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IDENTIFYING THE INFLUENCE OF DESIGN PARAMETERS OF SINGLE-CHAMBER HYDROSTATIC BEARING OF FUEL PUMP ON ITS MAIN CHARACTERISTICS

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The object of this study is hydrostatic processes in the fluid friction bearings of gear-type aviation fuel pumps.

The problem of the influence of the design parameters of a single-chamber hydrostatic bearing on its main characteristics was solved. The main characteristics were considered to be the load-bearing capacity and the flow rate of the working fluid. When determining the main characteristics of a single-chamber hydrostatic bearing, the Reynolds and flow balance equations were solved jointly. The resulting diagram of pressure distribution over the working surface of the bearing was used to determine the main characteristics.

The influence of the clearance, nozzle diameter, and chamber width on the load-bearing capacity and working fluid flow of a single-chamber hydrostatic bearing was studied.

It has been established that as the gap increases, the load-bearing capacity of a single-chamber bearing decreases, and the flow rate of the working fluid increases. As the nozzle diameter increases, the bearing's load-bearing capacity increases. Increasing the width of the chambers leads to an increase in the load-bearing capacity and flow of working fluid through the bearing. When the gap increases from 0.0125 mm to 0.0425 mm, the bearing capacity decreases by 1.156 times. The flow rate of working fluid through the bearing increases by 1.4 times. With an increase in the nozzle diameter from 1.5 mm to 3 mm, the bearing capacity increases slightly by approximately 1.02 times. Increasing the width of the chambers from 4 mm to 8 mm increases the load-bearing capacity by 1.29 times and increases the flow rate of working fluid by 1.4 times.

The results show that a single-chamber hydrostatic bearing can provide the required load-bearing capacity by selecting design parameters. The given mathematical dependences could be used for practical calculations of single-chamber hydrostatic bearings

Keywords: single-chamber bearing, nozzle diameter, chamber width, load-bearing capacity, lubricant consumption

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1. Introduction

The fuel pump is a very important component of an aircraft engine. Due to a number of advantages, gear pumps are becoming widespread. Gear or toothed pumps are one of the most common and simple hydraulic machines. The most popular are gear pumps with external gearing. Gear pumps can be used to pump liquids of varying viscosities. They have high efficiency, which reaches 90 %. The structure of a gear pump is simple. Therefore, it is easy to maintain and has a compact design. Gear pumps operate with a filtration fineness of no worse than 100 microns. This is the best indicator among all positive displacement pumps. Rolling bearings are widely used as shaft supports in gear pumps. With the increase in the rotation speed of the shafts of modern machines, rolling bearings do not provide the necessary service life and reliability of the machines. Rolling bearings are being replaced by fluid friction sliding bearings. In the classification of fluid friction sliding bearings, hydrostatic bearings occupy an important place. One of the main advantages of hydrostatic bearings is the ability to use kerosene as a lubricant for the working fluid of the machine. Kerosene in fuel pumps is under high pressure and can be used as a working fluid in the bearing. Hydrostatic bearings enable reliable operation of the pump under various operating modes.

Designing reliable hydrostatic bearings requires extensive theoretical and experimental research into their basic characteristics. The lack of necessary information on the design of hydrostatic bearings with different structures of their working surface necessitates research into this area.

2. Literature review and problem statement

Study [1] addresses the problem of uncertain lubrication characteristics and tribological characteristics of an aircraft gear pump support bearing under severe operating conditions. The oil supply pressure was increased and the pressure change in the oil film was analyzed. At the same time, the eccentric load was simultaneously applied and increased. It has been shown that with increasing speed, the pressure inside the oil film increases and the bearing capacity increases significantly. In addition, it is shown that eccentric loading can increase the stiffness and damping effects. However, the cited work does not pay attention to the dynamic phenomena inside the bearing, which can be significant under eccentric loading. Paper [2] shows that aircraft fuel pump support bearings operate under extreme conditions of high temperature, high pressure, and low lubricant viscosity environments. The main cause of lubrication failure and abnormal wear is the breakdown of the thin lubricating film. The paper reports a study on ultrasonic measurement of the dynamic thickness of the oil film of a fuel pump. The film thickness in the plain bearing was measured using a complex model and a resonant model. Measurements were carried out under various operating conditions. However, the work does not pay attention to the use of hydrostatic effects inside the lubricant, which do not allow the temperature of the lubricant to increase significantly. In [3], an elastic-hydrodynamic lubrication model was built to study the reliability of hydrodynamic lubrication of support bearings inside aircraft gear pumps operating at high speeds and with low viscosity of the medium. This model takes into account the elastic deformation of the bearing bushing by jointly solving the Reynolds lubrication equation with the influence matrix. The devised method was used to calculate the reliability and sensitivity of the hydrodynamic characteristics of lubrication of plain bearings. However, the cited work does not pay attention to the influence of lubricant temperature on the reliability of the hydrodynamic characteristics of plain bearings. Work [4] discusses gear pumps for fuel systems of aircraft engines. The radial load applied to the gears is fully absorbed by the hybrid support bearings. They are lubricated using low-viscosity aviation fuel. Considering the operating conditions of these bearings, elastic deformations have a significant impact on their performance and equilibrium position. A numerical model was built for the design of hybrid support bearings for fuel pumps. However, the cited work does not pay attention to the influence of the centrifugal force from shaft imbalance on the pressure distribution in the lubricating film. Work [5] describes modeling and experimental substantiation of the movement of a floating bearing sleeve in a pump with an external gear. Connection graphs were used to model the behavior of a gear pump. The devised model was experimentally tested in laboratory and field tests by installing a pump in a drilling machine. However, the work does not pay attention to the deformation of the sleeve on the nature of its movement. In [6], a new simulation model of a gear pump with support bearings is considered. The built model consists of various models: pump hydrodynamic model, bearing lubrication model, shaft motion estimation model and geometric model. However, the work does not pay attention to the use of hydrostatic effects in the model built, which would eliminate the influence of surface roughness on the operation of the sliding bearing. Paper [7] studied the transient lubrication characteristics of hydrodynamic journal bearings during the external gearing cycle of a gear pump. A numerical model is constructed to evaluate transient lubrication during initial startup of an aircraft gear pump. The flow inside the pump is analyzed using a computational fluid dynamics model. However, the work does not pay attention to the analysis of wear of the working surfaces of the bearing under the modes of starting and stopping the pump shaft, and wear changes the clearance in the bearing and its characteristics. In [8], the study of the characteristics of the starting transition lubricant and the behavior of the plain bearing in a gear pump is reported. The results showed that the load fluctuation caused by the change in the transition lubricant layer has a noticeable effect on the movement of the pump shaft during startup. However, the work does not pay attention to the use of nonlinear equations of motion of the shaft inside the bearing but only considers the micromotion of the center of the shaft axis. Under start and stop modes, the bearing operates at large eccentricities when its characteristics are nonlinear. In [9], an increase in the temperature of support bearings in a gear-type pump was considered. The temperature rise of the journal bearings in the pump was assessed for bearings with and without herringbone grooves. The results obtained by using a herringbone groove on the inner wall of the bearing were effective in reducing the temperature rise. However, the work does not pay attention to the use of hydrostatic bearing designs, which can significantly reduce the temperature of the lubricant. Work [10] examines the transient characteristics of the lubrication of plain bearings inside an aviation fuel gear pump under complex variable loads. A model for calculating transient processes has been built. On this basis, an analysis of the stability of the trajectory of the center of the shaft axis and an analysis of the lubrication characteristics during transient processes were carried out. However, the cited work does not pay attention to the use of nonlinear equations of motion of the rotor inside the bearing. Using the trajectories of the shaft movement, one can more accurately analyze the stability of its movement. Work [11] considers the problem of lubrication of an aircraft pump gear shaft under high load conditions. A three-dimensional model for calculating thermohydrodynamic lubrication of a bearing film and a model for contact of microroughnesses of a rough surface have been constructed. The transient field of the gear pump was analyzed based on the calculation of the orbit of the radial motion of the gear shaft. The calculation results are compared with experimental data. However, the cited work does not pay attention to the influence of temperature on the dimensions of the bearing and shaft, which leads to a change in the clearance in the bearing and a change in its characteristics. Work [12] considers modeling the operation of a high-speed turbopump in a liquid-propellant rocket engine. A model is proposed that takes into account the misalignment coefficient for a system with a double rotor connecting a gear coupling and a rolling bearing. The influence of misalignment parameters on the dynamic characteristics (critical speed and vibration response) of the tested rotor was obtained by numerically solving the coupled model built. However, the work does not pay attention to replacing the rolling bearing with a plain bearing. This would significantly improve the dynamic characteristics of a high-speed turbopump. In [13], the problem of the influence of the design parameters of a fuel pump bearing on its static characteristics was considered. Load-bearing capacity, lubricant consumption, and operating temperature conditions were considered as static characteristics. The design of a hydrostatic bearing with two load-bearing chambers on the working surface was considered. Three options for the circumferential arrangement of chambers relative to the line of action of the external load were studied. A comprehensive assessment of the effect of increasing the temperature of the working fluid on the consumption of lubricant and bearing capacity is given. It has been established that the angular arrangement of chambers in a bearing can have a significant impact on the static characteristics of a hydrostatic bearing. However, the work does not pay attention to the analysis of the influence of the width of the chambers on the static characteristics of a hydrostatic bearing. Work [14] examines the analysis of various possible design schemes for the working surface of a hydrostatic bearing and proposes the most rational scheme. The geometric and operating parameters of the hydrostatic bearing were selected to enable the necessary load-bearing capacity and acceptable flow rates of the working fluid. The dependence of the bearing load-bearing capacity on the nozzle diameter was derived. Some recommendations have been devised for the design of hydrostatic bearings for fuel pumps. However, the work does not pay attention to the design of a bearing with one supporting chamber.

The literature [1–14] lacks information on studying the design of a hydrostatic bearing with one load-bearing chamber. In addition, there is no analysis of the characteristics of a hydrostatic bearing at different chamber widths.

3. The aim and objectives of the study

The purpose of this work is to identify the influence of the design parameters of a single-chamber hydrostatic bearing of a fuel gear pump on its main characteristics. This makes it possible to establish the possibility of using a single-chamber bearing in the fuel pump supports.

To achieve the goal, the following tasks were set:

– to identify the influence of the gap in a single-chamber hydrostatic bearing on the flow rate of the working fluid and its load-bearing capacity;

– to establish the influence of the width of the chambers in a single-chamber hydrostatic bearing on the flow of working fluid and its load-bearing capacity.

4. The study materials and methods

The object of my study is hydrostatic processes in the fluid friction bearings of gear-type aviation fuel pumps. For a hydrostatic bearing with one load-bearing chamber, it is proposed to study the influence of the clearance and width of the chamber on the flow rate of the working fluid and the load-bearing capacity of the bearing.

When constructing mathematical dependences, assumptions generally accepted in the hydrodynamic theory of lubrication were adopted. The inertial terms in the Navier-Stokes equations were assumed to be small in comparison with the viscous ones. The external load on the bearing was assumed to be constant. When building mathematical relationships for calculating the characteristics of a single-chamber hydrostatic bearing, methods of fluid mechanics were used. The pressure distribution over the working surface of the bearing was the basis for determining the main characteristics of the bearing.

The numerical implementation of the constructed mathematical dependences was carried out in the Excel software (developed by Microsoft, USA).

The diagram of the considered single-chamber hydrostatic bearing of the fuel pump is shown in Fig. 1, 2.

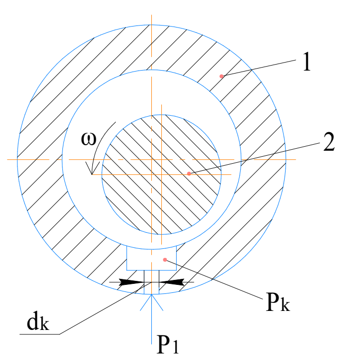

Fig. 1. Diagram of a single-chamber hydrostatic bearing

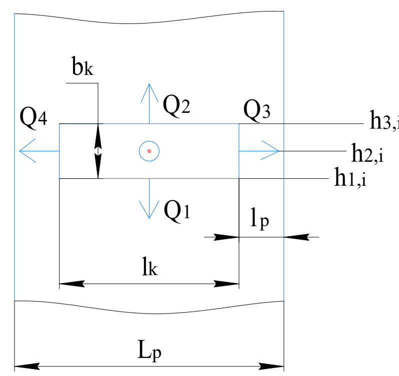

Fig. 2. Sweep of the working surface of a hydrostatic bearing

In the above diagram of hydrostatic bearing 1, there is one supporting chamber on the working surface, the pressure in which is designated P_K. The chamber is located at the bottom of the bearing along the line of action of the external force. Shaft 2 rotates inside the bearing with angular velocity ω. The working fluid is supplied to the carrier chamber under high pressure P_1. A nozzle with a small diameter d_K is installed at the entrance to the chamber. The working fluid of the pump, in this case kerosene, is used as the working fluid. Through the ends of the bearing, the working fluid flows to the drain.

The basis for determining the load-bearing capacity of the bearing and the flow rate of the working fluid was the function of pressure distribution over the working surface of the bearing. The pressure distribution was determined from the numerical solution of the Reynolds equations and the flow balance. The balance of working fluid flow rates was recorded from the condition of equality of flow rates through the input compensating device (nozzle) and flow rates along the chamber contour:

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

where Q_v is the flow rate of working fluid through the input compensating device;

Q_1, Q_2, Q_3, Q_4 – flow rates of the working fluid along the chamber contour.

Write the flow rate through the input compensating device using the well-known hydraulic formula [15] in the following form:

$$Q_v = \psi_v \cdot \pi \cdot r_1^2 \cdot \sqrt{\frac{2}{\rho}\left(P_1 - P_k\right)}, \tag{2}$$

where ψ_v is the input coefficient equal to 0.62–0.82;

r_1 – nozzle radius;

ρ – density of the working fluid;

P_1 – supply pressure at the inlet to the nozzle;

P_k is the pressure in the chamber.

The flow rates along the chamber contour were recorded taking into account the portable and gradient flow of the working fluid [16]:

$$\begin{aligned} Q_1 &= -\frac{\omega \cdot R \cdot h_1 \cdot l_k}{2} + \frac{h_1^3 \cdot P_k \cdot l_k}{12 \cdot \mu \cdot l_{mk}}, \\ Q_2 &= \frac{\omega \cdot R \cdot h_2 \cdot l_k}{2} + \frac{h_2^3 \cdot P_k \cdot l_k}{12 \cdot \mu \cdot l_{mk}}, \\ Q_3 &= Q_4 = \frac{h_{12}^3 \cdot (P_k - P_{10}) \cdot b_k}{12 \cdot \mu \cdot l_p}, \end{aligned} \tag{3}$$

where h_1, h_2, h_{12} are clearances in the bearing (Fig. 2);

μ – dynamic viscosity of the working fluid;

ω – angular speed of shaft rotation;

R – radius of the working surface of the shaft;

l_k – chamber length;

l_p – length of the end separator of the chambers;

l_{mk} – length from the edge of the chamber to the edge of the lubricating layer;

b_k – chamber width;

P_{10} – working fluid pressure at the bearing drain.

From the equation for the balance of working fluid flow rates (1) it follows that the expression for determining the pressure in the chamber takes the form:

$$P_k = -\frac{a_9}{2} + \sqrt{\left(\frac{a_9}{2}\right)^2 - a_{10}},$$

where:

$$a_1 = \frac{\omega \cdot R \cdot h_1 \cdot l_k}{2}; \quad a_2 = \frac{h_1^3 \cdot l_k}{12 \cdot \mu \cdot l_{mk}};$$

$$a_3 = \frac{\omega \cdot R \cdot h_2 \cdot l_k}{2}; \quad a_4 = \frac{h_2^3 \cdot l_k}{12 \cdot \mu \cdot l_{mk}};$$

$$a_5 = \frac{h_{12}^3 \cdot b_k}{12 \cdot \mu \cdot l_p}; \quad a_6 = \psi_{in} \cdot \pi \cdot r_1^2 \cdot \sqrt{2/\rho};$$

$$a_7 = -a_1 + a_3; \quad a_8 = a_2 + a_4 + a_5;$$

$$a_9 = \frac{2 \cdot a_7 \cdot a_8 + a_6^2}{a_8^2}; \quad a_{10} = \frac{a_7^2 - a_6^2 \cdot P_1}{a_8^2}.$$

To determine the pressures on separators, the Reynolds equation was written [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 the circumferential and axial coordinate axes in the bearing;

h – current gap in the lubricant layer;

U – shaft peripheral speed;

P – pressure in the lubricant layer.

The Reynolds equation is generalized to the case of turbulent flow of the working fluid using the turbulence coefficients k_x and k_z. Turbulence coefficients were determined using the well-known Constantinescu method. The Reynolds equation was solved numerically using the finite difference method.

The bearing capacity was determined as the sum of the bearing capacities of the chambers, circumferential and end separators of the bearing:

$$W_p = W_k + W_{pk} + W_{pmk}, \qquad (6)$$

where W_K is the load-bearing capacity of the chambers;

W_{PK} – load-bearing capacity of the end separators of the chamber;

W_{PMK} is the load-bearing capacity of the bearing circumferential separators.

The flow rate of the working fluid was determined by the calculated pressure in the chamber.

$$Q = \psi_v \cdot \pi \cdot r_1^2 \cdot \sqrt{\frac{2}{\rho}} \cdot \sqrt{P_1 - P_k}, \qquad (7)$$

where P_1 is the supply pressure of the working fluid;

P_K – pressure in the chamber.

The formulas above make it possible to calculate the bearing capacity and the flow rate of the working fluid.

5. Results of investigating the load-bearing capacity and flow rate of working fluid of a single-chamber hydrostatic bearing

5. 1. Results of investigating the main characteristics of a single-chamber bearing at various clearance values

The study of the load-bearing capacity and flow rate of the working fluid of a single-chamber bearing was carried out on the basis of a joint solution of the Reynolds equations and a flow balance. The presence of one chamber on the working surface of the bearing made it possible to determine the pressure in the chamber in a non-iterative manner. The Reynolds equation was solved numerically using the finite difference method. The boundary conditions when solving the Reynolds equations were the pressure in the chamber and the pressure difference across the drain at the ends of the bearing. Partial derivatives in the Reynolds equation were written as finite differences using a five-point template. Given the initial pressure values at the grid nodes, the pressures were determined iteratively with a pre-defined accuracy.

After determining the pressure distribution function over the working surface of the bearing, the load-bearing capacity of the bearing and the flow rate of the working fluid were calculated.

The calculation of the load-bearing capacity and flow rate of the working fluid of a single-chamber hydrostatic bearing was carried out with the following values of design parameters:

1. Bearing diameter D_p=14.5 mm.
2. Shaft rotation frequency ω=855 s^{-1}.
3. Bearing supply pressure P_1=8 MPa.
4. Bearing length L_p=13 mm.
5. Working fluid kerosene TC–1 at a temperature of 100 °C.
6. Chamber length l_p=9 mm.
7. Number of chambers k=1.

The results of calculating the load-bearing capacity and flow rate of the working fluid of a single-chamber hydrostatic bearing of a fuel pump at various clearances are shown in Fig. 3, 4.

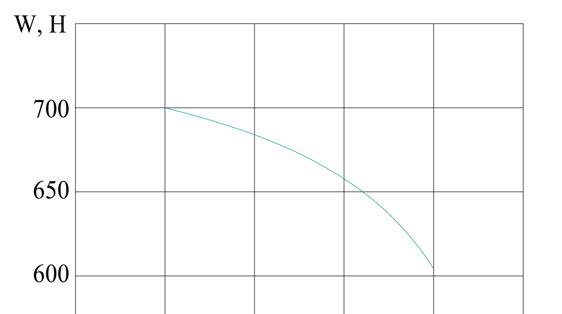

Fig. 3. Dependence of the load-bearing capacity of a single-chamber bearing on clearance

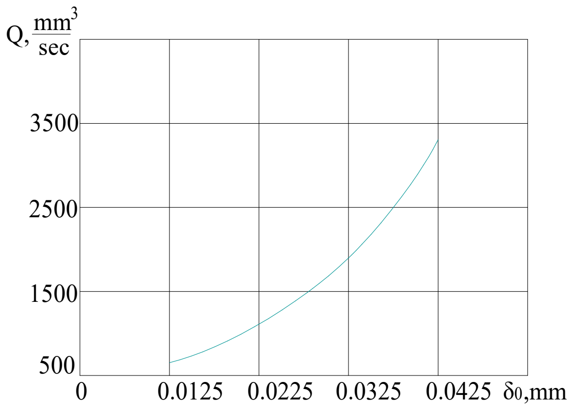

Fig. 4. Dependence of working fluid flow rate in a single-chamber hydrostatic bearing on clearance

Fig. 3, 4 clearly demonstrate that as the gap in the bearing increases, its load-bearing capacity decreases, and the flow rate of the working fluid increases. When the gap increases from 0.0125 mm to 0.0425 mm, the bearing capacity decreases slightly by 1.156 times. The flow rate of working fluid through the bearing increases by 1.4 times. The law of change in these characteristics is nonlinear.

Work [14] reports the results of calculating the load-bearing capacity of bearings with two and three load-bearing chambers. With a bearing clearance of 0.0225 mm, the load-bearing capacity of a double-chamber bearing is approximately 1.38 times higher than a single-chamber bearing. The load-bearing capacity of a three-chamber bearing is 1.46 times less than a single-chamber bearing with the same clearances.

The flow rate of working fluid in a single-chamber bearing is the smallest of the three considered designs of working surfaces.

An important factor in ensuring the required load-bearing capacity of a hydrostatic bearing is the diameter of the nozzle. Fig. 5 shows the dependence of the load-bearing capacity of a single-chamber hydrostatic bearing on nozzle diameter.

Fig. 5 shows that with an increase in the nozzle diameter, the load-bearing capacity of a single-chamber bearing increases slightly by approximately 1.02 times.

The above analysis reveals that the load-bearing capacity of a single-chamber bearing is greater than that of a three-chamber bearing and less than that of a double-chamber bearing. The working fluid consumption of a single-chamber bearing is minimal compared to three-chamber and two-chamber hydrostatic bearings.

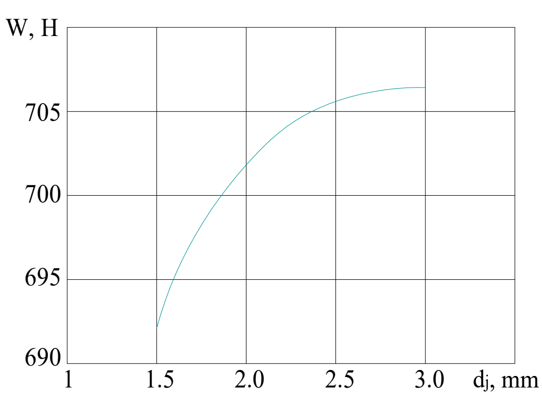

Fig. 5. Dependence of the load-bearing capacity of a single-chamber hydrostatic bearing on the diameter of the nozzle

5. 2. Results of investigating the load-bearing capacity and flow rate of working fluid at different chamber widths

For a single-chamber hydrostatic bearing, studying the influence of the width of the chambers on its main characteristics is very important. Also, due to the width of the chambers, the load-bearing capacity of a single-chamber bearing can be increased. The results of calculating the pressures in the chambers and the load-bearing capacity of a single-chamber bearing at different chamber widths are shown in Fig. 6, 7.

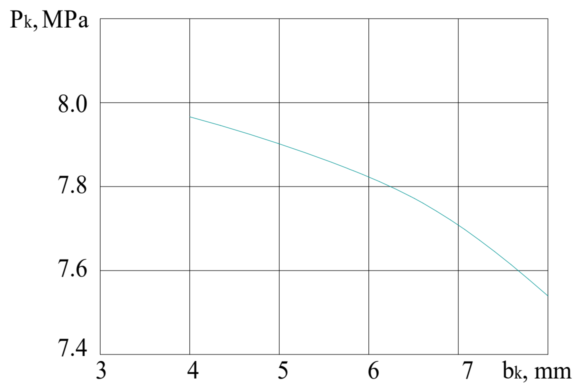

Fig. 6. Dependence of pressure in the chamber of a single-chamber hydrostatic bearing on the width of chambers

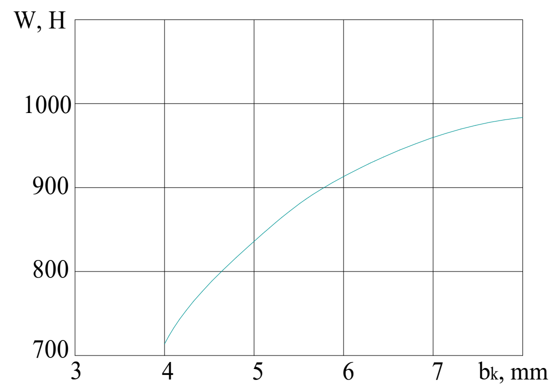

Fig. 7. Dependence of the load-bearing capacity of a single-chamber hydrostatic bearing on the width of chambers

Fig. 6 demonstrates that as the width of the chamber increases, the pressure in it decreases. The decrease in pressure in the chamber is explained by an increase in the flow of working fluid through the end separators of the chamber. With an increase in the chamber width from 4 mm to 8 mm, the pressure in the chamber decreases by 1.05 times. Fig. 7 demonstrates that with increasing chamber width, the load-bearing capacity of a single-chamber hydrostatic bearing increases. When the chamber width increases from 4 mm to 8 mm, the bearing capacity increases by 1.38 times. The increase in bearing capacity with increasing chamber width is explained by the fact that the pressure in the chamber decreases slightly by 1.05 times, and the chamber area increases significantly.

As the chamber width increases, the consumption of working fluid increases. This can be seen in Fig. 8. When the width of the chambers increases from 4 mm to 8 mm, the consumption of working fluid increases by 2.8 times.

Analysis of the results reveals that the load-bearing capacity of a single-chamber hydrostatic bearing is greater than a three-chamber one and less than a two-chamber one. The flow rate of working fluid in a single-chamber bearing is the minimum of the three considered designs of bearing working surfaces. By selecting the nozzle diameter and chamber width, the required load-bearing capacity of a single-chamber bearing can be ensured. Therefore, the considered single-chamber hydrostatic bearing can be recommended for practical use.

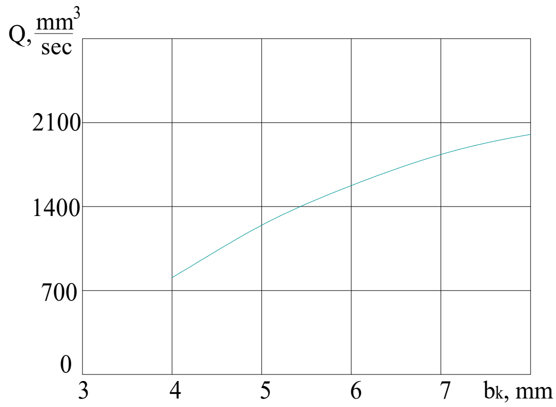

Fig. 8. Dependence of the working fluid flow rate in a single-chamber hydrostatic bearing on the width of chamber

6. Discussion of results of investigating the characteristics of a hydrostatic bearing with one supporting chamber

A special feature of the proposed calculation procedure is assessing the possibility of using single-chamber hydrostatic bearings as shaft supports for gear-type fuel pumps. A scheme of a hydrostatic bearing with one supporting chamber located under the line of action of the external force is proposed. The above analysis of the influence of the design parameters of a single-chamber hydrostatic bearing on its characteristics was not considered in [1–17]. The calculation results show the possibility of using a single-chamber hydrostatic bearing in fuel pump supports. The results of calculating the load-bearing capacity, pressure in the chamber, and flow rate of the working fluid are shown in Fig. 3–8.

Fig. 3, 4 clearly demonstrate that with an increase in the gap in a single-chamber bearing from 0.0125 mm to 0.0425 mm, the load-bearing capacity decreases by 1.156 times, and the flow rate of the working fluid increases by 1.4 times. The decrease in load-bearing capacity is explained by an increase in the flow of working fluid through the increasing gap. Fig. 5 shows that with an increase in the nozzle diameter, the load-bearing capacity of a single-chamber bearing increases slightly by approximately 1.02 times. The increase in the load-bearing capacity of a single-chamber bearing is explained by an increase in pressure in the chamber.

Fig. 6, 7 clearly demonstrate that with an increase in the width of the chamber from 4 mm to 8 mm, the pressure in it decreases by 1.05 times, and the bearing capacity increases by 1.38 times. The increase in the load-bearing capacity of a single-chamber bearing is explained by the fact that the pressure in the chamber decreases slightly with increasing chamber width, and the chamber area increases significantly.

Fig. 8 shows that with an increase in the chamber width from 4 mm to 8 mm, the flow rate of the working fluid increases by 2.8 times. This is explained by the fact that as the width of the chambers increases, the outflow resistance decreases.

The proposed solution and results allow me to recommend a single-chamber hydrostatic bearing for gear-type fuel pumps.

The advantage of this study is an integrated approach associated with solving a complex hydromechanical problem. The proposed method allows for practical calculations of hydrostatic bearings for gear-type aviation fuel pumps.

A bearing with the given dimensions can be used in low- and medium-power aviation gas turbine engines.

A limitation inherent in this study is the need for thorough cleaning of the working fluid to avoid clogging of the nozzles installed at the inlet to the chambers. For other initial bearing parameters, the results and conclusions obtained in this work will not change qualitatively, and the difference can only be in quantitative indicators. When changing the dimensions of a bearing, the quantitative indicators of the bearing characteristics may change, but the qualitative indicators will not change.

The disadvantage of this study is that it does not take into account thermal and force deformations of the bearing working surfaces when calculating the bearing load-bearing capacity and working fluid flow.

The current study may be advanced by taking into account the non-stationary loading of a hydrostatic bearing. It is planned to conduct a comparative analysis with other analogs of the type of bearing considered in the present study.

7. Conclusions

1. It has been established that when the gap increases from 0.0125 mm to 0.0425 mm, the load-bearing capacity of a single-chamber hydrostatic bearing decreases by 1.156 times. The flow rate of working fluid through the bearing increases by 1.4 times.

2. It has been found that increasing the width of the chambers from 4 mm to 8 mm increases the load-bearing capacity of a single-chamber hydrostatic bearing by 1.29 times and increases the flow rate of working fluid by 1.4 times.

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.

Use of artificial intelligence

The author confirms that he did not use artificial intelligence technologies when creating the current work.

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