# Estimation of dynamic behavior and energy efficiency of thrust hybrid bearings with active control

Alexander Babin, Sergey Majorov, Leonid Savin

***Abstract*—** The present paper considers analysis of world tendencies of application of bearings with feedback control systems in various types of rotor machines. A possibility of qualitative improvement of operational characteristics has been highlighted for bearings with integrated additional functions of automated diagnostics and active control of geometric, rheological and force parameters. The present paper features a mathematical model, its implementation in a form of an algorithm, and numerical results including pressure and temperature distribution in a bearing in question, load capacity and rotor's axial trajectories. A quality index has been formulated for thrust tapered land hybrid bearings with a central feeding chamber that is based on a combination of minimum power loss due to friction and stability of motion of a rotor. Various combinations of roughness and axial gaps have been taken into account.

***Keywords*—**mechatronics, active control, fluid-film bearings, thrust bearings, dynamic behavior, energy efficiency.

## I. Introduction

INTEREST in modelling and experimental investigation of active bearings arose with the evolution of what is commonly referred to as Industry 4.0 [1]. Synergy-based technological overlap of mechanical engineering, electrical engineering and information technologies enable smart machines, and many cases emerge where traditional parts of machines could be replaced with mechatronic devices that extend functionality and enhance performance of these machines. In the field of rotating machinery, fluid-film bearings used to carry the load of a shaft are the most vital elements and thus become subjected to various enhancements, including the possibility to introduce active control systems. The idea behind these enhancements is to not only provide a stable operational regime, but also and very importantly to increase energy efficiency of a machine. And power loss in rotating machinery mainly occurs in bearings due to both friction and vibration. Machines subjected to loads varying in time or vibrations due to operational regimes require certain sets of dynamic characteristics of bearings, these sets are unique in each case, making design of bearings a challenge.

Investigation in the field of rotor's dynamic behaviour control in fluid-film bearings is carried out in the following fields: study of the possibility of profile adjustment of a gap between a rotor and a sleeve or a thrust disk of a bearing, e.g. tilting-pad and some foil bearings [2, 3]; study of the possibility of application of lubricants with adjustable parameters such as viscosity or density, e.g. ferromagnetic fluids [4, 5]; and study of hybrid fluid-film bearings [6, 7], where load capacity is generated as a combination of hydrodynamic forces in the film and hydrostatic force from the lubricant supplied into a bearing under pressure, where rotor's position can be adjusted by means of electrohydraulic devises like servovalves.

Application of servovalves proved feasible in many applications with journal bearings [8-10]. There are however very few studies regarding the possibility to apply the same approach to thrust bearings. The purpose of the present paper is to introduce active control system to a traditional thrust tapered land bearing and to estimate feasibility to use active control to enhance the performance of the bearing in question.

## II. Mathematical model of an Active thrust hybrid bearing

The operational principle of an ATHB (Figure 1) is as follows: the pump (1), located in the tank with the fluid (water, oil, etc.), supplies the lubricant into the housing (2) of the bearing (3), which supports the rotor (4) in the axial direction. The pressure data is acquired with the sensor (8). The data on the present position of the rotor is acquired with the proximity sensor (7). Data from the sensors is analogue and is acquired with the analogue input modules of a DAQ system (5). This system also controls the servovalve (6) by means of adjusting the opening rate $\xi$ generating an analogue voltage signal, and the pump (discrete output module).

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Alexander Babin is with the research lab ‘Modelling of Hydro and Mechanical Systems’ at the Orel State University, Orel, Naugorskoe sh., 29, 302020, Russia (tel.: +79290606012, e-mail: alex.mech.osu@gmail.com).

Sergey Majorov is with the research lab ‘Modelling of Hydro and Mechanical Systems’ at the Orel State University, Orel, Naugorskoe sh., 29, 302020, Russia (e-mail: sergeyostu@yandex.ru).

Leonid Savin is with the department of Mechatronics, Mechanics and Robotics at the Orel State University, Orel, Naugorskoe sh., 29, 302020, Russia (e-mail: savin@ostu.ru).

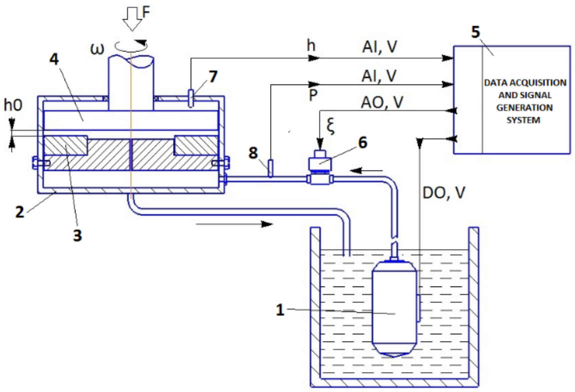



Fig. 1 principle diagram of an ATHB

To simulate the operation of an ATHB one has to pay close attention to two basic problems: a problem of hydrodynamic lubrication theory based on the Reynolds equation and the energy balance equation, and a problem of theory of automatic control. Merging a model of a hybrid bearing with a model of the control system yields a model of an active bearing. Here, the controlled variable is the axial gap between the rotor's thrust disk and the bearing in question. By means of controlled supply of lubricant under pressure, one could change the pressure distribution in the bearing thus controlling the value of the gap. The first stage is consequently modelling a passive thrust hybrid bearing (THB). The photo and calculation diagram of a THB are presented in the Fig. 2.

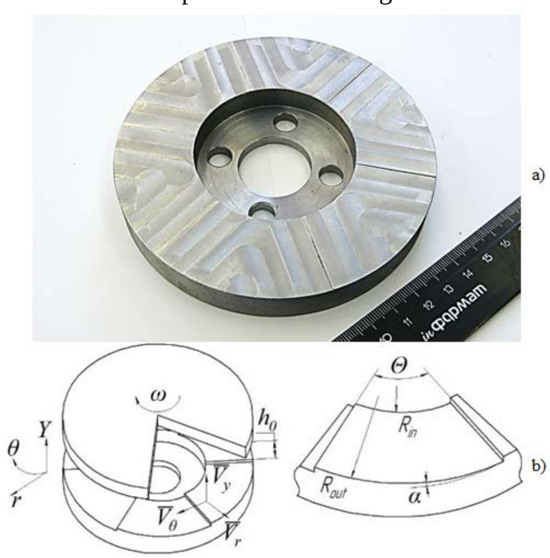



Fig. 2 photo of a THB (a), calculation diagram of a THB (b)

Here $R_{in}$ and $_{\mathrm{Rout}}$ are inner and outer radiuses of the bearing accordingly, $V_r$, $V_\theta$ and $V_y$ are velocity components in the polar coordinate system, $h_0$ – minimum film thickness, $\Theta$ – angular extent of a single pad, $\alpha$ – angle of inclination of a pad.

To obtain the pressure distribution, the generalized Reynolds equation is derived to be solved iteratively with the energy balance equation, which allows taking into account possible viscosity variation in the fluid film due to temperature rise that occurs as a result of action of friction between the layers of the fluid. The generalized Reynolds equation takes the following form:

$$\frac{\partial}{\partial\theta}\left[\frac{1}{r}F_2\frac{\partial p}{\partial\theta}\right]+\frac{\partial}{\partial r}\left[rF_2\frac{\partial p}{\partial r}\right]=\omega r\frac{\partial}{\partial\theta}\left(\frac{F_1}{F_0}\right)+r\frac{\partial h}{\partial t}, \quad (1)$$

where $\omega$ – angular velocity of the shaft, $p(r,\theta)$ – local pressure, $F_0$, $F_1$, $F_2$ – Dowson's functions determined as in [11]:

$$F_0=\int_0^h\frac{dy}{\mu},\; F_1=\int_0^h\frac{ydy}{\mu},\; F_2=\int_0^h\frac{y}{\mu}\left(y-\frac{F_1}{F_0}\right)dy,$$

where $h(\theta)$ – axial gap function, $\mu(r,\theta)$ – local viscosity, which is determined as a distributed scalar value and is a function of temperature:

$$\mu=\mu_0e^{-\lambda(T-T_s)},$$

where $\mu_0$ – viscosity of the supplied fluid, $T_s$ – temperature of the supplied fluid, $\lambda$ – temperature/viscosity coefficient.

Here it has to be noted that the shape of the axial gap is modelled using Bezier curves that allow higher calculation accuracy of pressure gradient in the circumferential direction. In the Fig. 3 results of modelling and actual gap measurement are presented along with the photo of the measuring setup.

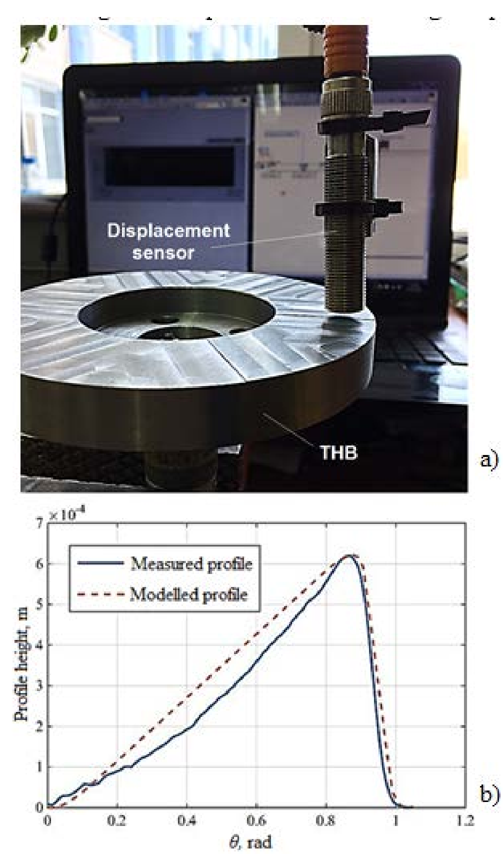



Fig. 3 photo of the profile measurement system (a), comparison between profile measurement and modelling (b)

It could be seen, that modelling results are accurate enough, however, there is a slight deviation due to mechanical treatment of the surface of the THB.

In order to obtain the temperature distribution the energy equation taking into account the action of dissipation forces in the fluid is derived in the following form:

$$\rho c_p\left(-\frac{h^2}{12\mu}\frac{\partial p}{\partial r}\frac{\partial T}{\partial r}+\left(-\frac{h^2}{12\mu r}\frac{\partial p}{\partial \theta}+\frac{\omega r}{2}\right)\frac{\partial T}{r\partial \theta}\right)= \\ =\mu\left(\left(\frac{1}{12}\left(\frac{h}{\mu}\frac{\partial p}{\partial r}\right)^2\right)+\left(\frac{1}{12}\left(\frac{h}{\mu}\frac{\partial p}{r\partial \theta}\right)^2+\left(\frac{\omega r}{h}\right)^2\right)\right), \quad (2)$$

where $\rho$ – density of the fluid, $c_p$ – heat capacity of the fluid, $T(r,\theta)$ – local temperature.

Equations (1) and (2) and the additional expressions are solved iteratively using a finite difference method with the following boundary conditions for pressure and temperature:

$$p\big|_{r=Rout}=p_a,\ p\big|_{r=Rin}=p_s,$$

$$p\big|_{\theta=0}=p\big|_{\theta=\Theta},\ \frac{\partial p}{\partial \theta}\bigg|_{\theta=0}=\frac{\partial p}{\partial \theta}\bigg|_{\theta=\Theta},$$

$$T\big|_{r=Rin}=T_s, T\big|_{\theta=0}=T\big|_{\theta=\Theta}, \frac{\partial T}{\partial \theta}\bigg|_{\theta=0}=\frac{\partial T}{\partial \theta}\bigg|_{\theta=\Theta},$$

where $p_s$ – supply pressure, $p_a$ – ambient pressure.

The iteration process is then considered converged if the convergence criterion based on difference in load capacity of a bearing on the present and the previous iteration. Character pressure and temperature distributions in thrust hybrid bearings are presented in the Fig. 4.

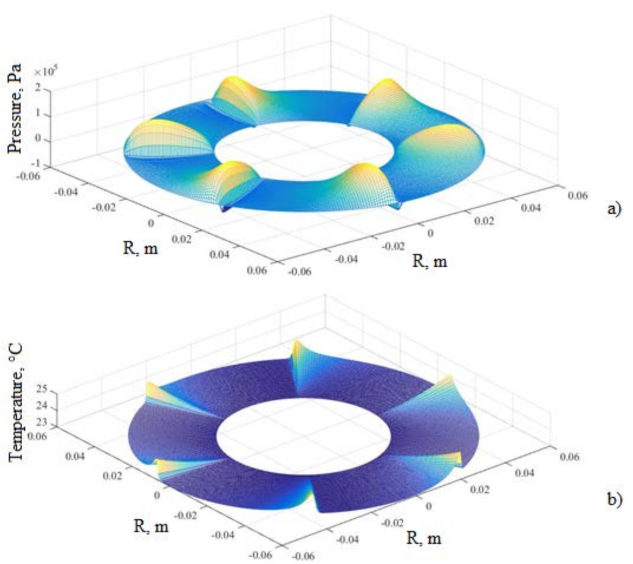



Fig. 4 pressure (a) and temperature (b) distribution in a THB

Load capacity of a bearing is obtained by means of integration of pressure distribution over the area of the bearing:

$$R_Y=z_p\int_0^{\Theta}\int_{Rin}^{Rout}p(r,\theta)rd\theta dr,$$

where $z_p$ – number of fixed pads.

To estimate dynamic behavior of a rotor on the ATHB a model of a dynamic system has to be developed. The diagram is presented in the Fig. 5

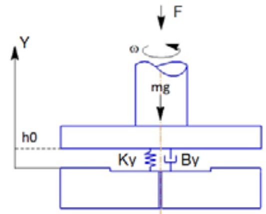



Fig. 5 rotor-bearing dynamic system

Under steady conditions the rotor takes the equilibrium position, when the hydrodynamic forces counterbalance the inertia forces, i.e. $R_{Y0}=mg$, where $m$ – rotor's mass, $g$ – freefall acceleration. When the rotor's position changes under the influence of some impulse, some additional forces start acting on the rotor, so, assuming these displacements are relatively small, the reaction force of a bearing could be linearized using Taylor series. The reaction force of the bearing could be determined as follows:

$$R_y=R_{Y0}-K_Y\Delta Y-B_Y\Delta\dot{Y},$$

and, consequently, the motion law of the system is as follows:

$$m\ddot{Y}-B_Y\dot{Y}-K_YY=F(t), \quad (3)$$

where $F(t)$ is some external force as a function of time $t$.

(3) is solved using the Runge-Kutta method of the 4$^{th}$ order.

In order to introduce the model of the control system to the developed model of the bearing, the structure of the named system has to be considered (Fig. 6).

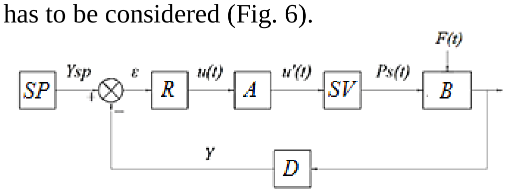



Fig. 6 Structure of the control system of an ATHB

The control system consists of a module that sets the value of a set point (SP), and the data channel (D), the obtained data is compared to the set point and the error ε is the used to generate a control signal by means of application of a PID controller based regulator (R). The signal is then amplified (A) and sent to the actuator – servovalve (SV) – thus changing the supply pressure. These actions affect the position of a rotor in the bearing (B) that could be subjected to some external force F(t).

The aim of control is to maintain a desired value of the axial gap between the rotor and the THB. The control principle is essentially regulation of in-flow pressure to increase or decrease the bearing's reaction force which results in increase or decrease of the axial gap. Here, the regulator is the core element of the control system and is based on the PID control law, where the output signal is a function of the current error and the desired position *Ysp* (set point):

$$u(t)=K_p\cdot\varepsilon(t)+K_i\cdot\int_0^t\varepsilon(t)dt+K_d\cdot\frac{\partial\varepsilon(t)}{\partial t},$$

where $u(t)$ – output signal from the regulator, $Kp$, $Ki$, $Kd$ – proportional, integral and differential components of the regulator accordingly, $\varepsilon(t)$ – error function.

The kind of electrohydraulic actuators used in the present research is simulated using the known relation between the pressure it supplies the lubricant with and the voltage on the input of the valve. The most important term here is the time constant of the valve $T_{SV}$, i.e. the time it takes for a fully closed valve to fully open. This is a crucial characteristic for systems subjected to high-frequency periodic loads.

The expression that allows simulation of a servo valve is as follows:

$$T_{SV}\frac{dp(t)}{dt}+p(t)=K_{SV}u(t), \tag{4}$$

where $K_{SV}$ – gain coefficient of a servo valve, $u(t)$ – control signal, i.e. voltage on the servo valve's input.

The given equations (1), (2), (3) and (4) with additional expressions such as function of viscosity make a mathematical model of an ATHB which is going to subjected to study in the present paper.

## III. Numerical Results and Discussion

Problems of stability are the primary problems that are solved when developing bearings for high-speed rotating machinery. Axial stability is considered in various types of pumps for oil extraction, aircraft and spacecraft engines, etc.

The purpose of the present study is to estimate the influence of the control system on the axial stability and energy efficiency of an ATHB. Stability analysis is based on estimation of the magnitude of oscillations of a rotor under complex external axial loading and relies on the premise that the desired position should be as close to the set point as possible. Energy efficiency estimation is based on calculation of power loss due to friction:

$$N_{FR}=M_{FR}\omega,$$

where $M_{FR}=r\int\limits_{Rin}^{Rout}\int\limits_{0}^{\Theta}\left[\frac{h}{2}\frac{\partial p}{r\partial\theta}+\frac{\omega r\mu}{h}\right]rd\theta dr$ - friction torque.

Operation of the ATHB has been modelled under the influence of a following external axial force:

$$F(t)=A_D\sin\left(\frac{\omega t}{10}+\frac{\pi}{2}\right)\sin\left(\frac{\omega t}{5}+\frac{\pi}{2}\right)-A_D,$$

where $A_D$ – amplitude, taken as 25, 35, 50, 75 N.

In the Fig. 7 results of calculation of rotor's trajectories are shown for various $A_D$ settings.

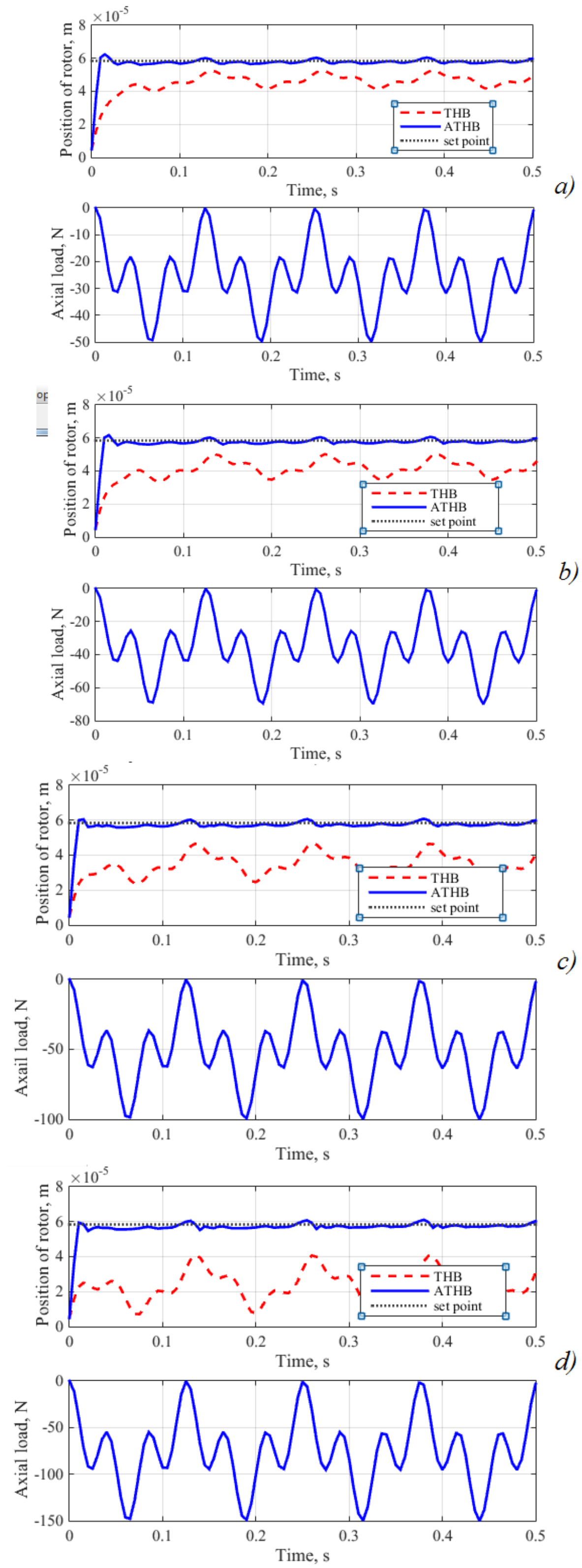



Fig. 7 results of rotor's trajectories calculation under various axial load (a) - $A_D$ =25, b) - $A_D$ =35, c) - $A_D$ =50, d) - $A_D$ =75)

First, the regulator has been tuned manually based on minimum transition period and control error. Chosen regulator parameters are: Kp = 7.5, Ki=15, Kd=0.05.

Throughout this part the following modelling parameters were used : *Rout* = 0.06 m, *Rin* = 0.03 m, α = 0.25°, $z_p$=6, lubricant – $H_2O$ @ $23^0$C , $\omega$ = 500 rad/s, Tsv = 0.005 s, rotor's mass is 5.5 kg.

In the Fig. 8 comparison results are shown for THB and ATHB regarding the stability and energy efficiency enhancement.

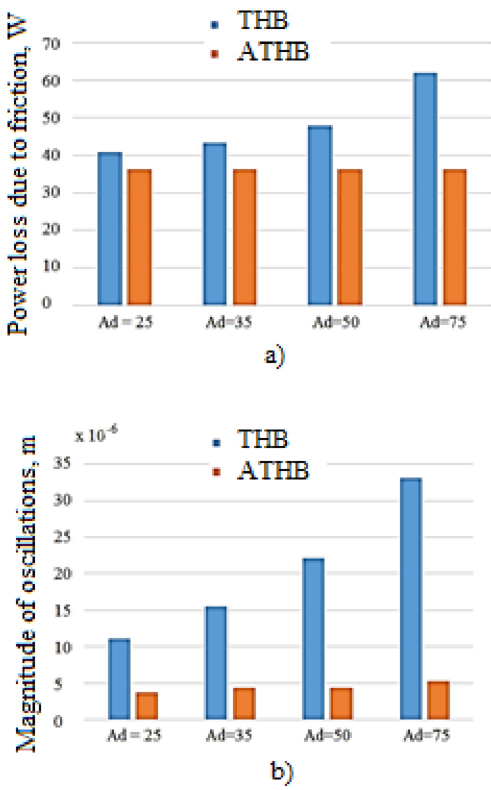



Fig. 8 comparison results of operation of THB and ATHB

During simulations, magnitude of oscillations has been calculated for comparison purposes, and at frequency of 100 Hz the data on power loss has been collected. Then, a mean value of power loss due to friction has been determined.

It could be seen, that application of ATHB significantly decreases the magnitude of oscillations thus enhancing stability (Fig. 6,a). With a practically constant magnitude in the case of an ATHB, magnitude of oscillations of a rotor increases in the case of a passive THB with the increase of the amplitude of the external axial force. Decrease of magnitude by the ATHB in this case is up to 85 % and the resulting trajectories are considered stable in comparison to a passive THB. Moreover, the position of the rotor only slightly deviates from the set point, which is very important in terms of positioning accuracy.

Comparison of power loss due to friction (Fig. 6,b) shows a 42% decrease in the case of the ATHB. And once more, the value of power loss remains constant in the whole range of simulated amplitudes, whereas in the THB the power loss increases with the increase of the amplitude of the axial external force.

Based on the obtained numerical results, it could be highlighted that the application of ATHBs significantly increases stability and energy efficiency of a rotor machine. In case a machine is subjected to changing loads and needs to be designed with a sufficient level of rotor's positioning accuracy, it could be seen, that introduction of a relatively simple in terms of simulation and execution control system greatly improves the performance of the bearing and the machine at a whole.

## IV. Experimental Setup

In order to verify the presented model and possibly identify some new or unaccounted for effects, a test rig has been developed. A series of experiments has been planned for carrying out in the nearest future to support the idea of an ATHB. A photo of the test rig is presented in the Fig. 9 and in more detail in the Fig. 10.

The main advantage of the developed test rig is possibility to study both radial and thrust active bearings at the same time. For verification purposes within the present research it is planned to only focus on ATHBs, so instead of sliding journal bearings, two rolling element bearings are installed so that there is such radial gap that the rotor could perform axial motion without restraining friction in these bearings. These bearings also provide higher stability in radial direction.

An ATHB is installed in its housing where one side is made of plexiglass in order to provide some visual information like observable cavitation or turbulence, etc. In Fig. 10a and Fig. 10b more detailed photos of the servovalve setup and the ATHB are presented.

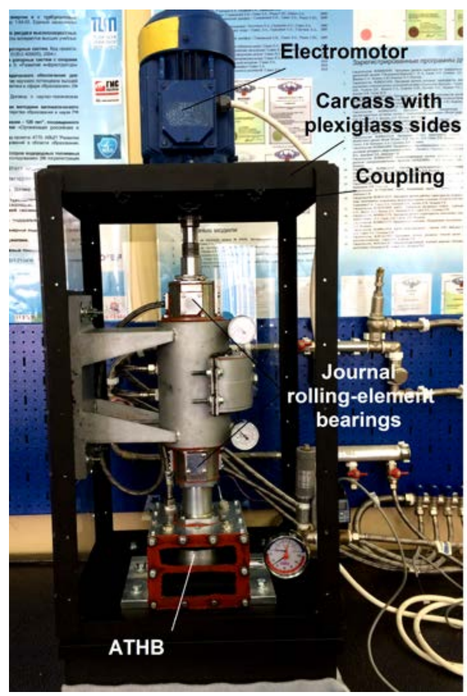
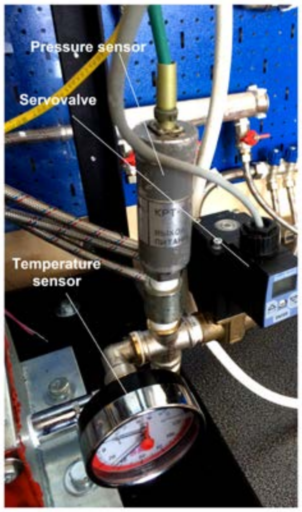
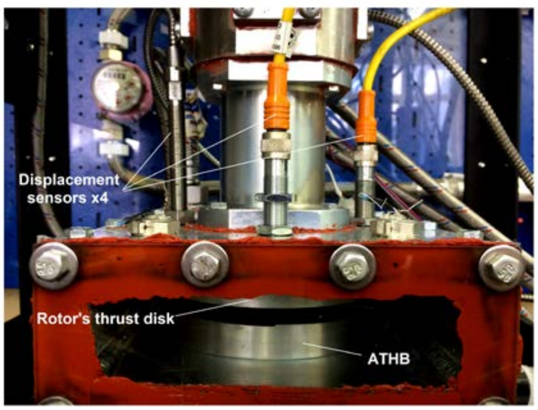



Fig. 9 test rig with an ATHB (general view)




a)




b)

Fig. 10 a) – servovalve setup, b) – ATHB in more detail

As shown in the Fig. 10a, pressure is controlled right before the fluid enters the feeding chamber, and it is possible to also determine the temperature of the inflow. In order to determine, whether radial oscillations occur that influence the performance of the ATHB and to possible determine shape deviation of the rotor's thrust disk, four displacement sensors are mounted around the thrust disk (Fig. 10b). One of these sensors is used to acquire data about rotor's position for the control system to be used as current rotor's position.

## V. Results and discussion

Results presented in the present paper show feasibility of application of active pressure control to increase axial stability of rotors with thrust hybrid bearings. Numerical results show a more than 80% decrease of amplitude of vibration if active control is applied. It has also been determined, that with varying amplitude of the external force the actual rotor's displacement on the ATHB is more stable, while amplitude of vibrations of the rotor on the passive THB increases.

It has also been determined that energy efficiency of the rotor system increases in terms of decreased power loss due to friction in the bearing. Along with decreased amount of energy dissipated due to vibrations, positioning of the rotor so, that the height of the axial gap provides less friction and enough hydrodynamic force, allows energy efficiency to be significantly increased.

Finally, a test rig has been developed to verify the obtained numerical results. Experimental studies are the first step of future research, as it provides initial data about the accuracy of the presented model. Possible identification of effects that have not been accounted for allows development of a more accurate model.

## Acknowledgments

This work was supported by the Ministry of Education and Science of the Russian Federation under Project no. 9.2952.2017/4.6 (development and modernization of the test rig) and by the Russian Science Foundation under Project no. 16-19-00186 (formulation and solution of the problem in all sections of the present paper).

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