Studies of the efficiency of plain bearings used in microturbines

Исследования эффективности подшипников скольжения, применяемых в микротурбинах.

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Abstract: *The article assesses the possibility of using plain bearings (using the example of radial bearings) in high-speed microturbines (up to 60,000 rpm). The results of mathematical modeling of the thermal state of bearings and the oil layer between the shaft and the bearing surface are presented. A detailed description of the stand, designed to analyze the effectiveness of radial and thrust composite plain bearings, is given. The test bench allows testing of radial, thrust and rolling bearings, taking into account the deformation of the shaft from the weight of the turbine and compressor wheels of the microturbine. The results of an experimental study demonstrate high accuracy of calculations.*

KEYWORDS*: PLAIN BEARING, MICROTURBINE, TEST BENCH, FLUID BEARING, OIL WEDGE*

1. *Introduction*

High-speed (up to 60,000 rpm) microturbines are a perceptive type of engine, providing high specific power, low vibrations, low fuel requirements [1, 2, 3]. However, at the same time, the efficiency (efficiency) of such engines is often lower than that of piston engines. One of the possible ways to increase the efficiency is the use of plain bearings in the design of microturbines [4, 5]. The main advantages of such bearings are vibration resistance, low noise level during operation, compact radial dimensions [6].

2. *Mathematical modeling of fluid bearings*

Friction losses in sliding bearings depend on the sliding speed and the area of wetted surfaces [7]. With a constant bearing diameter, the only way to reduce the bearing area is to reduce its length. However, it is necessary to take into account the distortion of the shaft under the influence of the load. Therefore, when calculating the bearing, the clearance should provide not only the displacement of the shaft axis to create an oil wedge, but also the minimum thickness of the oil layer even with a strong skew of the shaft.

An important issue is the determination of the required radial clearance in the bearing. The minimum possible clearance between the rotor and the bearing stator (critical oil layer thickness) should not be less than the total height of the microroughness, multiplied by a safety factor of 2 [8].

One of the key parameters is the coefficient pv [9, 10], which determines the indestructibility of the oil film with a lack of lubrication. The calculated bearing must meet these parameters [11, 12].

In the course of work, a variant of a radial plain bearing was considered. Based on the calculations, the skew of the shaft reaches 0.045 mm.

The pressure distribution with a diametral clearance of 0.1 mm is shown in Fig. 2.1 and Fig. 2.2.

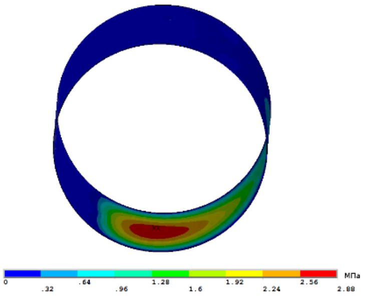

Fig.2.1 *- Bearing pressure without skew shaft*

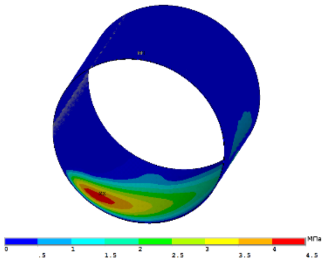

Fig.2.2 *- Bearing pressure with skew shaft*

The ratio of the bearing force created under such conditions to the required one as $К_{pv}$. If $К_{pv}$ is more than 100% - this means that the created lifting force is sufficient for bearingless bearing operation. The calculation results are shown in table 2.1.

Table 2.1. Bearing Calculation Results

Shaft position	δ, mm	e, mm	h_{min} mm	$N_{тр}$, kW	$К_{pv}$, %
Without skew	0,05	0.023	0,026	0,76	190
With skew	0,05	0.021	0,007	0,81	35

, δ - radial clearance, e - eccentricity, h_{min} - minimum thickness of the oil layer, $N_{тр}$ - friction loss power.

3. *Calculation of the thermal state of the bearings*

According to the results of mathematical modeling, the maximum power of the generated heat reaches 0.81 kW. In this calculation, we neglect the heat from the bearing to the environment and bearings, assuming that in the steady state all heat is removed by oil.

Oil consumption Q will be equal (1).

$$Q = \frac{A}{\rho \cdot c \cdot (t_{out} - t_{in})} \qquad (1)$$

,where A - amount of heat; ρ - oil density; c - specific heat of oil; t_{out} - oil temperature at the outlet of the bearing; t_{in} - oil temperature at the inlet to the bearing.

Substituting the characteristics of the oil, and given a temperature difference of not more than 20 degrees, we obtain Q=21,2 cm^3/s.

4. *Test bench for microturbine bearing prototypes.*

The test bench was designed and manufactured for bench testing of bearings designed for use in a microturbine. The stand is designed to test radial and thrust bearings with a shaft speed of up to 60,000 rpm.

The design of the stand provides the following types of tests of microturbine bearings:

-measurement of temperature indicators of the tested bearings;

-measurement of shaft rotation frequency;

-measurement of the moment of resistance of the tested bearings.

The developed design of the test bench allows you to simulate the loads that arise during the operation of a typical design of a microturbine with cantilevered wheels. The general view of the test bench is shown in Fig. 4.1.

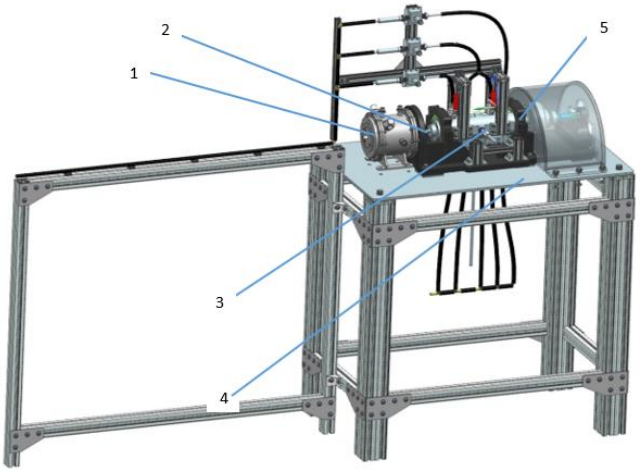

Fig. 4.1 *- General view of the stand for testing bearings*

The electric motor (1) is a drive device for the shaft (2) passing through the bearing housing (3), which is mounted on the mounting table (4) through the bearing assembly (5). The movable housing design allows measuring the moment of rolling resistance that occurs in composite bearings.

The electric motor allows you to accurately control the shaft speed, as well as measure the relative torque. The electric motor is equipped with a bellows coupling to compensate for misalignment of the bearing housing shaft relative to the motor shaft.

The shaft of the test bench (Fig. 4.2) is made stepped, which allows you to test different models and types of plain bearings on this bench without changing the shaft.

Fig. 4.2 *- Shaft of the test bench*

On the lower edge of the bearing housing (Figure 4.3) there are five threaded holes designed to drain the oil. Adapters are screwed into each of these holes. They are equipped with temperature sensors used to assess the temperature of the drained oil. The drain line allows you to combine all the exits from the bearing housing, and bring the drained oil back to the oil station. The hoses used in the drain line have low bending resistance, which allows us to neglect the forces arising from the deformation of the hoses to determine the torque in the tested bearings.

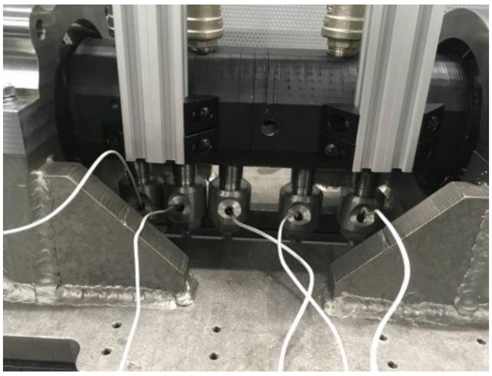

Fig. 4.3 *- The lower edge of the bearing housing*

A threaded hole is made in the side of the housing for installing the rod, designed to be moved to the load cell (Fig. 4.4). In addition, the bar provides space for the installation of balancing weights. On the other side of the housing, a threaded hole is also provided for installing the rod with balancing weights, which is necessary to minimize the imbalance of the entire assembly.

Fig 4.4 *– Load cell installation*

The supply line is mounted on a mounting frame, which allows you to transfer the center of rotation of the inlet fitting of the line to the axis of rotation of the housing and thereby minimize the measurement error. The supply line is equipped with three turbine type liquid flow meters, as well as taps that allow you to adjust the oil flow depending on the type of bearing being tested.

A protective cover closes the shaft exit from the bearing housing (Fig. 4.5). This casing is necessary as an element of protection in case of failure of the investigated bearings and possible uncontrolled movement of the shaft of the bench with a simulation mass having high kinetic energy.

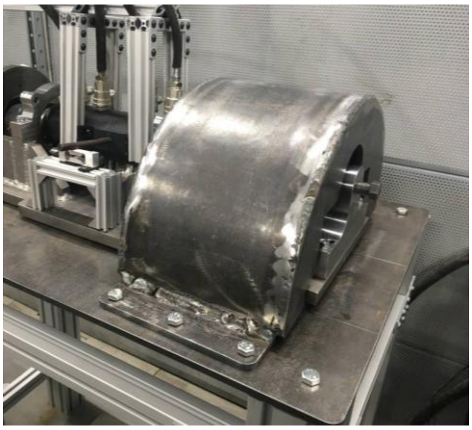

Fig. 4.5 *- Protective cover mounted on a stand*

An oil hydraulic station is used as a device for injecting and maintaining the temperature of the lubricant into the tested plain bearings. Technical characteristics of the stand are shown in table 4.1

***Table 4.1** - Technical characteristics of the test rig for testing bearings*

Range of measurement of frequency of rotation of the electric motor, min^{-1}	0…60000
Range of measurement of torque on the shaft of an electric machine, N·m	0…1
The accuracy of measuring the relative torque on the shaft of the electric machine, %	5
The accuracy of measuring the speed, %	5
Class precision load cell	C3
Weight, kg	243
Overall dimensions, mm	2334x617x1626

5. The results of the experiment.

In the framework of this article, one type of test is considered that corresponds to the conditions of the mathematical modeling. The conditions of the experiment:

Shaft rotation speed 60,000 rpm;
Initial oil temperature 24 ° C;
Oil consumption: 21.2 cm3 / s.

A graph of the temperature of the oil at the outlet of the test bearing relative to is shown in Figure 5.1.

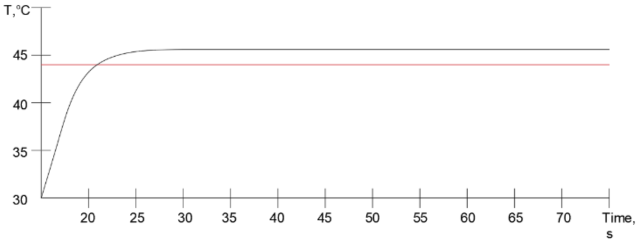

***Fig. 5.1** - Graph of oil temperature*

As you can see from the graph, the difference between the calculated and experimentally obtained values is less than 1.5ºC, which is less than 5%.

5. Conclusion

In this article, a mathematical model was developed for calculating the distribution of oil and temperature in a liquid radial bearing. The model provides high accuracy of calculations, and also allows you to take into account the speed of rotation of the shaft and its skew.

A bench was also designed for testing the sliding bearings of a microturbine with a shaft rotation speed of not more than 60,000 rpm. The design of the stand allows you to analyze information about the temperature state of plain bearings under various conditions of oil supply. In addition, the design of the stand is made in such a way that it is possible to measure the moment of resistance that occurs in the bearings with minimal errors.

6. Acknowledgments

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