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Diagnostic research of adaptive hydrodynamic segmental bearings with elastic ring in a rotor system

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1. Introduction

Hydrodynamic bearings are the bearings of fluid friction, the work of which is based on forming lubricant wedge. The wedge is formed through eccentric rotation of rotor [1]. Properties of hydrodynamic supports compared with bearings of other types are these: high accuracy of rotation, big durability, good characteristics of vibrations decrement, and high rotation speeds [2]. So these bearings are used generally for fast-moving mechanisms that are high loaded, where roll bearings could not work: for rotors, for modern turboagregates, for powerful systems of refrigeration and so on [3-6].

There are a lot of bearings of different designs that are working with sliding friction, designs of some of them are very similar, but every element is of some specific work and design characteristics, e.g. increased vibrations, increased friction, decreased bearing capacity and so on [7]. All these indices are very important, because on it depend not only rotary systems, but work productivity, quality and other parameters of all mechanisms [1, 3, 8-10].

One of unacceptable phenomenon that occurs in rotary systems is the operation in resonant regime [11]. Rotors of latter-day machines are rotating with high frequencies, the implication is that these systems are working in very large diapason of vibrations, in what can be resonant frequencies of several rows, selfvibrations and so on [9, 11]. At rotation at resonant frequency of rotor rotation, dynamic powers increase, decrease sustained work time of machine parts, they are wearing or broken, etc. [12]. The resonant regime is attributed to crash of regimes of bearing work, because at the work in such a regime all the mechanism, in which the bearing is embedded, can be damaged. Using rotary systems with different types of bearings the very important is not only to know resonant and other dangerous frequencies of the construction, but to have the construction, that would give opportunity of undisturbed stable work of the machine [9, 11, 13-5].

2. Research object

Different additional elements, such as contacts of segments, are used to upgrade the work characteristics of adaptive hydrodynamic bearings. Separate strips or one elastic ring [7] could be used to connect bearing segments. Such construction could get stable work of rotors in wide diapason of rotation frequencies. The construction of a bearing with the connecting ring which connects the segments is researched in this article.

Fig. 1 shows such adaptive hydrodynamic bearings with the elastic ring.

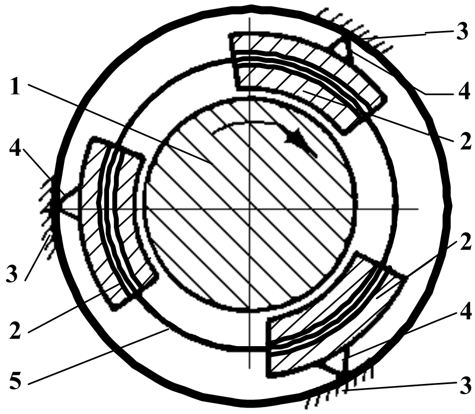


Fig. 1 Adaptive hydrodynamic bearing with elastic ring which is easily moved and connects the segments: *1* - rotor, *2* - segments, *3* - spindle neck, *4* - adaptive thrust, *5* - elastic ring

More adaptive hydrodynamic bearings of similar designs are described in scientific literature [2].

3. Experimental stand

The research stand Fig. 2 was used for experiments. The stand consists of the spindle head that is composed of rotor, adaptive hydrodynamic segmental bearings, lubrication and refrigeration system, regulator of rotor step-less rotation frequency, electromotor, vibration measurement and result analysis system.

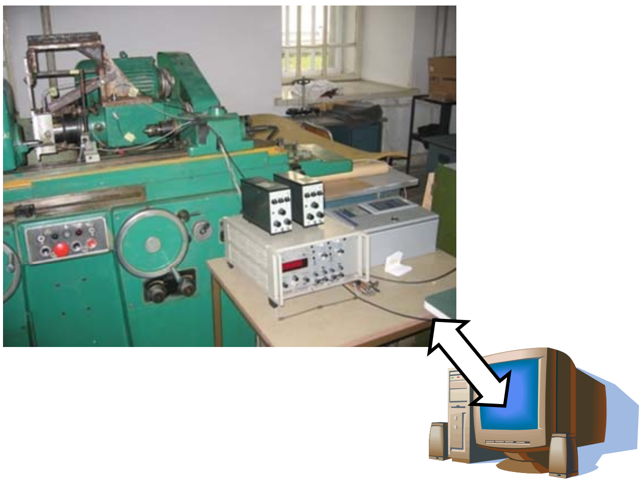


Fig. 2 Stand of experimental researches

Segments of adaptive hydrodynamic bearings with elastic ring are assembled in this research stand; other adaptive hydrodynamic bearings of similar designs could be assembled in it.

More similar research stands are described in different science works [1, 6, 12].

4. Methodology of result analysis and experimental measurements

Experimental researches were done according to special methodology.

1. All transducers (vibrochanges, vibrospeed, vibroacceleration, phase) that are appropriate for vibromeasurements are installed.
2. Boosters, supply units and computers are interconnected.
3. Computer, supply units and boosters are actuated.
4. Transducers are adjusted.
5. Regulation component (EMOTRON EDU 2.0) of stepless rotation frequency of electric motor is actuated and wanted frequency of rotation is determinate (vibromeasurements were done at rotor rotating with the frequency from 0 to 8000 rev/min, measurement results were fixed gradually by 100 rev/min..
6. Special fixing computer programme „Experiment“ of data is actuated.
7. Vibromeasurements are done.
8. Primary received measurement signals are analyzed, different primary data formats (vibrochanges, vibroaccelerations, vibropowers) and data analyze formats (spectrums, orbits, correlations and others) are obtained by the analysis of vibromeasurement results.
9. Experiments are repeated changing frequency of the rotor rotation. Primary vibrosignals of rotor rotation with different frequencies are obtained, etc.
10. Results obtained from experimental vibromeasurements are analyzed with the help of different computer programmes (Origin, Master Data, Excel, LabView, CADMS, Matlab, Simulink and others).
11. Analysis of experimentally obtained research results for determining the factors influencing the searched system negatively and positively is performed.

5. Rotation of elastic ring

In the coordinates rotating together with the elastic ring the equation of its motion, taking into account the Carioles and centrifugal accelerations as with some simplifications and assumptions, will be as follows

$$\left.\begin{aligned}
&\frac{\rho r_y^2\left(1-\nu^2\right)}{E}\left(\frac{\partial^2 u_\varphi}{\partial t^2}+2\omega_y\frac{\partial u_r}{\partial t}-\omega_y^2 u_\varphi\right)-\frac{\partial^2 u_\varphi}{\partial\varphi^2}-\frac{1-\nu\partial^2 u_\varphi}{2}\frac{\partial^2 u_\varphi}{\partial\zeta^2}-\frac{\partial u_r}{\partial\varphi}-\frac{M_k\left(1-\nu^2\right)}{Eh_2\pi r_y^2}\left(\frac{\partial u_\varphi}{\partial\varphi}+\frac{\partial u_r}{\partial\zeta}\right)+\\
&+\frac{h_y^2}{12_y^2}\left(\frac{\partial^3 u_r}{\partial\zeta^2\partial\varphi}+\frac{\partial^2 u_r}{\partial\varphi^2}-\frac{\partial^2 u_r}{\partial\varphi^2}-\left(1-\nu^2\right)\frac{\partial^2 u_\varphi}{\partial\zeta^2}\right)=0\\
&\frac{\rho r_y^2\left(1-\nu^2\right)}{E}\left(\frac{\partial^2 u_r}{\partial t^2}+2\omega_y\frac{\partial u_\varphi}{\partial t}-\omega_y^2 u_\varphi\right)+\frac{\partial u_\varphi}{\partial\varphi}+u_r-\frac{M_k\left(1-\nu^2\right)}{Eh_y\pi r_y^2}\left(\frac{\partial u_r}{\partial\varphi}+\frac{\partial u_\varphi}{\partial\zeta}\right)+\\
&+\frac{h_y^2}{12_y^2}\left(\nabla^4 u_r-\left(2-\nu\right)\frac{\partial^2 u_\varphi}{\partial\varphi^2}-\frac{\partial^3 u_\varphi}{\nabla\zeta^2\partial\varphi}\right)-\frac{N_\varphi\left(1-\nu^2\right)}{Eh_y}\left(\frac{\partial^2 u_\varphi}{\partial\varphi^2}+\frac{\partial u_r}{\partial\zeta}\right)=0
\end{aligned}\right\}\qquad(1)$$

where M_k is rotational moment acting at the ends of the elastic ring; ω_y is angular velocity of the elastic ring. The set of motion Eq. (1) is solved in the form of running waves

$$\left.\begin{aligned}
u_\varphi &= A\cos\left(\bar{\lambda}\xi-n\varphi-\omega t\right)\\
u_r &= B\sin\left(\bar{\lambda}\xi-n\varphi-\omega t\right)
\end{aligned}\right\}\qquad(2)$$

where A, B are amplitudes of running waves; n is number of waves found on the circumference; ω is angular frequency of the running waves; L is ring length. By substituting Eqs.(1 and 2) we obtain the characteristic equation

$$\left.\begin{aligned}
&\left\{\frac{\left(1-\nu^2\right)}{2}\bar{\lambda}^2+n^2-\frac{\rho r_y^2\left(1-\nu^2\right)}{E}\omega_y^2-\frac{M_k\left(1-\nu^2\right)}{Eh_y\pi r_y^2}\bar{\lambda}n-\frac{\rho r_y^2\left(1-\nu^2\right)}{E}\omega_y^2+\frac{h_y^2}{12_y^2}\left[n^2+\left(1-\nu^2\right)\bar{\lambda}^2\right]+\right.\\
&\left.+\frac{N_z\left(1-\nu^2\right)}{Eh_y}\bar{\lambda}^2\right\}A+\left\{n-\frac{2\rho r_y^2\left(1-\nu^2\right)}{E}\omega_y\omega-\frac{M_k\left(1-\nu^2\right)}{Eh_y\pi r_y^2}\bar{\lambda}+\frac{h_y^2}{12_y^2}\left(n^3+\bar{\lambda}^2 n\right)\right\}B=0\\
&\left\{n-\frac{2\rho r_y^2\left(1-\nu^2\right)}{E}\omega_y\omega-\frac{M_k\left(1-\nu^2\right)}{Eh_y\pi r_y^2}\lambda+\frac{h_y^2}{12_y^2}\left[\left(2-\nu^2\right)\bar{\lambda}^2 n+n^3\right]+\frac{N_\phi\left(1-\nu^2\right)}{Eh_y}n\right\}A+\\
&+\left\{1-\frac{\rho r_y^2\left(1-\nu^2\right)}{E}\left(\omega^2+\omega_y^2\right)+\frac{h_y^2}{12_y^2}\left(\bar{\lambda}^2+n^2\right)-\frac{M_k\left(1-\nu^2\right)}{Eh_y\pi r_y^2}\bar{\lambda}n+\frac{N_\phi\left(1-\nu^2\right)}{Eh_y}n\right\}B=0
\end{aligned}\right\}\qquad(3)$$

Wave velocity for any rotational velocity of the cylinder is sought accepting that the determinant of Eq. (3) is equal to zero.

$$\begin{vmatrix} \frac{1-\nu}{2}\bar{\lambda}^2+n^2-\frac{\rho r_y^2\left(1-\nu^2\right)}{E}\omega_y^2-\frac{M_k\left(1-\nu^2\right)}{Eh_y\pi r_y^2}\bar{\lambda}n- \\ -\frac{\rho r_y^2\left(1-\nu^2\right)}{E}\omega_y^2+\frac{h_y^2}{12r_y^2}\left[n^2+\left(1-\nu^2\right)\bar{\lambda}^2\right] & n-\frac{2\rho r_y^2\left(1-\nu^2\right)}{E}\omega_y\omega-\frac{M_k\left(1-\nu^2\right)}{Er_y^2h_y\pi}\bar{\lambda}+ \\ & +\frac{h_y^2}{12r_y^2}\left(n^3+\bar{\lambda}^2n\right) \\ n-2\frac{\rho r_y^2\left(1-\nu^2\right)}{E}\omega_y\omega-\frac{M_k\left(1-\nu^2\right)}{Er_y^2h_y\pi}\bar{\lambda}+ & 1-\frac{\rho r_y^2\left(1-\nu^2\right)}{E}\left(\omega^2+\omega_y^2\right)+\frac{h_y^2}{12r_y^2}\left(\bar{\lambda}^2+n^2\right)- \\ \frac{h_y^2}{12r_y^2}\left[\left(2-\nu\right)\lambda^2n+n^3\right]+\frac{N_\varphi\left(1-\nu^2\right)}{Eh_y}n & -\frac{M_k\left(1-\nu^2\right)}{Er_y^2h_y\pi}\bar{\lambda}n+\frac{N_\varphi\left(1-\nu^2\right)}{Eh_y}n^2 \end{vmatrix}=0 \tag{4}$$

Expansion of the determinant (4) has the following form

$$\begin{aligned} &\left\{\frac{\left(1-\nu^2\right)}{2}\bar{\lambda}^2+n^2-\frac{\rho r_y^2\left(1-\nu^2\right)}{E}\omega_y^2-\frac{M_k\left(1-\nu^2\right)}{Eh_y\pi r_y^2}\bar{\lambda}n-\frac{\rho r_y^2\left(1-\nu^2\right)}{E}\omega_y^2+\frac{h_y^2}{12_y^2}\left[n^2+\left(1-\nu^2\right)\bar{\lambda}^2\right]\right\}\times \\ &\times\left[1-\frac{\rho r_y^2\left(1-\nu^2\right)}{E}\left(\omega^2+\omega_y^2\right)+\frac{h_y^2}{12_y^2}\left(\bar{\lambda}^2+n^2\right)^2-\frac{M_k\left(1-\nu^2\right)}{Eh_y\pi r_y^2}\bar{\lambda}n+\frac{N_\phi\left(1-\nu^2\right)}{Eh_y}n^2\right]- \\ &-\left[n-\frac{2\rho r_y^2\left(1-\nu^2\right)}{E}\omega_y\omega-\frac{M_k\left(1-\nu^2\right)}{Eh_y\pi r_y^2}\lambda+\frac{h_y^2}{12_y^2}\left(n^3+\bar{\lambda}^2n\right)\right]\times\left[n-\frac{2\rho r_y^2\left(1-\nu^2\right)}{E}\omega_y\omega-\right. \\ &\left.-\frac{M_k\left(1-\nu^2\right)}{Eh_y\pi r_y^2}\bar{\lambda}+\frac{h_y^2}{12_y^2}\left[\left(2-\nu\right)\lambda^2n+n^3\right]+\frac{N_\phi\left(1-\nu^2\right)}{Eh_y}n\right]=0 \end{aligned} \tag{5}$$

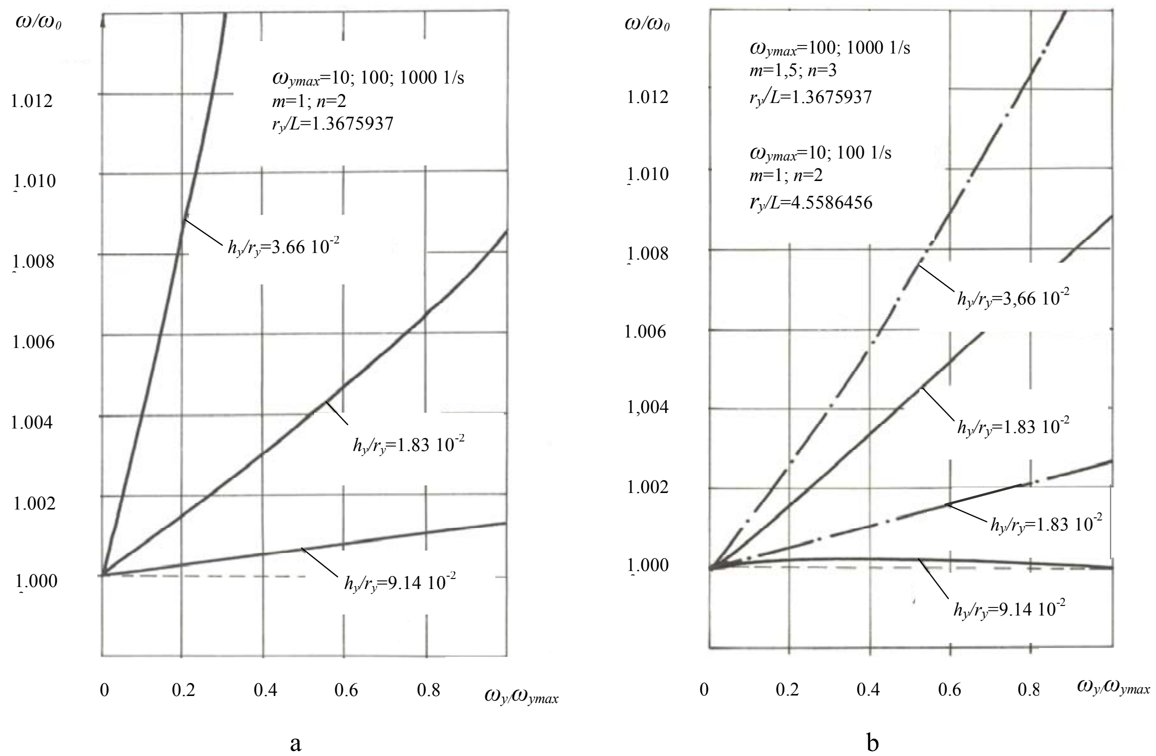
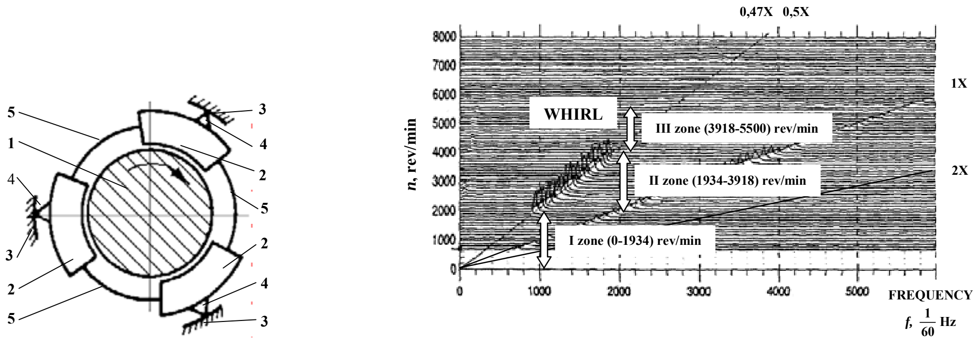


Fig. 3 Frequency response of angular velocity of the rotation of elastic ring: a - in the case of long elastic ring; b - in a case of the hydrodynamic suspension of short and long elastic elements

Calculation results of the fundamental frequencies of rotation velocity of the elastic ring are shown in the diagram Fig. 3. This diagram also shows the influence of geometrical parameters and other physical constants on the frequency change.

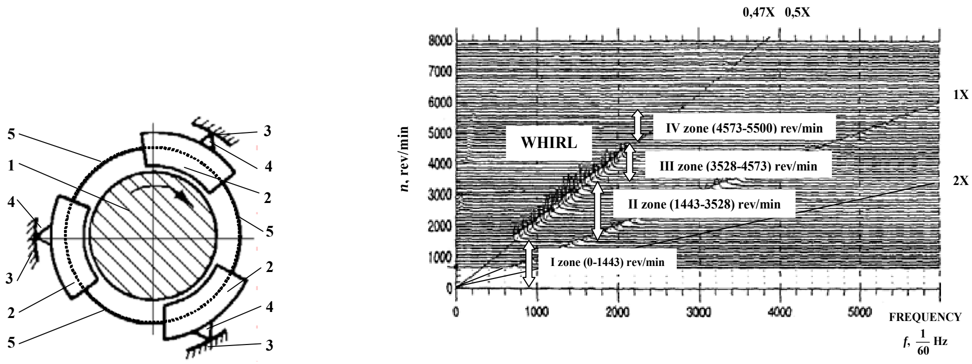
6. Results of measurement, their comparison, analysis and discussion

Comparing the obtained results with “Step of Bently and Muzcynska” graph [13] one can seen that between the rotary systems with simple sliding bearings and segmental bearings is a difference with respect so far as the bands of resonant frequencies.



a b

Fig. 4 Rotary system: a - adaptive hydrodynamic bearing with the segments connected with elastic strips: *1* - rotor, *2* - segments, *3* - body, *4* - adaptive tilting pad, *5* - elastic strips connecting segments; b - graph of resonant frequencies of rotary system, when clearance between rotor and bearing 50 μm



a b

Fig. 5 Rotary system: a - adaptive hydrodynamic bearing with elastic ring that is moved easily and is connecting segments: *1* - rotor, *2* - segments, *3* - body, *4* - adaptive tilting pad, *5* - elastic ring; b - graph of resonant frequencies of rotary system, when clearance between rotor and bearing 50 μm

Two rotary systems with similar constructions of adaptive hydrodynamic bearings are compared (Fig. 4, a and 5, a).

The system (Fig. 4, a) starts working from 0 to 1934 rev/min (I zone) and is working steadily in that range, when exceeding 1934 rev/min spontaneous vibrations of rotor appear, which continue up to 2921 rev/min. As the rotor reaches a rotational frequency of 2921 rev/min the spontaneous vibrations decrease and that effect continues until the frequency reaches 3918 rev/min (II zone). No spontaneous vibrations are observed when the spindle revolves in the range from 3918 rev/min to 5500 rev/min (III zone).

The system (Fig. 5, a) starts working from 0 to 1443 rev/min (I zone) and is working steadily in that range, after exceeding of 1443 rev/min spontaneous vibrations of rotor appear, which continue up to 2516 rev/min. After reach of that frequency (2516 rev/min) the spontaneous vibrations decrease and this effect continues until the frequency reaches 3528 rev/min (II zone). Rotation of the rotor is stabilized at range 3528-4573 rev/min (III zone). No spontaneous vibrations are observed when it revolves in the range from 4573 rev/min to 5500 rev/min (IV zone).

Elastic rings connecting segments of a bearing improve work characteristics of the bearing, because these bearings have two flows of lubricant: carrying and circulating. At circulating flows the liquid whirl through moving ring changes the influence of rotation frequency on excitation of vibrations.

Though researched rotary systems are working with great rotation frequencies, but in the limits from 0 to 5500 rev/min only one resonant frequency is outstretched.

Knowledge of resonant frequencies of dynamic system enables to escape the work that could lead to sudden breakdowns.

7. Conclusions

One can see that though the design of researched bearings is very similar, the results can be differenced by getting graphs of resonant frequencies.

One can see that if to compare Figs. 4 and 5 the system of adaptive hydrodynamic segmental bearings is more stable and work of the bearings (Fig. 5) is more secure. The research shows that it results less unstable work zone of this bearing. Lots of other rotary system bearings also are less stable.

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ROTORINĖS SISTEMOS ADAPTYVIŲJŲ HIDRODINAMINIŲ SEGMENTINIŲ GUOLIŲ SU TAMPRIU ŽIEDU DIAGNOSTINIAI TYRIMAI

R e z i u m ė

Straipsnyje aprašytas tyrimo objektas - adaptyvusis hidrodinaminis guolis su segmentus jungiančiu slankiuoju tampriuoju žiedu. Kai naudojami tokios konstrukcijos guoliai rotorius gali stabiliai dirbti platesniame sukimosi dažnių diapazone. Šie guoliai turi du tepalo srautus: nešantįjį ir cirkuliacinį. Tamprusis žiedas slopina tepalo cirkuliacinio srauto sūkurius ir taip mažina rotoriaus sukimosi dažnių įtaką virpesių žadinimui. Aprašyta eksperimentinių tyrimų įranga ir metodika. Gauti ir aptarti eksperimentinių matavimų rezultatai, suformuluotos išvados.

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DIAGNOSTIC RESEARCHES OF ROTOR SYSTEM ADAPTIVE HYDRODYNAMICAL SEGMENTAL BEARINGS WITH ELASTIC RING

S u m m a r y

Research object which is described in this article is adaptive hydrodynamic bearing with connecting moving ring which connects segments. This construction of the bearings gives more stable work of the rotors in the wider diapason of rotation frequencies. These bearings have two flows of lubricant: carrying and circulating. Elastic ring is damping whirls of circulating flow and decreasing the influence of rotor rotation frequencies on excitation of vibration. A mechanism of experimental research and methodology of experimental research is given. Results of the experimental research are obtained, they are discussed and conclusions are formulated.

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ДИАГНОСТИЧЕСКИЕ ИССЛЕДОВАНИЯ РОТОРНОЙ СИСТЕМЫ С АДАПТИВНЫМИ ГИДРОДИНАМИЧЕСКИМИ СЕГМЕНТНЫМИ ПОДШИПНИКАМИ С УПРУГИМ КОЛЬЦОМ

Р е з ю м е

В данной статье описан исследуемый объект – адаптивный гидродинамический подшипник с сегменты соединяющим упругим кольцом. С подшипником такой конструкции можно получить стабильную работу ротора в широком диапазоне вращения ротора. Эти подшипники имеют два потока масла: несущий и циркулярный. Упругое кольцо гасит вихри циркулярного потока масла и уменьшает влияние частоты вращения ротора на возбуждение колебаний. Описан стенд и методика экспериментальных исследований. Результаты экспериментальных измерений оговорены, сформулированы выводы.

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