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DETERMINING THE INFLUENCE OF WORKING FLUID TEMPERATURE CHANGE ON THE CHARACTERISTICS OF A SINGLE-CHAMBER HYDROSTATIC BEARING AT DIFFERENT VALUES OF DESIGN PARAMETERS

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The object of this study is hydrostatic processes in a single-chamber hydrostatic bearing of an aviation gear-type pump.

The task addressed is the influence of design parameters of a single-chamber bearing on its characteristics, taking into account changes in the temperature of the working fluid. The main characteristics considered are the carry-load capacity and flow rate of the working fluid. Determining these characteristics is associated with the joint solution of the Reynolds equations and the flow balance. The basic characteristics of the hydrostatic bearing were determined on the basis of the obtained pressure distribution function in the working fluid layer.

The influence of the eccentricity and diameter of the hydrostatic bearing on its characteristics has been studied, taking into account changes in the temperature of the working fluid.

It was found that at zero eccentricity the temperature of the working fluid increased by 5.6 °C and was 105.6 °C. At an eccentricity of 0.018 mm, the temperature of the working fluid increased by 15.6 °C and was 115.6 °C. With an increase in eccentricity from 0 mm to 0.018 mm, the maximum increase in the working fluid consumption in a single-chamber hydrostatic bearing due to an increase in temperature was 19 %. The maximum decrease in the bearing load carrying capacity due to an increase in the working fluid temperature was 17 %; and at a working eccentricity of 0.018 mm, it did not exceed 1.83 %.

With an increase in the bearing diameter, the working fluid temperature increased. At a bearing diameter of 14.5 mm, the increase in the working fluid temperature was 4.59 °C; and at a diameter of 43.5 mm, the working fluid temperature increased by 15.6 °C.

The results demonstrate that an increase in the working fluid temperature with an increase in eccentricity and bearing diameter has a negligible effect on its load carrying capacity and working fluid consumption

Keywords: single-chamber bearing, working fluid temperature, load carrying capacity, eccentricity, lubricant consumption

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1. Introduction

The performance of an aircraft engine depends on the reliability of the fuel pump. Gear-type fuel pumps are most widely used in aviation. They have a simple design, high efficiency, and are easy to use. Gear pumps are used both for pumping fuel and for supplying oil to the moving parts of the machine. For many years, rolling bearings were used as shaft supports of gear pumps. The tendency to increase the speed of rotation of the shafts of modern high-speed machines leads to an increase in vibration overloads, vibration displacement, and an increase in the amplitudes of the oscillating shafts. Under new working conditions, sliding bearings are becoming more and more common. Hydrostatic bearings occupy an important place in the classification of sliding bearings.

Hydrostatic bearings have a number of advantages compared to other supports. One of the important advantages of hydrostatic bearings is the possibility of using kerosene as a lubricant for the working body of the machine. The presence of kerosene under high pressure is an additional factor that allows it to be used as the working body of a gear pump. The operation of the pump supports under liquid mode makes it possible to practically eliminate the wear of the working surfaces of the support elements. Owing to this, the reliability and resource of the pump, and therefore the engine as a whole, increases significantly.

The proposed special structures of the working surfaces of the bearing (three-chamber, two-chamber, and single-chamber) require extensive theoretical and experimental research. The lack of sufficient information on the design of such bearings makes research on this problem relevant.

2. Literature review and problem statement

In work [1], the sliding bearings of the gear aviation fuel pump are considered. The problem of the influence of structural parameters of the fuel pump bearing on its static characteristics is solved. Bearing capacity and operating temperature conditions were considered as static characteristics. These characteristics were determined based on the pressure distribution function in the working fluid layer. An option with two power chambers located on the working surface of the hydrostatic bearing is adopted. Three variants of the circumferential arrangement of the chambers relative to the line of action of the external load are studied. A quantitative assessment of the effect of increasing the temperature of the working fluid on lubricant consumption and bearing capacity is given. It was established that with an increase in the angle of the chambers relative to the line of action of the external load, the flow rate of the working fluid in the bearing increases, and its carrying capacity decreases. An increase in the temperature of the working fluid leads to a decrease in the carrying capacity of the bearing by 2.5 % and an increase in the consumption of the working fluid in the bearing – by 4.6 %. However, the paper did not pay attention to the study of the characteristics of the hydrostatic bearing of a gear pump with one bearing chamber on its working surface. In work [2], the influence of structural and operational parameters of the hydrostatic bearing of the fuel pump on its static characteristics is considered. The load carrying capacity and flow rate of the working fluid were considered as static characteristics. The characteristics were built on the basis of the pressure distribution function in the lubricating layer. It was determined from the joint solution of the Reynolds equations and the cost balance. The carrying capacity was determined by multiple integration of the pressure distribution function in the lubricating layer. The consumption of the lubricant was determined by the calculated pressures in the chambers. Variants of the working surface of the bearing with two bearing chambers were considered. Due to the fact that the load in the pump during its operation acts in one direction, the scheme of the working surface of the bearing with two chambers is adopted. The consumption of the working fluid in a two-chamber bearing was less compared to a three-chamber bearing, one of the factors significantly affecting the load carrying capacity of the bearing is the diameter of the nozzle installed at the entrance to the chamber. It was established that the dependence of the load carrying capacity of the hydrostatic bearing on the diameter of the jet is non-linear. When the nozzle diameter increased from 1 mm to 2.3 mm, the load carrying capacity of the bearing increased by 2.83 times. The amount of fuel required for the operation of the hydrostatic bearing was 1 % of the fuel pumped by the pump. However, the paper did not pay attention to the analysis of the characteristics of the bearing, which has one bearing chamber on the working surface. In [3], the problem of the influence of design parameters of a single-chamber hydrostatic bearing on its main characteristics was solved. When determining the main characteristics of a single-chamber hydrostatic bearing, the Reynolds equations and the cost balance were solved together. The obtained diagram of pressure distribution on the working surface of the bearing was used to determine the main characteristics. The influence of the gap, nozzle diameter, and chamber width on the load-bearing capacity and flow rate of the working fluid of a single-chamber hydrostatic bearing was investigated. It was found that with an increase in the gap, the carrying capacity of a single-chamber bearing decreases, and the flow rate of the working fluid increases. With an increase in the nozzle diameter, the bearing capacity of a single-chamber bearing increases. An increase in the width of the chambers leads to an increase in the bearing capacity and flow rate of the working fluid through the bearing. When the gap increased from 0.0125 mm to 0.0425 mm, the bearing capacity decreased by 1.156 times. The consumption of the 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 slightly increased, by approximately 1.02 times. Increasing the width of the chambers from 4 mm to 8 mm increased the carrying capacity by 1.29 times and increased the flow rate of the working fluid by 1.4 times. However, the work did not pay attention to the study of thermal modes of operation of a single-chamber bearing at different values of the eccentricity and diameter of the bearing. In [4], a transient calculation model was built to study the transient behavior of the lubrication of sliding bearings inside a gear pump of aviation fuel under a complex perturbation of an alternating load. The model took into account the preservation of the mass of the oil film cavitation boundary and the influence of the nonlinear dynamic environment of the external load. An analysis of the stability of the trajectory of the center of the axis and an analysis of the transient characteristics of the lubricant were carried out. The results showed that the error between the calculated results of the bearing model and the experimental data is less than 1.2 %. The influence of the dynamic load created by the gear fuel pump on the trajectory of the bearing axis is reflected in the increase in the displacement of the unstable stage of the trajectory and the disappearance of the position of static equilibrium. Comparative studies have shown that the equilibrium position is better for a bearing with a large width-to-diameter ratio or a smaller clearance ratio. When the clearance coefficient increases from 0.2 % to 0.6 %, the tendency of the axis position change at the unstable stage changes from two-peak to one-peak. However, the paper did not pay attention to the influence of thermal processes on the behavior of the bearing axis under the action of a variable external load. Study [5] gives a model for modeling vibration and dynamic loads in gear pumps with external gearing. The calculation was performed in the Matlab/Simulink program. Gear vibrations are excited due to variable pressure forces and variable stiffness of gear engagement. This model takes into account the stiffness and damping coefficient of sliding bearings, as well as the bending stiffness of gears. The influence of pressure and rotation speed on dynamic forces in the bearing is analyzed. However, the work did not pay attention to the consideration of the forces caused by the imbalance of the shafts on which the gears are installed. In [6], the optimization model for calculating the liquid friction sliding bearing of a gear pump is considered. The optimization is performed according to the diameter of the shaft, the ratio of the width of the bearing to its diameter, and the gap in the bearing. The minimum thickness of the oil film and the deflection of the shaft neck are used as the basis for the optimization model. After optimization calculations, the structure of the gear pump was improved. However, the work did not pay attention to the influence of thermal processes in the sliding bearing on the optimization of the gear pump design. In [7], a strategy for optimizing the design of bearings based on CFD modeling and data on lubricant characteristics is proposed. The working surface of the bearing is covered with an unstructured hexahedral mesh and the external load of the bearing is calculated by the CFD method. On the basis of modeling data, optimization of bearing design is performed using a special algorithm. As a result of the optimization, the value of the eccentricity is 0.82, and the ratio of the bearing width to its diameter is 1.19. At the same time, the load carrying capacity of the bearing was 5300 N, the increase in fuel temperature was 40 degrees Celsius. However, the authors did not pay attention to taking into account the change in fuel temperature when calculating the characteristics of the bearing and determining its structural dimensions.

In [8], a new mathematical model was built for determining the eccentricity of the position angle, the minimum oil film, as well as the misalignment angle relative to the *x* and *y* axes.

A test setup was designed for the experimental assessment of the influence of operating parameters on the misalignment angle. Experimental results show that the minimum thickness of the oil film can be reduced to 18 % due to misalignment. Consequently, for more accurate prediction of the internal engine-pump, the misalignment effect should be taken into account in calculations at an early stage of design. However, the work did not pay attention to the effect of misalignment on the main characteristics of the support. In work [9], the difficulties of designing gear oil pumps are considered. The adaptive design of the bearing sleeve is implemented in accordance with the coupling between the shaft and the support. Owing to the technology of adaptive design, the shortcomings of the three-dimensional solid structure are overcome, and the efficiency of the structure is increased. However, the work did not pay attention to the analysis of the main characteristics of the oil gear pump. In work [10], the lubrication characteristics of the coupled internal gearing of a gear pump under the conditions of operation at low speed and high pressure are considered. Significant wear and strong heating are observed between the toothed crown and the body. Lubricating characteristics of the moving pair of the gear ring of the pump housing can be improved by designing a reasonable bearing lubrication structure. In the cited study, a multi-purpose model was built, consisting of a dynamic model of the gear ring components and a model of liquid lubrication of the gear ring body. The results showed that mechanical losses are reduced by 31.52 %. However, the work did not pay attention to the use of hydrostatic effects in the lubricant, which make it possible to significantly reduce losses due to mechanical friction. In work [11], a numerous model of hybrid sliding bearings for use in fuel pumps was constructed. The numerical model of the bearing was implemented using the Newton-Raphson method. However, the model built does not pay attention to the thermal modes of operation of the fluid friction sliding bearing, which can significantly affect the main characteristics of the bearing and the operation of the pump as a whole. In [12], a hydrostatic bearing was designed for a two-arc screw-gear hydraulic pump. The paper reports a new idea of the analysis, which makes it possible to reduce the leakage due to the end gap caused by the axial force and to improve the volume efficiency of the gear hydraulic pump. However, the work did not pay attention to the influence of the change in the temperature of the lubricant, which can affect the size of the gap and the dimensions of the parts as a result of their expansion when the temperature increases. A compact pump reinforced with a ceramic structure was designed in [13]. To reduce the deformation of the housing under high pressure, a system was constructed using bearing balls that can precisely adjust the gap between the gears and the housing. Internal leakage resistance, which is complex, was used to estimate internal leakage. However, the work did not pay attention to the use of screw-groove and labyrinth seals, which allow the non-contact method to almost completely eliminate leaks. In [14], a model of the oil film in a hydrostatic bearing used in a gear pump operating at high pressures and high speeds using the method of computational hydromechanics was built. At a certain eccentric position of the shaft in the bearing, the distribution of pressure in the oil film, the flow rate of hydraulic oil and leakage at different speeds of rotation were calculated. The research results showed that at a certain position of the shaft in the bearing, a pressure drop is created in the groove of the assembly and a vortex is formed. The phenomenon of the vortex becomes more obvious with an increase in the speed of rotation. The output mass flow rate also decreases with increasing rotation speed. However, the work did not pay attention to dynamic phenomena inside the bearing caused by high rotation speeds, at which the shaft moves inside the bearing along certain trajectories.

Studies [1–14] lack information on investigating the influence of the temperature of the working fluid on the characteristics of a single-chamber hydrostatic bearing of a gear-type fuel pump.

3. The aim and objectives of the study

The aim of this work is to identify the effect of changing the temperature of the working fluid of a single-chamber hydrostatic bearing on its characteristics at different values of the design parameters. This makes it possible to establish the values of the design dimensions of the bearing that ensure its reliable operation.

To achieve this goal, the following tasks were set:

– to identify the effect of the temperature of the working fluid of a single-chamber hydrostatic bearing on its load-bearing capacity and flow rate of the working fluid at different values of eccentricity;

– to identify the effect of changing the temperature of the working fluid of a single-chamber hydrostatic bearing on its load-bearing capacity and flow rate of the working fluid at different values of the bearing diameter.

4. The study materials and methods

The object of the study is hydrostatic processes in sliding bearings of gear-type aviation fuel pumps. The hypothesis of the study assumed the possibility of assessing the characteristics of a single-chamber hydrostatic bearing based on changes in the temperature of the working fluid.

The mathematical model was constructed under the assumptions generally accepted in the hydrodynamic theory of lubrication. Calculations were performed for the case of a constantly acting external load (in a stationary setting). Calculations were performed in the Excel program (developed by Microsoft, USA).

The structure of a hydrostatic bearing with one bearing chamber located in the lower part of the bearing was considered (Fig. 1).

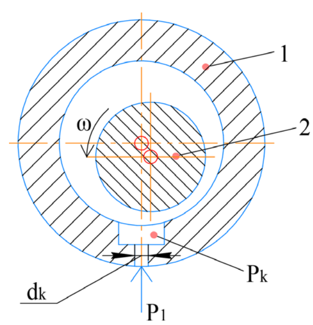

Fig. 1. Structural diagram of a single-chamber hydrostatic bearing: 1 — hydrostatic bearing; 2 — shaft

The diagram of a hydrostatic bearing shows one bearing chamber. The pressure in the chamber is designated P_K. Inside bearing 1, shaft 2 rotates with an angular velocity ω.

The working fluid is supplied to the chamber under high pressure P_1, through the inlet pressure compensator jet (diameter d_K). After passing through the bearing slot path, the working fluid enters the drain through the bearing ends.

The bearing load-carrying capacity and working fluid flow rate were determined based on the pressure distribution function in the lubricant layer. The pressure in the lubricant layer was determined from the joint solution of the Reynolds equations and the flow rate balance. Mathematical relationships for determining the bearing capacity and working fluid flow rate for a single-chamber hydrostatic bearing are given in [3].

When the working fluid moves in the bearing slot path, as a result of internal friction caused by viscosity, mechanical energy is converted into thermal energy. The heat generated as a result of friction heats the bearing working fluid. The physical properties of liquids are characterized by density, heat capacity, and viscosity. With those minor temperature differences that occur in plain bearings of liquid friction, only viscosity can change somewhat more than density and heat capacity. Consequently, viscosity has a dominant effect on the behavior of the working fluid in the bearing and on its integral characteristics as a whole. For an approximate estimate, the following figures can be given. Whereas with a temperature difference of 50 °C the viscosity changes by at least 3 times, the heat capacity changes by no more than 15 %, and the density by no more than 4 %.

Thermal calculation of a single-chamber hydrostatic bearing of an aviation fuel pump will allow a quantitative assessment of the increase in the temperature of the working fluid. A very important result of the thermal calculation is the assessment of change in the load-carrying capacity and flow rate of the working fluid as a result of an increase in the temperature of the working fluid.

The increase in the temperature of the working fluid was determined from the following expression:

$$\Delta T = \frac{N_p}{Q \cdot C_p \cdot \rho}, \tag{1}$$

where Q is the flow rate of the working fluid through the tract of a single-chamber bearing; ρ is the density of the working fluid; C_p is the heat capacity of the working fluid; N_p is the total power loss due to friction and pumping in the bearing.

The flow rate of the working fluid through the tract of a single-chamber hydrostatic bearing was determined by the known values of pressures in the chambers. The pressures in the chambers were determined from the equation of the balance of the flows through the input compensating device and along the contour of the supporting chamber.

Unlike hydrodynamic bearings, in hydrostatic bearings the working fluid is supplied under high pressure. Therefore, in these bearings, the power losses for pumping N_{pp} are of great importance. They are determined by the flow rate of the working fluid through the bearing tract Q and the pressure of the working fluid supply to the supporting chambers P_1:

$$N_{pp} = Q \cdot P_1. \tag{2}$$

Friction power losses are generally determined by double integration of the shear stress distribution function in the working fluid layer. The shear stress distribution function, in accordance with Newton's formula, is proportional to the viscosity of the working fluid and the lubricant velocity gradient across the layer thickness.

Considering that friction power losses in hydrostatic bearings are insignificant, the magnitude of these losses was determined using a simplified relationship:

$$N_{pt} = \frac{\mu \cdot \omega^2 \cdot R_p^3 \cdot L_p \cdot 2 \cdot \pi}{\delta_0}, \tag{3}$$

where μ is the dynamic viscosity of the working fluid; ω is the angular velocity of the shaft inside the bearing; R_p is the radius of the single-chamber hydrostatic bearing; $\delta_0 = R_p - R_b$ is the radial clearance in the bearing; R_b is the radius of the shaft inside the bearing.

Total power losses due to friction and pumping:

$$N_p = N_{pp} + N_{pt}. \tag{4}$$

The above mathematical model makes it possible to estimate the temperature increase in a single-chamber hydrostatic bearing and to establish the effect of changes in the temperature of the working fluid on the characteristics of the bearing.

5. Results of investigating the characteristics of a single-chamber bearing

5. 1. Results of investigating the characteristics of a single-chamber bearing at different eccentricity values

The study on the bearing capacity and flow rate of the working fluid was carried out based on the combined solution of the Reynolds equations, the balance of flows, and the function of change in clearance along the bearing circumference. The pressure in the chamber was determined from the equality of the flow rate of the working fluid at the inlet to the fluid flow rates along the chamber contour. The flow rates of the working fluid were recorded taking into account both gradient and transfer flows, as well as turbulent flow of the lubricant. Accounting for the turbulence of the lubricant flow was carried out using the imperial formulas proposed by Constantinescu.

The pressure in the lubricant layer on the remaining working surface of the bearing was determined from the numerical solution of the Reynolds equations. The numerical solution of the Reynolds equation was performed using the finite difference method. The working surface of the bearing was covered with a grid with a uniform pitch. Using a five-point template and the longitudinal-transverse sweep method, the fluid pressures at the grid nodes were determined. The boundary conditions for solving the boundary value problem were the pressure in the chamber and the pressure at the drain along the bearing ends. The calculation continued until the required accuracy was achieved.

The load-carrying capacity was determined by integrating the grid function of pressure distribution over the bearing surface. The flow rate of the working fluid was determined by the known pressure in the chamber.

The calculation of the load-carrying capacity and flow rate of the working fluid of a single-chamber hydrostatic bearing was performed with the following main values of its design dimensions:

1. Radial clearance $\delta_0 = 0.0225$ mm.
2. Shaft speed $\omega = 855$ s^{-1}.
3. Pressure of the working fluid supply at the inlet to the chamber $P_1 = 8$ MPa.
4. Nozzle diameter $d_1 = 2$ mm.
5. Number of chambers $K = 1$.

6. Kerosene working fluid TC-1 at a temperature of 100 °C.

7. Dynamic viscosity of kerosene at t=100 °C, μ= =0.4*10^{-9} N*s/mm^2.

8. The density of kerosene at t=100 °C, ρ=0.74*10^{-6} kg/mm^3.

9. Bearing diameter D_p=43.5 mm.

The results of calculating the pressure in the chamber for different eccentricity values in the range from 0 mm to 0.018 mm are shown in Fig. 2.

From Fig. 2 it is evident that with increasing eccentricity the pressure in the chamber increases significantly (approximately 6 times). The increase in pressure in the chamber is explained by a decrease in the gap and an increase in the resistance to the outflow of liquid from the chamber.

The dependence of the load-carrying capacity and flow rate of the working fluid on the eccentricity in a single-chamber hydrostatic bearing is shown in Fig. 3, 4.

It is evident from Fig. 3, 4 that the load-carrying capacity of a single-chamber hydrostatic bearing increases significantly with increasing eccentricity. With an increase in eccentricity from 0 mm to 0.018 mm, the bearing load-carrying capacity increases by 5.69 times. A characteristic feature of a single-chamber hydrostatic bearing is the presence of a load-carrying capacity even at zero eccentricity (in this case, 475.38 N). The flow rate of the working fluid through the bearing with an increase in eccentricity decreases significantly, by approximately 18.26 times. This is a significantly more significant reduction than in the structure of a hydrostatic bearing with uniformly distributed chambers around the circumference. This is a significant advantage of a single-chamber hydrostatic bearing. The law of change in the given characteristics is nonlinear.

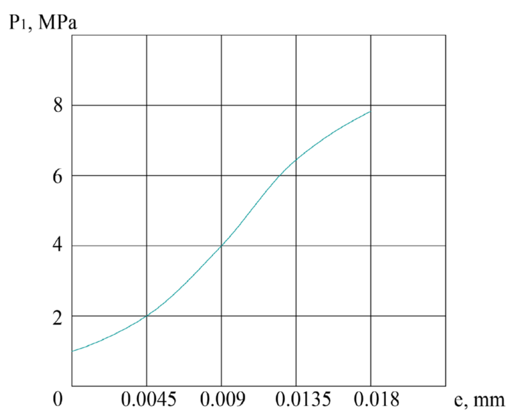

Fig. 2. Dependence of chamber pressure on eccentricity e in a single-chamber hydrostatic bearing

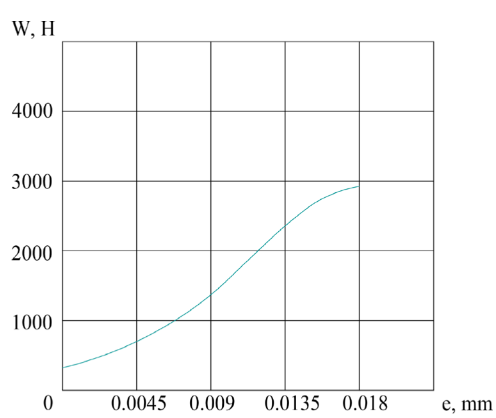

Fig. 3. Dependence of the load-carrying capacity on eccentricity e in a single-chamber hydrostatic bearing

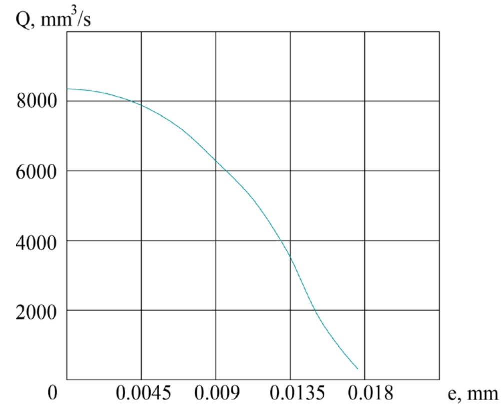

Fig. 4. Dependence of fuel consumption on eccentricity e in a single-chamber hydrostatic bearing

The dependence of change in the temperature of the working fluid (kerosene) on eccentricity in a single-chamber bearing is shown in Fig. 5.

Fig. 5 demonstrates that with increasing eccentricity, the temperature of the working fluid in a single-chamber hydrostatic bearing increases. At zero eccentricity, the temperature of the working fluid increased by 5.6 °C and was 105.6 °C. At an eccentricity of 0.018 mm, the temperature of the working fluid increased by 15.6 °C and was 115.6 °C. The increase in temperature with increasing eccentricity is associated with an increase in power losses due to friction and pumping of the working fluid.

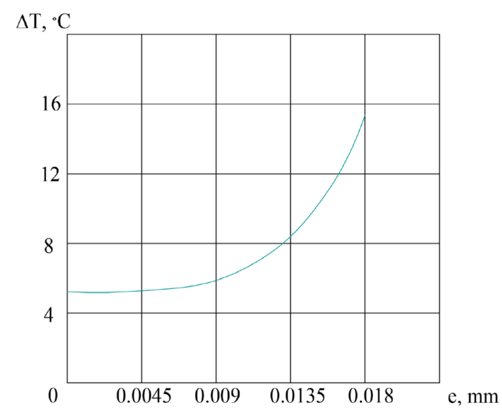

Fig. 5. Dependence of change in the temperature of the working fluid (kerosene) on eccentricity e in a single-chamber hydrostatic bearing

Calculation of the load-carrying capacity and flow rate of the working fluid taking into account the increase in temperature allowed me to establish the quantitative effect of the temperature increase on the characteristics of the bearing.

With an increase in eccentricity from 0 mm to 0.018 mm, the maximum increase in the flow rate of the working fluid was 19 %. The maximum decrease in the bearing load-carrying capacity due to an increase in the temperature of the working fluid was 17 %, and at a working eccentricity of 0.018 mm it did not exceed 1.83 %.

5. 2. Results of investigating the characteristics of a single-chamber hydrostatic bearing with different values of its diameter

The temperature regime of a single-chamber hydrostatic bearing with different values of its diameter was determined according to the total power losses due to friction and pumping. The load-carrying capacity and lubricant consumption were preliminarily determined with different values of the bearing diameter without taking into account the change in lubricant temperature. Then, the change in lubricant temperature was determined depending on the bearing diameter and the bearing characteristics taking into account the changed lubricant temperature. Fig. 6, 7 show the dependences of the load-carrying capacity and flow rate of the working fluid on bearing diameter without taking into account the change in temperature.

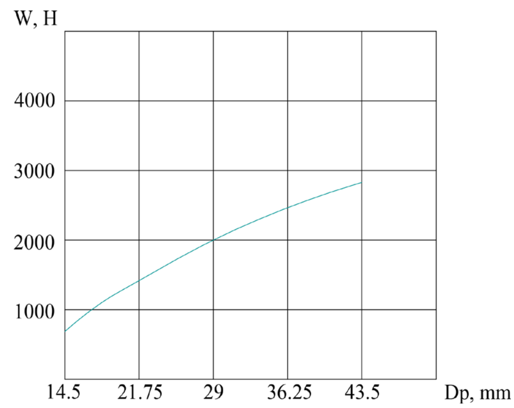

Fig. 6. Dependence of the load-carrying capacity on diameter D_p of a single-chamber hydrostatic bearing

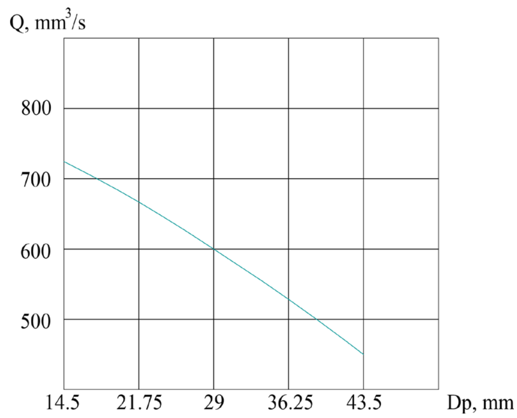

Fig. 7. Dependence of the flow rate of the working fluid on diameter D_p of a single-chamber hydrostatic bearing

From Fig. 6, 7 it is evident that the load-carrying capacity of a single-chamber hydrostatic bearing increases by 3.6 times with an increase in its diameter from 14.5 mm to 43.5 mm, and the lubricant consumption decreases by 1.58 times. The dependence of change in the temperature of the working fluid on the diameter of a single-chamber hydrostatic bearing is shown in Fig. 8.

It is evident from Fig. 8 that with an increase in the diameter of a single-chamber hydrostatic bearing, the temperature of the working fluid in it increases. With a bearing diameter of 14.5 mm, the temperature of the working fluid increases by 4.59 °C and was 104.59 °C. For a bearing diameter of 43.5 mm, the temperature of the working fluid increased by 15.6 °C and was 115.6 °C. Calculations taking into account the change in the temperature of the working fluid showed that the decrease in the load-carrying capacity due to an increase in temperature at different diameters did not exceed 0.5 %. The increase in the flow rate of the working fluid due to an increase in temperature was 6.4 %.

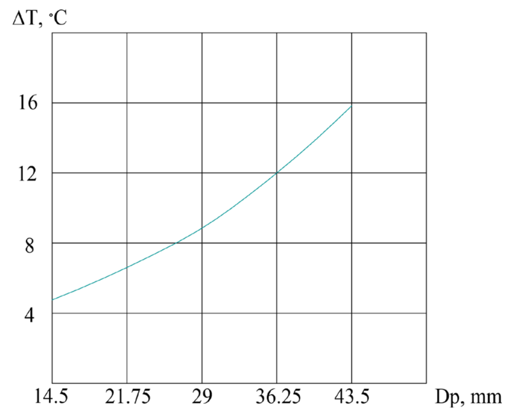

Fig. 8. Dependence of change in the temperature of the working fluid on the diameter of a single-chamber hydrostatic bearing

The results demonstrate that in a single-chamber hydrostatic bearing, a small change in the temperature of the working fluid is observed due to power losses due to friction and pumping. The main characteristics of the bearing also change insignificantly with an increase in the temperature of the working fluid.

6. Discussion of results based on investigating the characteristics of a hydrostatic bearing of a gear-type pump

The peculiarity of the proposed calculation procedure is to evaluate the thermal mode of operation of a single-chamber hydrostatic bearing of an aviation gear-type fuel pump. An algorithm for calculating the increase in the temperature of the working fluid has been proposed and the change in the temperature of the lubricant is taken into account when determining the characteristics of the bearing. The above analysis of the influence of design parameters of a single-chamber hydrostatic bearing on its characteristics, taking into account the change in the temperature of the working fluid, was not considered in [1–14]. The calculation results show that a single-chamber hydrostatic bearing maintains its operability under conditions of changing temperature of the working fluid.

The obtained results of calculating the load-carrying capacity and flow rate of the working fluid at its constant temperature and taking into account its increase due to power losses are shown in Fig. 2–8. It is evident from Fig. 2 that with an increase in eccentricity from 0 mm to 0.018 mm, the pressure in the chamber increases significantly (by about 6 times). The increase in pressure in the chamber is explained by a decrease in the gap and an increase in the resistance to the outflow of the working fluid from the chamber. It is evident from Fig. 3 and 4 that the load-carrying capacity of a single-chamber hydrostatic bearing increases significantly with an increase in eccentricity. With an increase in eccentricity from 0 mm to 0.018 mm, the bearing load-carrying capacity increases by 5.69 times. Unlike bearings with uniformly spaced chambers around the circumference, a single-chamber bearing has a load-carrying capacity at zero eccentricity. The flow rate of the working fluid through the bearing decreases significantly by about 18.26 times with an increase in eccentricity. This is explained by the fact that a single-chamber bearing does not have chambers located in its upper part, where there are large gaps and high flow rates of the working fluid.

It is evident from Fig. 5 that at zero eccentricity, the temperature of the working fluid increases by 5.6 °C and is 105.6 °C. At an eccentricity of 0.018 mm, the temperature of the working fluid increased by 15.6 °C and was 115.6 °C. The increase in temperature with increasing eccentricity is explained by the increase in power losses due to friction and pumping of the working fluid. Calculation of the bearing characteristics taking into account the increase in temperature showed that the bearing load-load-carrying capacity decreased by 17 %, and the flow rate of the working fluid increased by 19 % at the maximum value of eccentricity.

It is evident from Fig. 6, 7 that with an increase in the bearing diameter from 14.5 mm to 43.5 mm, the bearing load-carrying capacity increases by 3.6 times, and the flow rate of the working fluid decreases by 1.5 times. It is evident from Fig. 8 that with an increase in the bearing diameter, the temperature of the working fluid in it increases. With a bearing diameter of 14.5 mm, the temperature of the working fluid increased by 4.59 °C, and with a diameter of 43.5 mm, it increased by 15.6 °C. Calculations taking into account the change in the lubricant temperature showed that the decrease in the bearing load-carrying capacity with different diameters did not exceed 0.5 %. The increase in the flow rate of the working fluid due to an increase in temperature was 6.4 %.

A slight increase in temperature with an increase in the eccentricity and diameter of the bearing is explained by the high flow rates of the working fluid in hydrostatic bearings.

The results show that the single-chamber hydrostatic bearing can be used in gear-type fuel pumps.

The advantage of this study is the integrated approach associated with solving a complex hydromechanical and thermal problem. The proposed method makes it possible to perform practical calculations of single-chamber hydrostatic bearings. The considered bearing can be used in low-power aircraft gas turbine engines.

A limitation of this study is the likely clogging of the nozzles installed at the inlet to the chamber when the working fluid is not sufficiently purified. For bearings with other design parameters, the results and conclusions will not change qualitatively. The difference may only be in quantitative indicators.

A disadvantage of this study is the failure to take into account thermal deformations of the working surfaces of the bearing when calculating its characteristics.

The development of this study may involve studying dynamic phenomena in a single-chamber hydrostatic bearing.

7. Conclusions

1. It has been found that with increasing eccentricity, the temperature of the working fluid in a single-chamber hydrostatic bearing increases. With an increase in eccentricity from 0 mm to 0.018 mm, the increase in the flow rate of the working fluid in the bearing due to an increase in temperature was 19 %. The decrease in the bearing load-load-carrying capacity due to an increase in the temperature of the working fluid was 17 %.

2. It has been established that with an increase in the bearing diameter, the temperature of the working fluid in it increases. The decrease in the bearing load-carrying capacity due to an increase in temperature for different bearing diameters did not exceed 0.5 %. The increase in the flow rate of the working fluid due to an increase in temperature was 6.4 %.

Conflicts of interest

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

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Data availability

All data are available, either in numerical or graphical form, 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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