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Study of the Performance of a Thrust Sliding Bearing in Startup and Rundown Regimes

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Abstract—This article describes the experimental bench simulating the operation of thrust bearings of the main circulation pump (MCP) of powerful power plants. The coolant moves in them at high speeds (5–10 m/s), at high pressure (up to 150×10^5 Pa) and temperatures up to 300°C. This entails the requirements for increased reliability of this unit. The bench was adapted for testing both when the bearing is lubricated with oil and when lubricated with water. The substitution of mineral oils with water became possible thanks to a comprehensive improvement of the bearing through the introduction of new antifriction materials and design solutions. The bench was also equipped with measuring systems for recording the moment of resistance to the rotation of the disk, the rotational speed of the disk, the angular velocity of the disk, and the temperature field near the working surfaces of the thrust bearings. The startup–rundown operating regime was programmed with a special software function. As a result of the studies, it was shown that simulating the main rundown stage on a test bench when a thrust bearing is operating in a water-filled volume cannot provide a comprehensive assessment of the performance of the main thrust bearing, since under operating conditions the lubrication process may be disrupted due to partial drainage of the bearing, leading to lubrication starvation, a sharp deterioration in heat dissipation, and damage. These phenomena must be carefully studied on real objects.

Keywords: thrust bearing, oil lubrication, water lubrication, S-1-U antifriction material, specific load, nonstationary “startup–rundown” regime

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INTRODUCTION

One of the essential components of the main circulation pump, which determine the reliability of the entire unit, is the thrust bearing components, which absorb large axial loads (up to 19.6×10^4 N and higher). Thrust hydrodynamic bearings are widely used, which must endure very difficult operating conditions [1, 2]. The main requirements for bearings are as follows:

(1) High operational reliability under all possible operating regimes of the pump, including numerous starts and stops under load.

(2) Guaranteed performance during continuous operation of at least 10000 h.

This should be supplemented by the need to ensure fire safety of the bearing, minimal friction losses, and improvement of the economic performance of the unit. Such research is carried out by world scientific centers, for example, in the Fraunhofer-Institut für Werkstoffmechanik (Freiburg, Germany) work is underway on a technique that allows us to switch to water lubrication of sliding bearings. German scientists have applied special additives that allow the use of water. An electrochemical reaction between the aluminum in the bearing and the steel in the shaft converts water into an ionic liquid lubricant containing cations and anions. The advantages of this development can be considered that the lubricant has a lower viscosity than conventional oils, while at the same time friction is reduced and metal corrosion is prevented [1, 2]. One of the disadvantages of this development is the complexity of the bearing design.

The Mechanical Engineering Research Institute, Russian Academy of Sciences, conducted research on lubricants for sliding bearings, during which it was established that the replacement of mineral oils with water became possible thanks to a comprehensive improvement of the bearing through the introduction of new antifriction materials and design solutions.

Objective—To study the performance of a thrust sliding bearing in startup and rundown regimes and to find ways to solve the issues of using water pumped by the main circulation pump to lubricate its bearings. Particularly difficult operating conditions for main pump bearings are created during transient processes that occur, in particular, during startup and rundown under load. Therefore, the study of bearings under these conditions is a very urgent task.

EXPERIMENTAL

The use of S-1-U material in sliding bearings plays a peculiar role in this work. This composite material with high antifriction and lubricating properties was developed at the Mechanical Engineering Research Institute, Russian Academy of Sciences [5, 6]. It has found application in sliding bearings when lubricated with both mineral oils and low-viscosity liquids, including water. The S-1-U material has a strong metal base, on which bronze powder, which has a special composition, is securely fixed. The pores between the powder granules are filled with fluoroplastic, in addition, a thin layer of fluoroplastic is applied on top of the granules.

Powder, fixed on the base, creates a strong framework that ensures the absence of warping, swelling, and deformation of the working surface of the bearing material in conditions of long-term operation under alternating local overheating and overcooling, with insufficient supply and temporary absence of lubricating fluid.

The fluoroplastic film covering the frame granules, being soft and easily deformable plastically, facilitates working of the coupling surfaces without disturbing its strength, as the strength of fluoroplastic increases significantly in thin films.

The coefficient of linear expansion of PTFE is much higher than that of bronze. Therefore, when locally overheated (in the hot zone), the fluoroplastic expands out of the pores, which increases the local thickness of the plastic film, improves lubrication conditions, and, helping the main lubricant, facilitates the extinguishing of the resulting temperature flashover.

If the additional supply of fluoroplastic to the superheating zone is still insufficient, powder pellets are automatically added to eliminate the resulting temperature rise. They contain a low-melting component that turns into a liquid phase when excessively overheated, intensively absorbs heat during melting, and provides additional lubrication.

For the reasons stated above, this bearing material does not adhere to hard steel even at temperatures of about 400°C.

Unsteady operating regimes of the unit, especially starting and stopping under load, create the most severe conditions for friction pairs. Under starting conditions, the start of motion occurs either through direct contact of the rubbing surfaces, or through a very thin layer of boundary lubrication. When switching to water lubrication, the study of unsteady regimes becomes especially important, since, unlike mineral oils, water does not form reliable lubricating boundary layers [6]. All the advantages of water are mainly realized at sufficient sliding speeds when the liquid phase of friction is ensured.

At the beginning of motion there are, apart from the forces of inertia, four types of resistance to motion, each of which is the sum of elementary forces generated in separate zones. The first type of resistance to motion is caused by the presence of atomic forces (adhesion or sticking) in the contact zones. The second type of resistance is the result of deformation (scratching) of the surface of the softer element of the friction pair by the protrusions of the corresponding harder element. The third type of resistance to the beginning of movement arises from deformations and cutting of the viscous medium filling the depressions of the coupling surfaces. Resistance of the fourth type is created when a pair of contaminants from the atmosphere or resulting from wear are pressed into the surface of a softer element. All four types of resistance to motion at the moment of coupling depend primarily on the materials of the friction pair, the design parameters of the bearing, and the characteristics of the lubricant filling the cavities of the coupling surfaces [7–9].

It is equally important to ensure the operability of the bearing during the shutdown of the unit. For the MCP thrust bearing, according to the operating requirements, it must have a certain duration. A long rundown puts the bearings in more difficult operating conditions, forcing it to work longer in the sliding speed regime, which does not provide hydrodynamic friction, and in the so-called "creeping speed" regime, at which friction coefficients and, consequently, heat generation significantly increase. Side harmful effects occur during friction associated with self-oscillating processes.

The described bench is designed for load testing of thrust bearings in conditions close to full-scale. The bench is mounted on a monolithic concrete foundation and has a design that allows the creation of a new testing facility of relatively small dimensions with an axial force of up to 100 tf. The bench is based on a closed welded structure (Fig. 1) the internal cavity of which (bath) is filled with lubricating fluid. Columns (*1*), mounted on frame (*2*), serve as guides for moving movable lower crossbeam (*3*). Upper crossbeam (*4*) is fixed on the top of the column. The loading device is hydraulic cylinder (*5*), the pressure in which moves the lower crosshead upward along the columns, ensuring the pressing of the lower segments of footplate (*9*) to working disk (*6*). The disk, driven by a spline gear, slides under load along splines (*7*) and transmits forces to upper row of segments (*8*) of the thrust bearing. When fixed on the columns, crossbeam (*4*) hermetically seals the internal working cavity of the stand with lubricating fluid and test segments. Strong welded

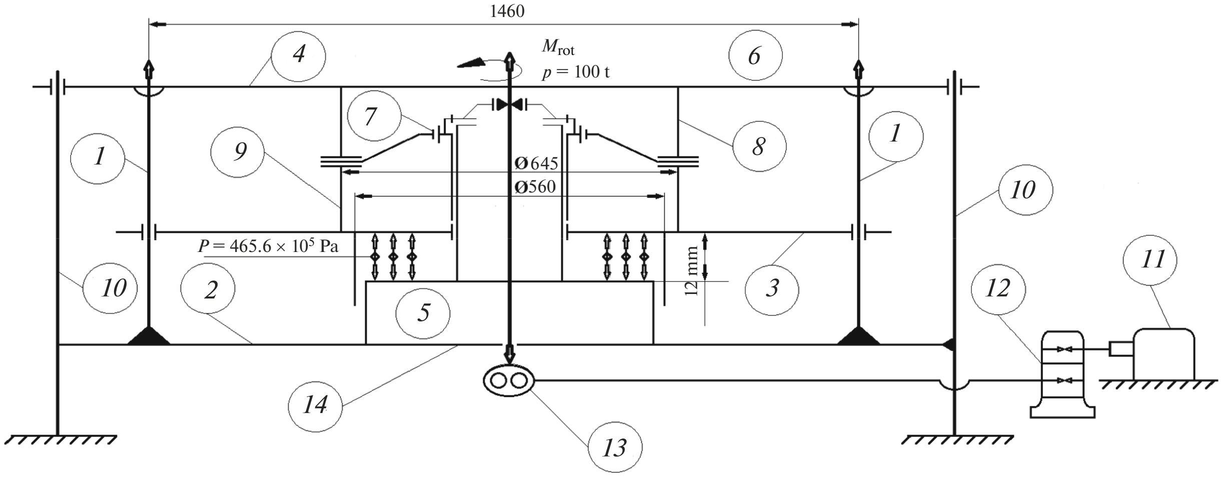

Fig. 1. Kinematic diagram of the bench.

frame (*10*) provides rigidity of the structure and absorbs forces from friction and weight of the stand elements.

The rotational motion of the disk is created by DK202B DC motor (*11*) (power 86 kW, rotation speed 1500 rpm), connected to the kinematic chain through gearbox (*12*) and bevel reduction gearbox (*13*) (i = 8.214). Vertical torsion bar (*14*) transmits rotation to the gear coupling engaged with the working disk.

The scheme of loading the thrust bearing through the lower segment implemented in the bench has the following advantages:

(1) There is no need to use a special load bearing of a different design, which would introduce additional errors when measuring friction parameters.

(2) The smallest possible dimensions of the machine.

The number of test segments in each row can vary from 3 to 16. In these tests, 3 segments were installed in the upper and lower rows. This made it possible to achieve the maximum remote load with the minimum pressure in the load cylinder. The specific load on the thrust bearing segment (P_{spec}) at the pressure in the hydraulic cylinder cavity of 18×10^5 Pa is 1200×10^5 Pa. The force compensating for the weight of the disc is generated at a pressure of 1×10^5 Pa in the hydraulic cylinder cavity. The sliding speed in friction pairs on a diameter of 645 mm reaches a value of 3.8 m/s at a disc speed of 100 rpm. For the calculation, the diameter of the middle circumference of the annular friction plate on the disc was taken. The optimum rotation speed of the disc from the point of view of vibration resistance of the structure and ensuring the fastest running-in of friction pairs is 100–110 rpm. For comparative tests of thrust bearings when lubricated with oil and water in the "startup and rundown" regimes, a significant modernization of the bench, drive, and measuring systems was required.

Water is an aggressive medium, so to prevent corrosion of the internal cavity of the cars, after thorough degreasing, they were coated with red lead.

The preparation of the working disk of the machine, made of stainless steel, included double-sided grinding of the friction planes and subsequent manual finishing with diamond paste. The non-parallelism of the planes, determined by measuring the thickness difference, was 0.04 mm. Manual grinding ensured the absence of marks from the grinding wheel and a high roughness class. In the initial version (when working on oil), the hydraulic system of the bench is a single whole, since the loading hydraulic cylinder and the machine bath itself are filled with oil circulating along a common line. The main elements of lines are presented in Fig. 2. To work on water, it was necessary to separate the lines supplying the machine bath and the loading device. The drain and discharge pipes connecting bath (*1*) with cooling system (*2*) were plugged, and oil was supplied to the hydraulic cylinder from additional tank (*3*). Water was pumped through a special pipeline after settling in separate container (*4*). The independent loading system is designed in such a way that it can be used for testing in both water and oil.

In order to eliminate the possibility of mixing water and oil, the movable connection of the lower cross-beam with the central cylindrical part was additionally sealed with a stuffing box.

Changing the disk rotation speed was initially achieved by using a 200-kW machine generator to power the DC electric motor with output voltage regulation. The disadvantages of the machine generator were as follows:

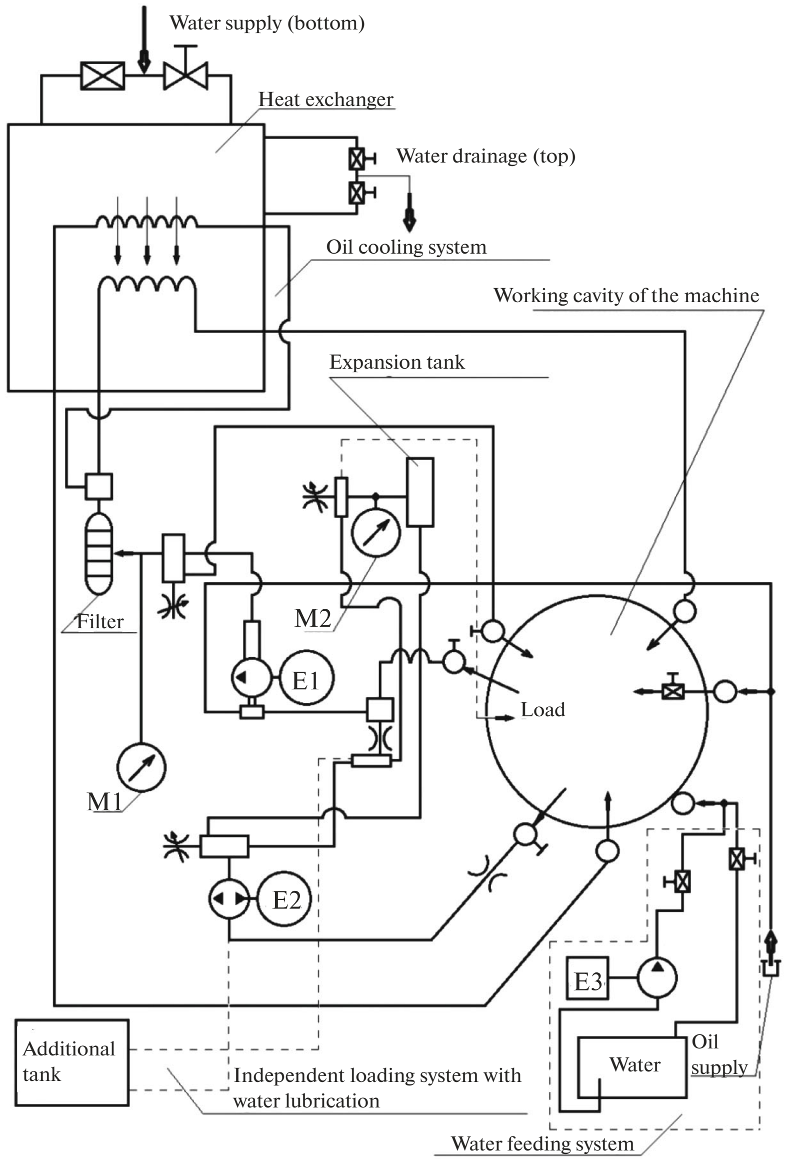

Fig. 2. Main elements of bench routes.

(1) Inability to smoothly accelerate the electric motor in the low frequency range (starting from zero).

(2) Rough adjustment of rotation speed in the upper frequency range.

(3) Difficulty in automating the management of transient processes.

Carrying out this work required replacing the machine generator with a thyristor electric drive of the KET-500/460-2-1-UKhL4 type. This significantly expanded the capabilities of the bench and made it possible to implement a test program related to the study of transient processes. KET-500 is a static nonreversible adjustable converter of three-phase alternating current of industrial frequency into direct current. It is designed to power the armature circuits of DC motors.

The KET-500 thyristor converter produces direct current, the voltage of which is regulated by resistor *R* in the control circuit with negative feedback (Fig. 3). To expand the capabilities of the drive and automate the control, a specially equipped computer was connected to the break in the feedback circuit. This made it possible to organize the transient process during testing according to any predetermined schedule. The software function generated by the computer can have almost any required form. Using the toggle switch, the control mode is transferred from manual (position 1) to automatic from the computer (position 2).

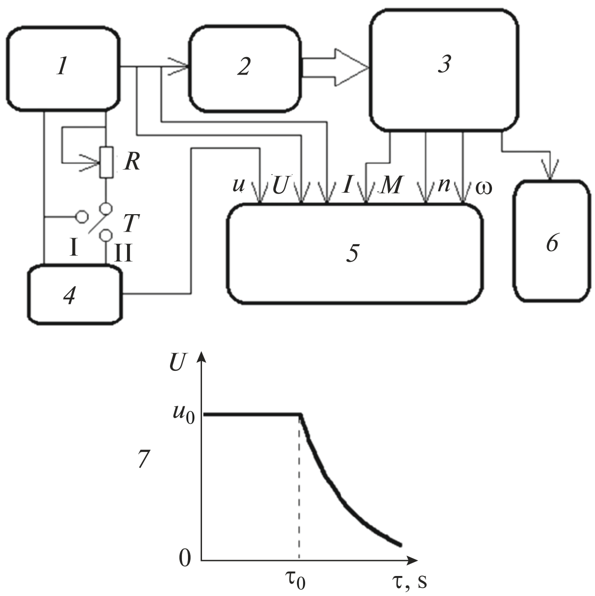

Fig. 3. Block diagram of software control of transient regimes of the bench and data recording of measuring systems: (*1*) electric drive thyristor CAT-500/460-2-1-UHL4; (*2*) DC electric motor; (*3*) experimental bench; (*4*) electronic computer; (*5*) N004 light-beam oscilloscope; (*6*) EPP-009 self-recording potentiometer; (*7*) software function implemented by a computer.

In this case, it was necessary to quickly accelerate the bench to the nominal part of rotation, and then switch to a smooth rundown for 3 min. Here, the rundown was performed by means of an exponential. Such a test cycle gives the closest approximation to the operating conditions of a real bearing of the main circulation pump.

The combination of startup and rundown in one test cycle was dictated not only by the desire to reduce the experiment time, but also by the capabilities of the system for measuring the moment of resistance to rotation, which does not have a special movable current collector (the experiment time is limited by the length of the twisted electrical wires between the strain gauges and the strain amplifier).

The form of the software function implemented on a computer is presented in Fig. 3, (*7*). It consists of a direct section (the voltage in the feedback circuit is set by resistor *R*), on which the exponential section accelerates (exponential function $U = U_0 e^{-\frac{\tau}{3T_0}}$ is preset by computer), where rundown takes place. The rundown time is $3T_0 \approx 3$ min. It is determined by the exponential constant specified by the computer.

The test bench was equipped with the following measuring systems:

(1) System for controlling the moment of resistance to rotation of the disk (*M*).

(2) Disc speed measurement system (*N*).

(3) Disk angular velocity measurement system (ω).

(4) Temperature control system.

M was measured using two strain gauges ($l = 30$ mm, $R = 124\ \Omega$), glued 200 mm from the upper support at an angle of 45° and 135° to the horizontal. The strain gauges were connected to the 8LNCh-7M strain amplifier (according to an active half-bridge circuit) through a plug connector located at the upper end of the torsion bar. The torsion bar deformation was measured in the second sensitivity subrange of the amplifier. An obligatory step was the calibration of the strain amplifier in combination with a recording device (N 004 light beam oscilloscope) for torque (*M*). Calibration was carried out using a manual hoist, which was connected through a diameter and a system of blocks with steel cables to a clutch disk with a diameter of 172 mm between the drive electric motor and the gearbox. The average calibration data taking into account the gear ratio of the gearbox ($i = 6.17$) and bevel gear ($i = 8.214$) was: 535.5 × 10 N m per 1 cm of deflection of the oscilloscope galvanometer beam.

The shaft rotation frequency was recorded using an induction sensor. The sensor was fixed on the upper crossbeam and generated an impulse when a steel flag mounted on a rotating shaft passed by it. The pulse was recorded by a light-beam oscilloscope and a Ch3-28 electron counting frequency meter.

In order to more accurately record the beginning and end of the disk motion and measure the rotation speed at low (“creeping”) speeds when starting and stopping, a measuring system is required that is sensitive to small (intra-rotation) angular movements of the disk. This function was performed by a photosensor that responded to signals from the lighting lamp arriving through a slotted separator mounted on the shaft every 2° of rotation of the disk. The sensor based on the FK-1 photoresistor was powered by a rectified stabilized current voltage of 1.5 V. The supply voltage of the illuminator (20 W incandescent lamps) is 9 V. Photosensor signals (ω) were supplied to an M 001.3 galvanometer (sensitivity 35 mm/mA) of a light-beam oscilloscope (see Fig. 3).

Temperature measurements were carried out with chromel–copel thermocouples in a silk winding and a shielding metal braid. To prevent short-circuiting when working in an aqueous environment, the thermocouples were additionally insulated with BF-4 adhesive and vinyl chloride casings. Data recording was carried out automatically by EPP 09-M3 electronic potentiometers in the range of 0–200°C.

The test bench control panel is equipped with current monitoring devices (M362 ammeter for 200 A) and voltage (voltmeter M340 for 450 V) in the power supply circuit of the drive motor.

The control panel is also equipped with pressure gauges for monitoring pressure at various points of the hydraulic line, including a reference pressure gauge for recording the pressure in the hydraulic loading device.

In connection with the installation of the KET-500 thyristor converter to power the drive motor for automation of experiments using a control computer, information about the operation of the KET-500 in the cycle of automatic control of transient processes was of considerable interest. Therefore, the galvanometers of the oscilloscope were fed signals proportional to the current and voltage of the supply circuit. A separate loop recorded the control signal generated by the computer.

Thus, all information about the operation of the computer, the KET-500 drive, and the bench, except for the temperature data, was recorded on the tape of the light-beam oscilloscope through the following channels (Fig. 3):

(1) *u*—software function of the computer.

(2) *U*—the program software performs the voltage function of the KTE-500 drive.

(3) *I*—change in current in the drive circuit.

(4) *N*—disk rotation speed (the bench performs the software function).

(5) ω—recording of the moment of breaking, stopping and intra-turn movements of the disc.

(6) *M*—moment of resistance (characteristic function of the friction process).

RESULTS AND DISCUSSION

The created bench is designed to conduct research into the performance of a thrust sliding bearing in startup and rundown regimes, that is, to analyze comparative tests of the S-1-U material under unsteady conditions when lubricated with oil and water.

Tests confirmed the high running-in qualities of the S-1-U coating. It is shown that a significant factor influencing the time and quality of running-in is the initial non-flatness of the friction surfaces, which, within acceptable limits, can occur on segments with the S-1-U coating. During the running-in tests, a method for correcting the installation of segments based on the results of measuring the temperature of control points of the near-surface layers of the segments was proposed and tested for a thrust bearing without automatic equalization of loads on the segments.

The process of development of strong frictional self-oscillations in the bench, caused by the specific characteristics of friction when the thrust bearing is lubricated with water, is determined by the backlash in the kinematic drive chain and the resonant parameters of the elastic torsion bar–disc system. At the same time, the features of friction during water lubrication must be considered at the design stages, taking into account their dynamic characteristics.

Taking into account the creation of stressful conditions during startup and rundown combined into one cycle, it was concluded that thrust bearings with the S-1-U material can be successfully used for main circulation pumps under startup and shutdown conditions under the load provided in the original test program (300×10^5 Pa) when lubricated with water. At the same time, no damage to friction surfaces on water was observed under both transient and stationary conditions.

These studies will be presented in more detail in the next article devoted to testing a thrust sliding bearing.

CONCLUSIONS

(1) The practical significance of this bench lies in its use in conducting comparative tests of the S-1-U material under unsteady conditions (startup and rundown under load) when lubricated with oil and water, during which it was modernized and re-equipped for testing thrust bearings: separation of hydraulic lines for loading and lubrication is ensured.

(2) The electric machine drive was substituted with a thyristor drive, software control of transient processes from an electronic computer was introduced, and measuring systems were improved.

(3) The bench has been prepared and re-equipped to study sliding bearings in startup and rundown regimes.

NOTATION

MCP	main circulation pump
ECM	electronic computing machine

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CONFLICT OF INTEREST

The authors of this work declare that they have no conflicts of interest.

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