius of the nozzle; ρ is the 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 carrier chamber.

The flow rates along the contour of the i-th chamber, taking into account the portable and gradient flow of the working fluid [16], are as follows:

$$Q_1 = -\frac{\omega \cdot R \cdot h_{1i} \cdot l_k}{2} + \frac{h_{1i}^3 \cdot \left(P_{k,i} - P_{k,i-1}\right) \cdot l_k}{12 \cdot \mu \cdot l_{mk}},$$

$$Q_2 = \frac{\omega \cdot R \cdot h_{2i} \cdot l_k}{2} + \frac{h_{2i}^3 \cdot \left(P_{k,i} - P_{k,i+1}\right) \cdot l_k}{12 \cdot \mu \cdot l_{mk}}, \qquad (3)$$

$$Q_3 = Q_4 = \frac{h_{12i}^3 \cdot \left(P_{k,i} - P_{10}\right) \cdot b_k}{12 \cdot \mu \cdot l_p},$$

where h_{1i} is the gap along the left edge of the chamber (Fig. 3); h_{2i} is the gap along the right edge of the chamber; h_{12i} – gap in the middle of the chamber; μ – dynamic viscosity of the working fluid; ω is the angular velocity of rotation of the shaft; R is the radius of the working surface of the shaft; l_k is the length of the chambers; l_p is the length of the end jumper of the chambers; l_{mk} is the length of the interchamber jumper; b_k is the width of the chambers; P_{10} is the pressure of the working fluid at the drain from the bearing.

From the equation of the flow balance of the working fluid (1), let us write the expression for determining pressures in the chambers:

$$\left(P_{k,i}\right)_{n+1} = a_{9i} \cdot \sqrt{P_1 - \left(P_{k,i}\right)_n} +$$
$$+ a_{11i} \cdot \left(P_{k,i-1}\right)_n + a_{12i} \cdot \left(P_{k,i+1}\right)_n + a_{10i}, \qquad (4)$$

where

$$a_1 = \frac{\omega \cdot R \cdot h_{1i} \cdot l_k}{2};\ a_2 = \frac{h_{1i}{}^3 \cdot l_k}{12 \cdot \mu \cdot l_{mk}};\ a_3 = \frac{\omega \cdot R \cdot h_{2i} \cdot l_k}{2};$$

$$a_4 = \frac{h_{2i}{}^3 \cdot l_k}{12 \cdot \mu \cdot l_{mk}};\ a_5 = \frac{h_{12i}{}^3 \cdot b_k}{12 \cdot \mu \cdot l_p};$$

$$a_6 = \psi_{2E} \cdot \pi \cdot r_1^2 \cdot \sqrt{2/\rho};$$

$$a_8 = a_2 + a_4 + 2 \cdot a_5;\ a_{9i} = a_6 / a_8;$$

$$a_{10i} = \frac{a_1 - a_3 + 2 \cdot a_5 \cdot P_1 - a_6}{a_8};$$

$$a_{11i} = a_2 / a_8;\ a_{12i} = a_4 / a_8.$$

Equation (4) is written in a form adapted for carrying out an iterative process. Given the initial values of the pressures in the chambers, using equation (4) yields the pressures in the chambers at n+1 step. The iterative process continues until the specified accuracy is obtained.

To determine the pressures on the interchamber bridges, let us apply the Reynolds equation [17]:

$$\frac{\partial}{\partial x}\left(\frac{h^3}{k_x} \cdot \frac{\partial P}{\partial x}\right) + \frac{\partial}{\partial z}\left(\frac{h^3}{k_z} \cdot \frac{\partial P}{\partial z}\right) = 6 \cdot \mu \cdot \frac{\partial (U \cdot h)}{\partial x}, \qquad (5)$$

where x and y are tangential and axial coordinates; h is the current gap at the inter-chamber jumpers; U is the circumferential speed of the shaft; P is the pressure on the interchamber bridges; μ is the dynamic viscosity of the working fluid.

The Reynolds equation is generalized to the case of turbulent flow of the working fluid using the turbulence coefficients k_x and k_z, which are determined by the Constantinescu method:

$$k_x = 1 + 0.044\left(\sigma_x^2 \cdot \mathrm{Re}\right)^{0.725},$$

$$k_z = 1 + 0.0247\left(\sigma_x^2 \cdot \mathrm{Re}\right)^{0.65},$$

where $\sigma_x = 0.125 \cdot \mathrm{Re}^{0.07}$; $\mathrm{Re} = (U \cdot h)/V$ – Reynolds number; V is the kinematic viscosity of the working fluid.

The current bearing clearance is determined by the following relationship:

$$h = \delta_0 - e \cdot \cos(\varphi - \beta_0),$$

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

The Reynolds equation was solved numerically using the finite difference method in combination with the longitudinal-transverse run method.

Setting the initial values of the pressures in the mesh nodes, the pressure values in the mesh nodes were determined by the longitudinal-transverse run method at the next step. The iterative process continued until the specified accuracy was obtained.

According to the known values of pressures in the chambers and assemblies of the inter-chamber jumpers, the carrying capacity of the bearing was determined. The carrying capacity of the bearing was defined as the sum of the bearing capacities of the chambers, inter-chamber, and end bridges of the bearing in projections onto the line of action of an external force and the direction perpendicular to it. Let's write the expressions for the vertical W_1 and horizontal W_2 components of the carrying capacity as a whole:

$$W_1 = W_{11} + W_{12} + W_{13} + W_{14},$$

$$W_2 = W_{21} + W_{22} + W_{23} + W_{24}, \qquad (6)$$

where W_{11} and W_{21} – load-bearing capacities of the chambers in projections onto the vertical and horizontal axes; W_{12} and W_{22} – load-bearing capacities of inter-chamber jumpers in projections onto the vertical and horizontal axes; W_{13} and W_{23} – load-bearing capacities of the ends of the inter-chamber jumpers in projections onto the vertical and horizontal axes; W_{14} and W_{24} are the load-bearing capacities of the ends of the chamber jumpers in projections onto the vertical and horizontal axes.

The flow rate of the working fluid through the bearing was determined by the known values of pressures in the chambers.

Let's record the flow rate of the working fluid through the hydrostatic bearing

– for a dual-carrier chamber structure:

$$Q = \psi_{in} \cdot \pi \cdot r_1^2 \cdot \sqrt{\frac{2}{\rho}} \cdot \sum_{i=1}^{2} \sqrt{P_1 - P_{ki}}; \quad (7)$$

– for a three-carrier chamber structure:

$$Q = \psi_{in} \cdot \pi \cdot r_1^2 \cdot \sqrt{\frac{2}{\rho}} \cdot \sum_{i=1}^{3} \sqrt{P_1 - P_{ki}}. \quad (8)$$

The flow rate of the working fluid was estimated under the main mode of operation of the gear pump and under the transient mode before entering the main mode.

5. The influence of the structural and performance parameters of the hydrostatic bearing of the gear pump on its performance

5. 1. Identification of the influence of the number of carrying chambers on the carrying capacity and flow rate of the working fluid of a hydrostatic bearing

The theoretical study of the carrying capacity and flow rate of the working fluid of the hydrostatic bearing was carried out on the basis of solving the equations of hydromechanics (4) and (5). Using these equations, the function of the pressure distribution in the working fluid layer was determined. The numerical implementation of the flow balance equation (4) makes it possible to determine the pressures in the carrier chambers in an iterative way. Given the initial values of pressures in the chambers, the following values of pressures in the chambers were determined using the equation of the balance of flows. The obtained values of pressures in the chambers were used for the next iteration as the initial ones. The initial values of pressures in the chambers were determined from the experience of designing hydrostatic bearings. They should be less than the supply pressures. This is due to the fact that the working fluid passes through the inlet pressure compensator of the nozzle installed at the inlet to the carrying chambers. The more accurately the initial values of the pressures in the chambers were set, the fewer iterations were needed to obtain the desired result.

The obtained values of pressures in the chambers and the pressure drop at the drain at the ends of the bearing, taken to be zero, were used as boundary values for determining the pressures at the inter-chamber bridges. The pressures on the interchamber bridges were determined from the solution of the Reynolds equation (5). This equation does not have an exact analytical solution and was solved numerically using the finite difference method. When the Reynolds equation was written in finite-difference form, the surface between the chambers was covered with a regular grid. The partial derivatives in equation (5) were written in finite-difference form using a five-point pattern. By setting the initial values of the pressures in the mesh nodes, using the method of longitudinal-transverse run along the rows and along the columns, the pressures in the mesh 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 of the average value of the pressures in the grid nodes at the first iteration, were used as the initial values for the next iteration. The iteration process continued until the specified accuracy was obtained in all nodes of the mesh.

After multiplying by the area of the chambers, the carrying capacity was calculated. It was defined as the sum of the bearing capacities of the chambers, inter-chamber, and end bridges of the bearing in projections onto the vertical and horizontal axes. On the surface of the chambers, the pressure was assumed to be constant and, after multiplying by the area of the chambers, the carrying capacity of the chambers was determined. The Simpson method was used to calculate the load-carrying capacity of inter-chamber jumpers. The flow rate of the working fluid through the bearing was determined according to formulas (7), (8) on the basis of the calculated pressures in the chambers.

The calculation of the carrying capacity and flow rate of the working fluid for hydrostatic bearings with two and three carrying chambers was carried out at the following values of structural and operational parameters:

1. Bearing diameter D_p=14.5 mm.
2. Shaft speed ω=855 s^{-1}.
3. The feeding pressure of the bearing P_1=8 MPa.
4. Bearing length L_p=13 mm.
5. The load on one bearing of the driven gear F_1=1024.2 N.
6. The load on one bearing of the drive gear F_2=669.5 N.
7. Working fluid kerosene TC-1 at a temperature of 100 °C.
8. Radial gap δ_0=22.5 μm.
9. The length of the chambers l_k=9 mm.
10. The width of the chambers b_k=4 mm.
11. The angle of the chambers relative to the vertical is 35°.
12. It is necessary to ensure the minimum clearance between the shaft and the bearings $h_{\min}$=4 μm.

The results of calculations of hydrostatic bearings with two and three carrying chambers are shown in Fig. 4, 5.

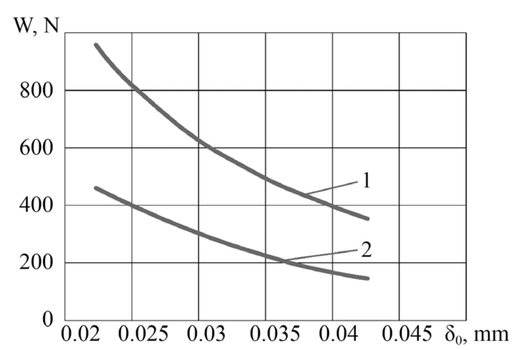

Fig. 4. Dependence of the carrying capacity of hydrostatic bearings on the radial clearance: 1 – bearing with two carrying chambers; 2 – bearing with three carrying chambers

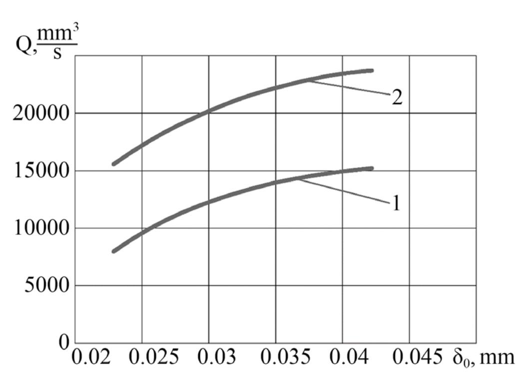

Fig. 5. Dependence of the flow rate of hydrostatic bearings on the radial clearance: 1 – bearing with two carrying chambers; 2 – bearing with three carrying chambers

Fig. 4 shows that with all the considered clearances in a hydrostatic bearing, the carrying capacity of a bearing with two carrying chambers is greater in comparison with the carrying capacity of a bearing with three chambers. With a bearing clearance of 0.0225 mm, the carrying capacity of a two-chamber bearing is about 2.16 times larger than a three-chamber bearing, and with a clearance of 0.0425 – 2.6 times. The flow rate of the working fluid of a two-chamber bearing is less than the flow rate of the working fluid of a three-chamber bearing (Fig. 5) for all considered clearances in the bearing. With a bearing clearance of 0.0225 mm, the working fluid consumption in a two-chamber bearing is 1.88 times less than a three-chamber bearing, and with a clearance of 0.0425 – 1.59 times.

The above analysis shows that a hydrostatic bearing with two carrying chambers has a larger carrying capacity and lower fluid consumption compared to a three-chamber bearing. For further analysis of the characteristics of the hydrostatic bearing for the gear-type aviation fuel pump, a two-carrier chamber scheme was adopted.

5. 2. Determining the diameter of the nozzle that provides the necessary carrying capacity of the bearing and evaluation of the flow characteristics of the pump

The results of calculating the carrying capacity of a hydrostatic bearing at different values of the nozzle diameter are shown in Fig. 6.

When constructing the plot, different values of the nozzle diameter were set in the initial data. In formula (2), when calculating the flow rate through the inlet compensating device (nozzle), the radius of the nozzle is used. After solving the flow balance equations (4) and Reynolds equations (5) together, the pressure distribution function over the working surface of the bearing was calculated. By integrating the grid function of pressure distribution according to formulas (6), the carrying capacity of the bearing was determined. The plot was built by setting various numerical values of the nozzle diameter, on the basis of which the carrying capacity of the bearing was determined.

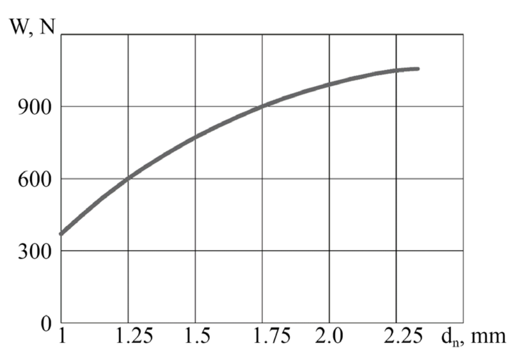

Fig. 6. Dependence of the carrying capacity of a hydrostatic bearing on the diameter of the nozzle

Fig. 6 shows that the dependence of the carrying capacity of the bearing on the diameter of the nozzle is non-linear. The required carrying capacity is provided with a nozzle diameter of 2.3 mm and is equal to 1024 N.

The flow rate of the working fluid through the hydrostatic bearing is calculated according to dependence (2). For a nozzle diameter of 2.3 mm, it was equal to 9045 mm^3/s. It is very important for any pump, including gear one, to ensure the necessary volumetric flow characteristics. Therefore, it was very important to assess the quantitative selection of the working fluid (fuel) for the operation of hydrostatic bearings. Under the main mode of operation of the fuel gear pump, the fuel extraction for the operation of the hydrostatic bearing was 1.3 % of the pump flow rate equal to 681900 mm^3/s.

The above analysis shows that hydrostatic bearings can be recommended as shaft supports for gear-type fuel pumps.

6. Discussion of results of investigating the characteristics of two-chamber and three-chamber hydrostatic bearings

A feature of the proposed method is the possibility of using hydrostatic bearings operating on the fuel pumped by the pump as shaft supports for gear-type pumps. An unconventional layout of the carrying chambers on the working surface of a hydrostatic bearing is proposed. In this statement, the calculation procedure in [1–13] was not considered. The results make it possible to recommend hydrostatic bearings with an unconventional arrangement of carrying chambers on its working surface as shaft supports for gear-type fuel pumps. The results of the calculation of the carrying capacity and flow rate of the working fluid are shown in Fig. 4–6.

Fig. 4 shows that with all accepted values of clearances in the bearing, the carrying capacity of a two-chamber bearing is approximately 2.16–2.6 times greater than a three-chamber bearing. This is due to the fact that in a three-chamber bearing, the third upper chamber creates a load that reduces the load capacity of the lower two chambers.

Fig. 5 shows that with all the considered values of clearances in the bearing, the flow rate of the working fluid in a two-chamber bearing is less than in a three-chamber bearing by about 1.59–1.88 times. This is due to the fact that an additional third chamber is made in the three-chamber bearing, into which the working fluid is supplied. In accordance with the terms of reference, the load acting on the bearings is constant in magnitude and direction. Therefore, the design of the working surface of the hydrostatic bearing with two carrying chambers was adopted and the problem was solved in a stationary setting. However, without dynamic evaluation, modern high-speed machines cannot be recognized as fully operational. Therefore, after solving stationary problems, it is planned to conduct a study in a dynamic statement. After that, it will be possible to make a final decision on setting the structural parameters for the hydrostatic bearing. The 35° chamber angle can also be adjusted after solving a dynamic problem.

Fig. 6 shows that the required carrying capacity of the hydrostatic bearing of 1024 N is provided with a nozzle diameter of 2.3 mm.

The calculation of the flow rate of the working fluid through the bearing showed that under the main mode of operation of the fuel gear pump, the fuel extraction for the operation of the hydrostatic bearing was 1.3 % of the pump flow rate equal to 681900 mm^3/s.

The proposed solutions make it possible to recommend hydrostatic bearings as shaft supports for gear-type fuel pumps. The new solution can be applied in addition to the considered gear pumps, also in rolling mills, additional supports for the faceplate of lathes, optical devices, radio telescopes, radar antennas, and hydrostatic guides.

The advantage of this study is an integrated approach associated with the solution of a complex hydrodynamic problem. The proposed method makes it possible to perform practical calculations of hydrostatic bearings for gear-type fuel pumps.

The limitations inherent in this study are related to the need to clean the fuel with filters in order to reduce the likelihood of clogging of the nozzles installed at the bearing inlet.

The disadvantage of this study is the failure to take into account the thermal phenomena in the bearing, which arise from power losses due to friction and pumping of the working fluid.

The development of this study may consist in the analysis of the dynamics of the pump shaft on hydrostatic bearings.

7. Conclusions

1. It has been established that the carrying capacity of a two-chamber hydrostatic bearing is approximately 2.16–2.6 times greater than the carrying capacity of a three-chamber bearing. It has been established that the flow rate of the working fluid in a two-chamber hydrostatic bearing is approximately 1.59–1.88 times less than in comparison with a three-chamber bearing. After solving the problem in a dynamic setting, it will be possible to make a final decision on the assignment of the structural parameters for the hydrostatic bearing.

2. It has been established that the required carrying capacity of the hydrostatic bearing of 1024 N is provided with a nozzle diameter equal to 2.3 mm. It was revealed that under the main mode of operation of the gear-type fuel pump, the selection for the operation of the hydrostatic bearing was 1.3 % of the pump flow rate equal to 681900 mm^3/s.

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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