

Comparison of the Dynamic Characteristics of Active and Passive Hybrid Bearings

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Abstract—One of the ways of reducing vibroactivity of rotor systems is to apply active hybrid bearings. Their design allows correction of the rotor's location by means of separately controlling the supply pressure of the lubricant into the friction area. In a most simple case, the control system is based on a P-regulator. Increase of the gain coefficient allows decreasing the amplitude of rotor's vibrations. The same effect can be achieved by means of increasing the pressure in the collector of a traditional passive hybrid bearing. However, these approaches affect the dynamic characteristics of the bearing differently. Theoretical studies show that the increase of the gain coefficient of an active bearing increases the stiffness of the bearing, as well as the increase of the pressure in the collector. Nevertheless, in case of a passive bearing, the damping properties deteriorate, whereas the active hybrid bearings obtain higher damping properties, which allow effectively providing the energy dissipation of the rotor vibrations and reducing the load on the constructional elements of a machine.

Keywords—Active bearings, control system, damping, hybrid bearings, stiffness.

I. INTRODUCTION

INCREASED vibroactivity in the rotor machines is one of the typical problems, which have to be dealt with during design and operation of such devices. Vibrations lead to the occurrence of increased load on the elements of the rotor machine. Here, the lifetime of such elements, as bearings and dampers, significantly decreases, for they are exposed to the most of the load. So, the problem of decreasing the vibroactivity of the rotor-bearing systems is a major and yet typical task. In the majority of cases, to provide the acceptable levels of vibration it is sufficient to qualitatively balance the rotor. However, in a number of cases during the operational cycle, it is either the external conditions or internal characteristics that change, which affects the operation of the whole machine. The internal factors include the changing imbalance of the rotor, which occurs due to the adverse technological environment, e.g. adhesion of polluting elements of the surroundings, or emergencies, e.g. damage of the system's elements. In such cases, it is reasonable to apply special means of decreasing vibroactivity, which include active hybrid bearings (AHB), that combine both hydrostatic and hydrodynamic effects while creating the load capacity.

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The aim of the present paper is to study the effect of the control system on the dynamic characteristics of the fluid-film bearings, such as stiffness and damping, which define the behavior of the rotor in the bearing, and to evaluate the efficiency of AHB in comparison to the traditional passive hybrid bearings.

II. AHB AS MEANS OF DECREASING THE VIBROACTIVITY OF THE ROTOR

One of the most obvious means of decreasing the amplitude of the rotor's oscillations is to increase the stiffness of the fluid-film bearing. In the case of the hybrid bearing, it can be achieved by increasing the supply pressure of the lubricant in the collector, from where the lubricant reaches the area of friction. Here, the reaction of the fluid film significantly increases and the amplitude of the rotor's displacement decreases. Another way to achieve this is to selectively increase the pressure separately in the feeding chambers of the fluid-film bearing. This pressure counteracts the undesirable displacement of the rotor and decreases the amplitude of the oscillations. Such principle is implemented in AHB, the general scheme of which is shown in Fig. 1.

AHB is a radial fluid-film bearing where four feeding chambers are distributed evenly along the inner surface of the circle of the sleeve 1. In order to obtain the data on the current location of the rotor 2 in the bearing, the displacement sensors 4 are fixed in the housing of the bearing. The sensors are located orthogonally, which allows to obtain two-coordinate data of the rotor's position. The signal is transmitted by the ADC to the computer, where the program of the regulator is designed. The data are then processed to generate the control signals, which are in their turn transmitted to the servovalves by the DAC. The servovalves are electrohydraulic devices which allow smooth regulation of pressure. The number of servovalves corresponds to the number of feeding chambers of the fluid-film bearing, every servovalve allows controlling the pressure in one corresponding chamber. The pressure in the chamber affects the pressure distribution in the lubricant film and forms a force acting on the rotor. This force can result in a change of rotor's position in the bearing, in other words, the problem of rotor's positioning is solved.

The distinguishable feature of the operation of a perfectly balanced rotor is absence of vibrations, and here the center of the rotor remains static. It takes a certain spot along the dynamic equilibrium curve (Fig. 2). This curve is a set of points which characterize the location of the balanced rotor's center depending on the change of one of the operational parameters, first of all – rotational frequency.

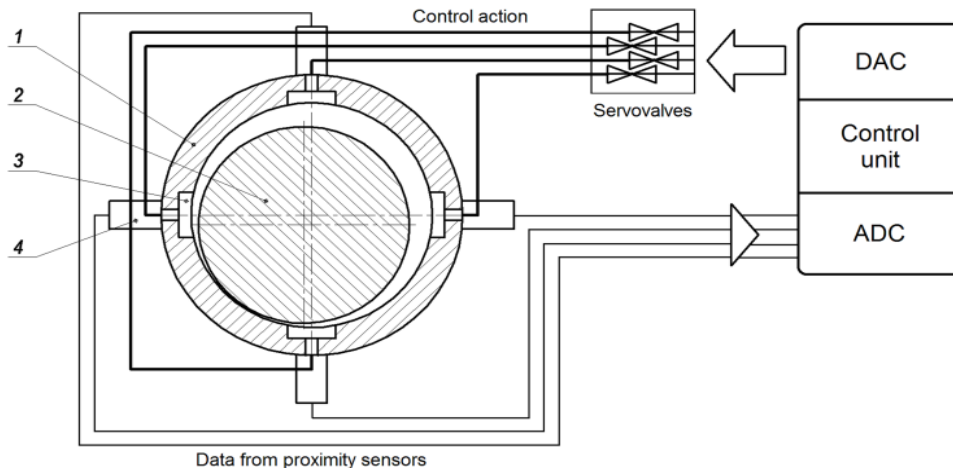


Fig. 1 Scheme of an AHB

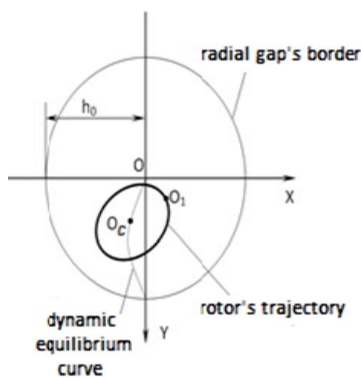


Fig. 2 Unbalanced rotor's trajectory

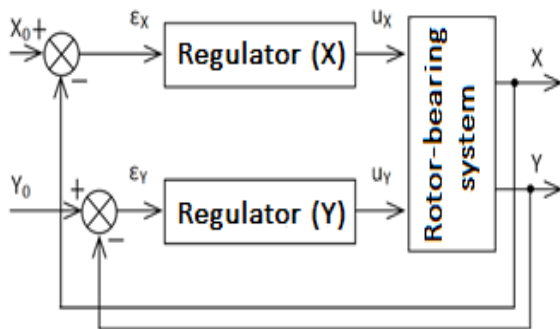


Fig. 3 The general scheme of the ACS

The center of an unbalanced rotor, on the other hand, moves along the closed curve around the point O_c , where the center of the balanced rotor is located. So, the vibrational motion of the rotor is formed, and in the general case, the frequency of such oscillations matches the rotational frequency of the unbalanced rotor. The aim of the control system of the hybrid bearing is, in terms of what has been said previously, to minimize the size of the rotor's trajectory.

The control system of an AHB is a closed-loop system. The control process is implemented by two independent loops,

where the X and Y coordinates of the rotor are the controlled parameters. The general scheme of the automatic control system (ACS) is shown in Fig. 3.

To implement the control, a constant calculation of the control error takes place:

$$\begin{cases} \varepsilon_X = X_0 - X; \\ \varepsilon_Y = Y_0 - Y. \end{cases} \quad (1)$$

where, X, Y – current coordinates of the rotor; X_0, Y_0 – set point of the rotor's location along the X and Y axes, that corresponds to O_c .

The control system of an AHB uses a P-regulator, where the control signal depends proportionally on the value of the control error:

$$\begin{cases} u_X = K_{pX} \cdot \varepsilon_X; \\ u_Y = K_{pY} \cdot \varepsilon_Y. \end{cases} \quad (2)$$

where K_{pX}, K_{pY} – gain coefficients of the P-regulator.

To study the characteristics of rotor system, a complex mathematical model of a rotor in an active hybrid fluid-film bearing was developed. Modeling was based on the fundamental equations of hydrodynamics and mechanics. Such approach is generally accepted and used by many authors when modeling the rotor systems with fluid-film bearings, for instance [1]-[3]. The developed program was used to model the operation of the rotor system with the following parameters. Bearing type – hybrid, active/passive, with 4 feeding chambers, length of the bearing $L=63$ mm, diameter $D=40$ mm, radial gap $h_0=100$ μm , length of the channel - $l_H=8$ mm, diameter $d_H=1$ mm; lubricant – water, initial pressure in the feeding chambers $p_0=0.4$ MPa; rotor's mass $m=3$ kg, imbalance $\Delta=2 \cdot 10^{-5}$ m, rotational frequency of the rotor 3000 rev/min.

The control system's parameters were taken as $K_p=\{0;1;2;5;5\}$ in order to model the operation of the ACS, here the 0 value of the proportional coefficient corresponds to

the absence of control and the operation of the AHB in a passive hybrid mode. This is justified by the fact that with any given control error in this case the control signal will be 0 and the pressure in all the chambers will remain constant. The set points were chosen according to the calculations of the center of the perfectly balanced rotor in a system, which operates under the same conditions. The modeled system initially was operating in a passive AHB mode, after $t=50$ ms the ACS was activated. The modeled rotor's trajectories are shown in Fig. 4.

As it can be seen from the modeling results, the activation of the ACS resulted in a decrease of the rotor's precession's amplitude. With an increase of the gain coefficient of the ACS, this effect enhanced and the amplitude decreased.

In order to compare the operation of the AHB with a passive hybrid bearing, where the decrease in the oscillations' amplitude is achieved by means of increasing the pressure in the collector, another series of trajectories were modeled (Fig. 5).

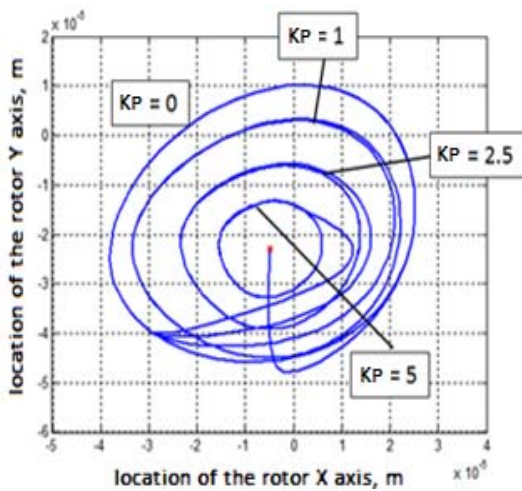


Fig. 4 Rotor's trajectory with various values of K_p

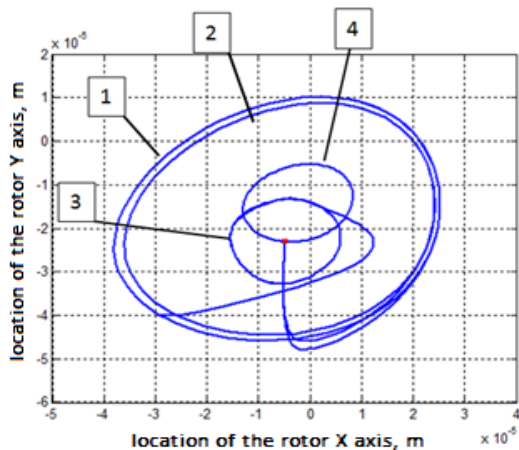


Fig. 5 Rotor's trajectories in AHB and passive hybrid bearing

The trajectory 1 corresponds to the behavior of an unbalanced rotor in a passive hybrid bearing with supply

pressure in the feeding chambers $p_0=0.4$ MPa. When an AHB was applied based on P-regulator with gain coefficient $K_p=5$, the trajectory 5 was obtained. Here, the amplitude decreased by 64% in comparison with the initial one, and the peak pulsations of pressure in the feeding chambers were registered at 0.52 MPa, which is 30% higher than initial pressure p_0 . Static increase of pressure in the collector of the passive hybrid bearing up to the same level of $p=0.52$ MPa resulted in a decrease of the amplitude by 3.5% relatively to the initial (trajectory 2). In order to achieve the trajectory, close to the trajectory 3 in Fig. 5, pressure in the collector has to be increased up to $p=3$ MPa, which is 7.5 times more than the initial pressure p_0 . In this case, the trajectory moves towards the center of the bearing's cross-sectional area (trajectory 4) due to a significant increase of bearing's stiffness.

To study and compare the dynamic characteristics of an active and a passive hybrid bearing, the stiffness and damping coefficients were modeled for the corresponding schemes.

III. DYNAMIC CHARACTERISTICS OF AN ACTIVE AND A PASSIVE HYBRID BEARING

To study the dynamic characteristics of the fluid film of the AHB, the rotor is represented as a single-mass oscillator which is supported by a system of elastic and damping elements (Fig. 6). Due to the smallness of the rotor's displacement, which is considered when studying damping and stiffness of the bearing, the non-linear properties of the fluid film can be neglected, and the system can be linearized. Such approach is justified, for instance, in [4]-[6].

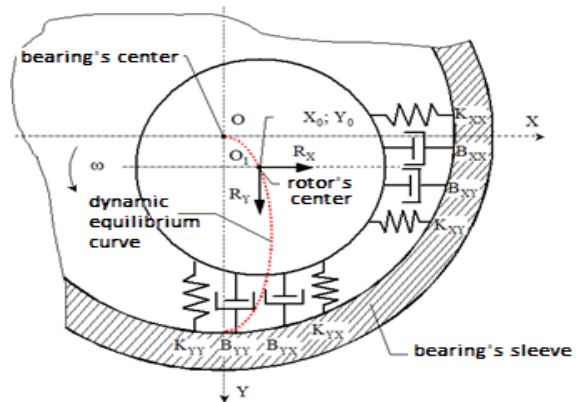


Fig. 6 Dynamic linearized model of the rotor system

To obtain the values of the damping and stiffness coefficients, a specific method which is based on a direct numerical integration of the modeled parameters of the rotor-bearing system in accordance to the calculation scheme (Fig. 7) was used.

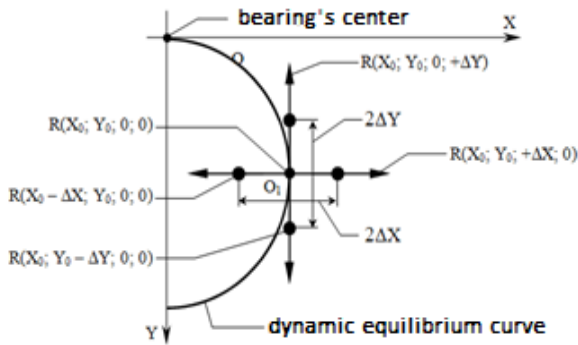


Fig. 7 Determination of the dynamic coefficients of the lubricant's film of the AHB

The expressions to determine the dynamic coefficients of the AHB take the following form [4]:

$$\begin{aligned} K_{X(Y)X} &\approx \frac{R_{X(Y)}(X_0 + \Delta X; Y_0; 0; 0) - R_{X(Y)}(X_0 - \Delta X; Y_0; 0; 0)}{2\Delta X}, \\ K_{Y(X)Y} &\approx \frac{R_{Y(X)}(X_0; Y_0 + \Delta Y; 0; 0) - R_{Y(X)}(X_0; Y_0 - \Delta Y; 0; 0)}{2\Delta Y}, \\ B_{X(Y)X} &\approx \frac{R_{X(Y)}(X_0; Y_0; +\Delta X'; 0) - R_{X(Y)}(X_0; Y_0; -\Delta X'; 0)}{2\Delta X'}, \\ B_{Y(X)Y} &\approx \frac{R_{Y(X)}(X_0; Y_0; 0; +\Delta Y') - R_{Y(X)}(X_0; Y_0; 0; -\Delta Y')}{2\Delta Y'}. \end{aligned} \quad (3)$$

Since the aim of the present research is to study the influence of the control system on the process of rotor displacement control, first of all, the change of the dynamic coefficient with various control parameters is of major interest while the absolute values of the damping and stiffness coefficients are inessential. Here, the direct coefficients K_{XX} , K_{YY} , B_{XX} , and B_{YY} consist of the full information on the relationships between the change in reaction forces of the bearing and the cinematic parameters of the rotor. The calculation of the cross coefficients is based on the same initial values and so can be omitted in order to simplify the representation and analysis of the results of the AHB research.

Due to the fact that the dynamic coefficients in (3) are determined for a static state of the rotor with its center on the dynamic equilibrium curve, the perfectly balanced rotor was modeled with zero imbalance. To create the displacements ΔX and ΔY in the corresponding directions, the external forces were applied as short-term force of 30 N equal to the inertia force which would act on the rotor with a mass of 3 kg. The force is applied to the rotor after it took a stable position on the dynamic equilibrium curve. For the cases with an active control system, the coordinates of the stable position of the rotor on the passive hydrodynamic bearing were chosen as set points.

To implement the comparative analysis, the dynamic coefficients of the lubricant's layer were modeled for a passive hydrostatodynamic bearing and the AHB, where the P-regulator is used and has constant coefficients $K_P=2.5$ and $K_P=5$. The obtained results of the modeling are shown in Fig. 8 for the stiffness coefficients and in Fig. 9 for the damping

coefficients.

From the obtained graphs, it is clear that the increase of the regulator's coefficient K_P results in the increase of the stiffness of the bearing due to the occurrence of additional pressure which counteracts the displacement of the rotor. The graph that represents the change of the stiffness coefficient relatively to the increase of the rotational frequency, for different ACS gain coefficients is positioned throughout the whole range of frequency parallel to the graph for the passive bearing. Here, there is a mutual tendency of stiffness coefficients to increase with the increase of the rotational frequency, which happens due to the increase of the hydrodynamic forces. So, whereas the absolute additional value of the stiffness coefficients remains practically constant for the whole range of rotational frequencies, the action of the naturally increasing hydrodynamic forces results in a relative increase of the stiffness coefficients decreases as the rotational frequency gets higher. At the rotational frequency of 500 rev/min the increase of K_P from 0 to 5 resulted in the increase of K_{YY} by two times, and at the rotational frequency of 3000 rev/min K_{YY} increased by 1.6 times. So, as the rotational frequency of the rotor in the AHB increases, the additional stiffness, created by the ACS gets less and less significant in comparison to the full stiffness of the bearing, provided by the increasing hydrodynamic forces.

Here one should note that the negative influence of the increased stiffness of the bearing on the transmission of vibrations from the rotor system to the constructional elements of the machine increases with the increase of the rotational frequency of an unbalanced rotor and, as a result, of the values of the disturbing forces. So, in the range of the frequencies where this effect can have a significantly negative influence on the operational process of the rotor machine, the increase of stiffness due to the presence of the ACS can be considered insignificant in comparison to the full stiffness of the bearing, and the application of the AHB to decrease the load on the machine's elements appears reasonable.

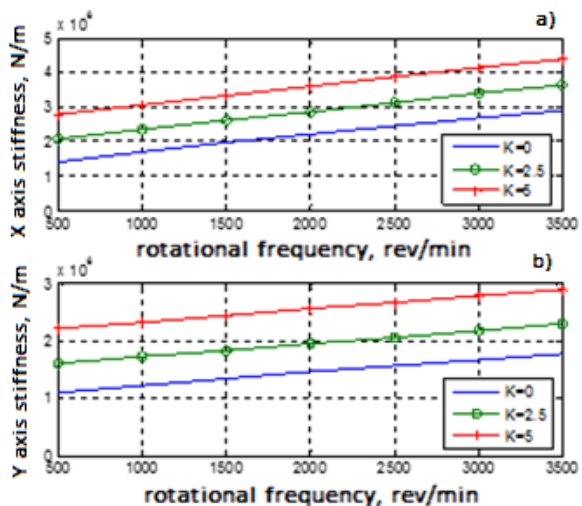


Fig. 8 X axis (a) and Y axis (b) stiffness coefficients

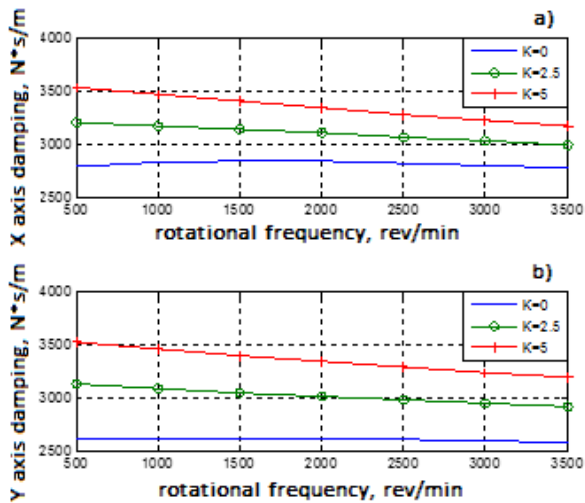


Fig. 9 X axis (a) and Y axis (b) damping coefficients

Damping coefficients of the fluid film of the AHB increase with an increase of the gain coefficient K_p . Since the AHB is used to decrease the vibrations level and other unwanted rotor displacement, this effect can be considered positive, for it provides higher level of energy dissipation and thus decrease of the amplitude of such displacement. With an increase of the rotational frequency the damping coefficients decrease, which occurs due to the effect of generation of additional control

pressure of similar value with increasing hydrodynamic pressure. At the same time, the mutual tendency of increasing damping properties with an increase of the gain coefficient K_p in comparison to the passive hybrid bearing preserves, which also means that the AHB is applicable to decrease the load on the machine at a whole. Similar graphs are shown in Figs. 10 and 11 for the passive bearing, here the K coefficient represents the increase of the pressure in the collector relatively to p_0 .

From the obtained results, it is clear that the increase of pressure in the collector results in a significant increase of the stiffness of the passive bearing. Here, the decrease of the ratio between the additional stiffness and the full stiffness of the bearing, unlike in the AHB, is indistinct. Stiffness contribution due to the increase of pressure is significantly greater than the contribution by the hydrodynamic forces.

Damping capacity of the lubricant's layer of the passive hybrid bearing with the increase of the pressure in the collector, on the other hand, unlike AHB, decreases proportionally to the increase of pressure. This leads to the decrease in energy of the unwanted displacement dissipation in the fluid film. Given the significantly increased stiffness of the bearing, this situation leads to the increase of the load on the machine's elements.

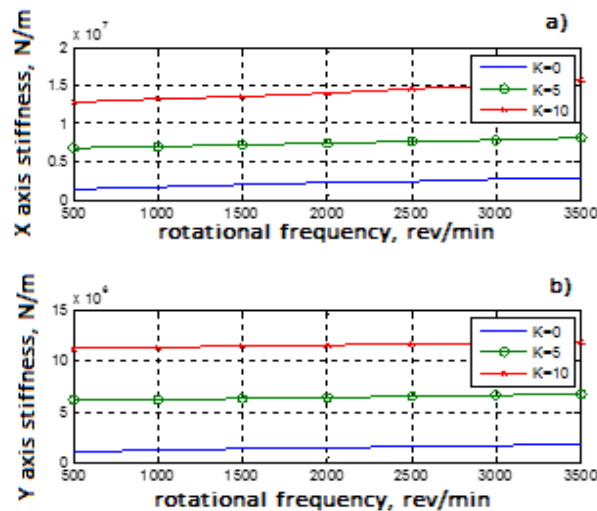


Fig. 10 X axis (a) and Y axis (b) stiffness coefficients

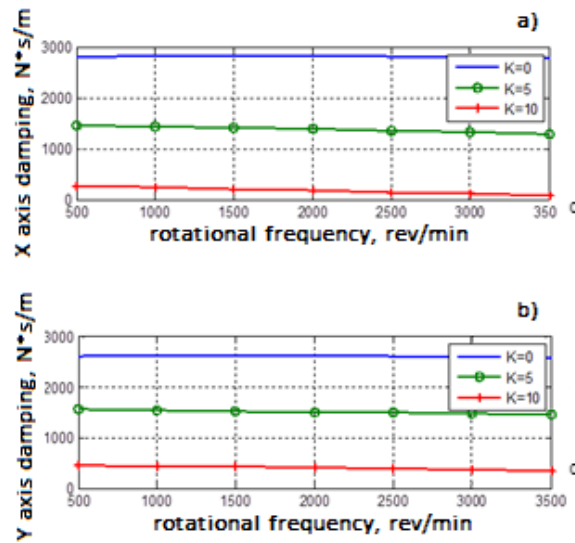


Fig. 11 X axis (a) and Y axis (b) damping coefficients

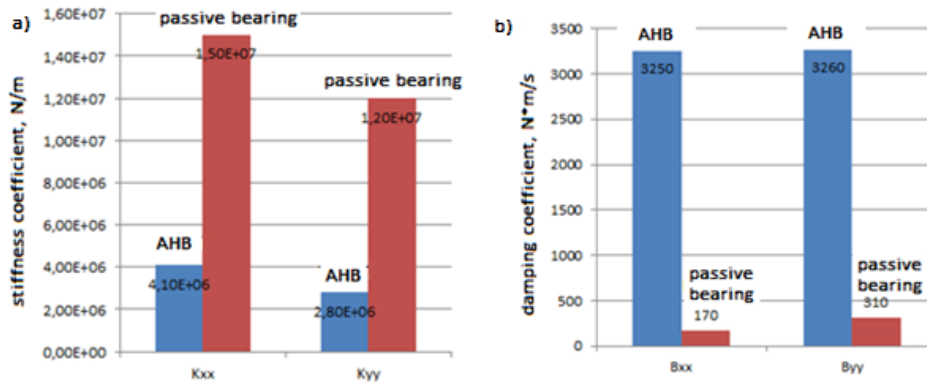


Fig. 12 Comparison of the dynamic coefficients of the AHB and the passive hybrid bearing

For a more detailed comparison of the active and passive bearings, the given parameters were modeled for similar operational conditions. In case of the AHB, its operation was modeled with $K_p = 5$, which corresponds to the trajectory 3, shown in Fig. 5. In case of the passive bearing, its operation was modeled with the parameters that provided the trajectory 4 in Fig. 5. So, both bearings provide a proportionate decrease of the amplitudes of the rotor's vibrations. The comparison of the stiffness (a) and damping coefficients (b) is shown in Fig. 12. As it can be seen from the diagram, stiffness of the passive bearing is 3.6 – 4.3 times higher than the stiffness of the active bearing. At the same time, the damping coefficient in the passive bearing is order of magnitude less than in the AHB. Such ratio of the parameters means that the AHB under the similar parameters of rotor's vibroactivity reduction provides much better dynamic coefficients. Energy of the mechanical oscillations in the active bearing will be transmitted to the fluid film, increasing its temperature, and then the lubricant will be removed from the friction area to cool down outside the bearing. In the passive bearing, on the contrary, the rotor's

vibration will be transmitted straight to the housing elements of the machine. In the majority of cases, this leads to shortening of the lifetime expectancy of such elements.

IV. CONCLUSION

The paper features the research on the dynamic characteristics of the active and passive hybrid bearings by means of modeling the rotor system's operation. The comparison and the analysis of the obtained data allow the following conclusions to be made:

- 1) Application of the rotor's position control system with separate control of the pressure in the feeding chambers of the hybrid bearing allows effectively decreasing the rotor's oscillations' amplitudes, which are caused by the imbalance, and here the extent of the amplitude's decrease enhances with the increase of the gain coefficient.
- 2) The control scheme of the AHB provides better results, than a proportional increase of pressure in the collector.

The same efficiency can be shown by the passive bearing only when the pressure is more than 5 times higher than initial.

- 3) The dynamic characteristics of the hybrid bearing in terms of decreasing the vibroactivity of the rotor machine exceed the dynamic characteristics of the passive hybrid bearing. The application of the AHB in rotor machinery allows the dissipation of energy and vibration's energy transmission at a higher rate, compared to the passive bearing. At the same time, the load on the machine's elements also decreases, and as a result, the lifetime expectancy is enhanced.

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