915. Experimental investigation of dynamics of bearings with adaptive hydrodynamic elements under external effects

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Abstract. Operation of mechanical dynamic systems with adaptive hydrodynamic elements is investigated in this paper. Experimental research is performed taking the change of clearance between the rotors and bearing into account and investigating the effects of change of temperature and frequency of rotation of the rotor. Principle scheme of measurement is proposed. Matrix of planning of the proposed experiment is described. Principal system of measurement and method of analysis together with the proposed pattern of changes of searching parameters are presented and the obtained results of investigations are described in detail. Recommendations for the design of dynamic vibrating systems with adaptive hydrodynamic elements are provided and they are used in industrial applications.

Keywords: dynamic system, vibrating system, adaptive hydrodynamic elements, bearings, clearance, frequency of rotation, rotor, effect of temperature.

Introduction

Hydrodynamic bearings are among the most important dynamical systems of different mechanical devices with rotary engines. They are widely used in various mechanical vibrating systems, in dynamic technological machines, high power steam turbines, turbo generators, turbo compressors, pumps, generators, etc. Sliding bearing failures account for 23 % of all causes of system failure according to EEI (Edison Electrical Institute) statistics data. Similar thermal power studies performed in EPRI (Electrical Power Research Institute) have shown that the most common failures occur due to problems caused by such bearings [1, 2].

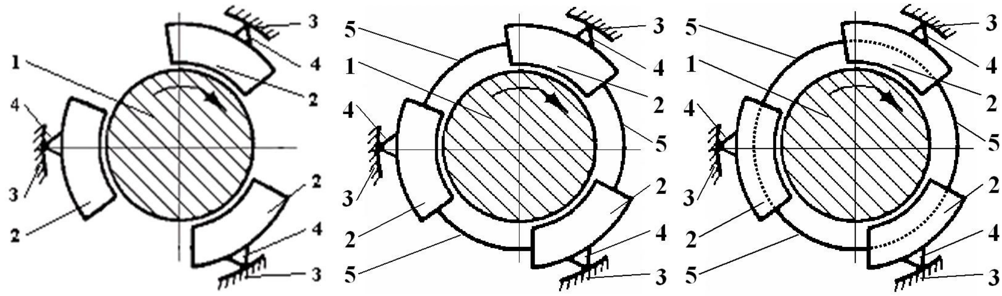
Research work in the field of diagnosis, development and analysis of mechanical dynamic rotor systems with sliding elements with hydrodynamic friction bearings has been performed much less extensively than for systems with rolling bearings. This is due to many reasons. One of them is that the process of degradation of those bearings is diagnosed with some problems. Waning hydrodynamic sliding friction bearings increase the gap between the rotor and the bearing pin. When the gap is significant, exploiting the system forward, operation of the oil wedge is snarled, this increases friction, effect of vibration, temperature, and also other parameters of the system are changed [3, 4, 5].

In particular conditions of operation the temperature in the zone of bearing operation reaches critical values, then the viscosity of lubricating film and the thickness of oil film are reduced [6, 7, 8]. After this the semi-bearing fluid lubrication regime takes place. As a result, this shortens the operating time of equipment and may result in a crash. In such events the process of operation of the system is affected and substantial losses are experienced. In order to avoid large damages, tests in rotating systems with adaptive hydrodynamic bearings with sliding friction have been performed [9, 10].

Object of research

Dynamic parameters are among the most informative parameters characterizing not only the individual unit of the device, but also the state of the whole system. Changing internal and external effects of the rotor system, the dynamic characteristics of the system, levels of vibrations and of temperature are changed. As a result the dynamic operating loads of systems are acting to the individual elements and create defects which may develop into serious faults.

Three designs of investigative adaptive hydrodynamic segmental bearings are described in Fig. 1: adaptive hydrodynamic segmental bearing without connecting segments of the elements (Fig. 1a), adaptive hydrodynamic segmental bearing with individual elastic strips that are connecting the segments (Fig. 1b) and adaptive hydrodynamic segmental bearings with the elastic ring that is connecting the segments (Fig. 1c).



a) b) c)

Fig. 1. Adaptive hydrodynamic segmental bearing: a) without connecting segments of the elements (1 – rotor, 2 – segment, 3 – housing, 4 – adaptive support), b) with individual elastic strips that are connecting the segments (1 – rotor, 2 – segment, 3 – housing, 4 – adaptive restraint, 5 – strips connecting the segments), c) with the elastic ring that is connecting the segments (1 – rotor, 2 – segment, 3 – housing, 4 – adaptive restraint, 5 – strips connecting the segments)

The proposed experimental setup

Dynamic mechanical systems with adaptive segmental hydrodynamic elements are used in the experimental tests performed on the proposed and developed experimental setup (see Fig. 2).

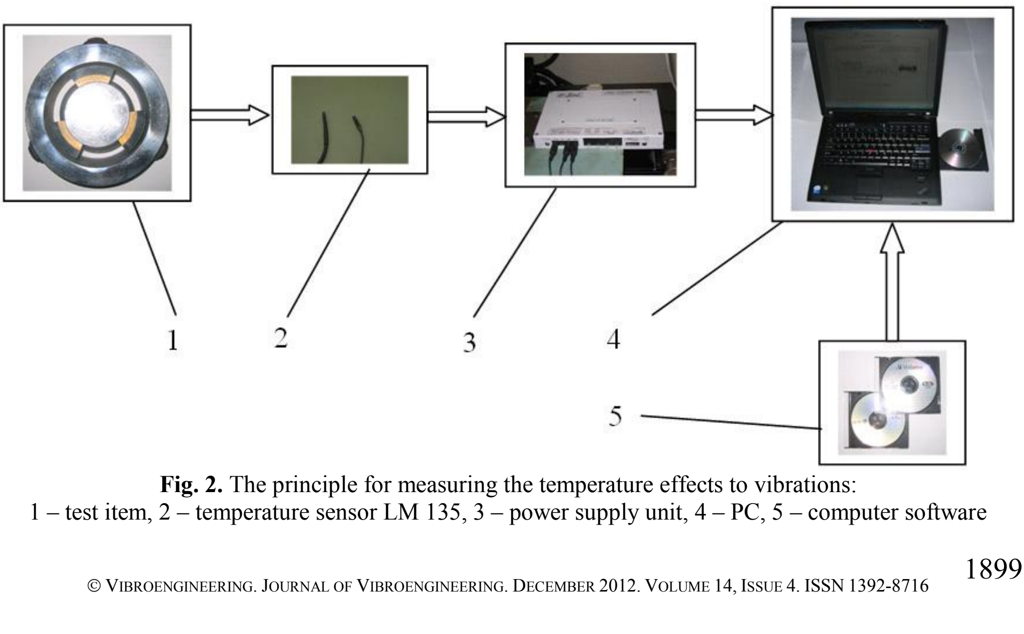
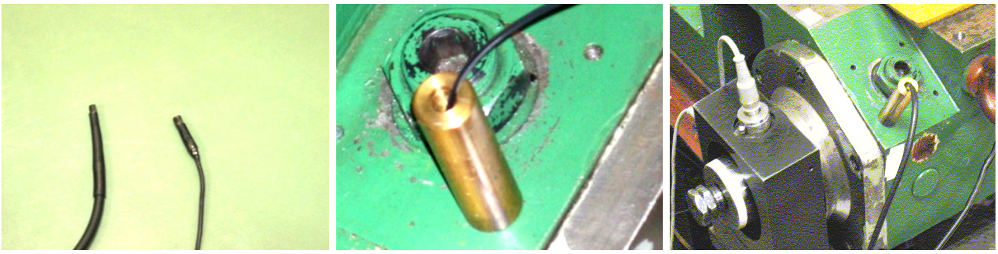


Fig. 2. The principle for measuring the temperature effects to vibrations:
1 – test item, 2 – temperature sensor LM 135, 3 – power supply unit, 4 – PC, 5 – computer software

The main components of the proposed and developed measurement and analysis system are: power supply unit, special THOMSON Microelectronics SGS LM 135 temperature sensors and a computer. Temperature sensors are fed with special power supply unit. The photos with temperature sensors LM 135 and the mounting stand are given in Fig. 3.



a) b) c)

Fig. 3. Elements of experimental setup: a) temperature sensors LM 135, b) place of the thermometer assembly, c) the mounting of thermometer in the stand

Data of the proposed experimental plan for the investigation of the rotor system

Non linear plan of second rank is used to estimate temperature dependence of rotor system as a function of time of operation of the rotor system and of the frequency of rotation of the rotor. It is not sufficient to use the plan of the first rank in order to describe the searching procedure. Primary data of the experimental plan are given in Table 1.

Table 1. Primary data of the experimental plan

Number of factors	Square of span α^2	Distribution of experiments			Number of experiments No.
		At the top of a cube	Points of span h_α	In the center of plan	
2	2	4	4	5	13

Primary data of experiment planning are determination of the size of square of spans and the points of span, which are calculated by using the equations (1), (2), (3), (4) [9, 10, 11].

Methodology of construction of the experimental plan for the rotor system

Experimental plan is determined by using the following methodology:

1. Primary data of the experiment plan are obtained (number of factors, square of span, distribution of experiments, number of experiments);
2. Factors are encoded;
3. Size of the span square is calculated;
4. Points of span are calculated;
5. Level of factors is estimated;
6. Interval of variation is estimated;
7. Bottom and top levels are estimated;
8. Matrix of planning encoding real values is obtained and lines of variation are estimated.

Experimental plan for the investigation of the rotor system

Plan of experiment is obtained on the basis of the planning methodology of experiment [11, 12].

Factors are encoded by a formula using polynomial:

$$X_i = \frac{x_i - x_{i0}}{\Delta x_i}, \tag{1}$$

where X_i – code meaning of the factor, x_i – the natural meaning of factor *i*, x_{i0} – primary (zero) factor, Δx_i – the interval of variation factor *i*:

$$\Delta x_i = \frac{x_i \max - x_i \min}{2}. \tag{2}$$

Size of the span square is calculated by the formula:

$$\alpha = 2^{\frac{k}{4}}, \tag{3}$$

where *k* is the number of factors.

Points of span are calculated by the following formula:

$$X_\alpha = X_0 \pm \Delta X * 1.41, \tag{4}$$

where X_0 denotes prime of level, ΔX is prime of variation.

13 experimental plans of measurements are used for performing experimental investigations: 8 experimental points are in the range of octagon tops and 5 points are in the centers.

Experiments were done in pseudo random order in order to avoid systematic errors. Two factors of variations are chosen for investigation: time of operation till failure of the rotor system *t* (min) and frequency of revolution of the rotor *V* (rev/min). Clearance between the rotor and the bearing of sliding is assumed constant and equal to *h* = 50 μm.

Factors of experiments and its levels of variation are given in Table 2.

Table 2. Factors of experiments and its primes of variation

Factors and interval of variation	Code meanings		Real meanings	
	X_1	X_2	*t* (min)	*V* (rev/min)
Main level	0	0	30	3000
Interval of variation	1	1	15	2000
Top level	+1	+1	45	4000
Bottom level	–1	–1	15	1000
Points of span: –	–1.41	–1.41	9	200
+	+1.41	+1.41	51	5800

Matrix of plan encoding real values is obtained and lines of variation are estimated [11, 12]. Matrix of the plan is given in Table 3.

Results of investigations

Bearings of all types and structures are heated during the process of operation of mechanical devices. Changes of temperature change parameters of the bearings. The implication is that primary parameters of bearings are substantially influenced. Temperature of bearing is equal to the temperature of the environment in the beginning of operation of the device. Temperature begins to increase and the higher is the frequency of rotation of the rotor, the increase of temperature is greater. Oil can be supplied to the bearings for decreasing of the temperature, also bearings can be cooled additionally, but this problem is still not fully solved.

Table 3. Matrix of the plan

Number of range	Matrix of the plan			Real meanings	
	X_0	X_1	X_2	t (min)	V (rev/min)
1.	+1	–1	–1	15	1000
2.	+1	+1	–1	45	1000
3.	+1	–1	+1	15	5000
4.	+1	+1	+1	45	5000
5.	+1	–1.41	0	9	3000
6.	+1	+1.41	0	51	3000
7.	+1	0	–1.41	30	200
8.	+1	0	+1.41	30	5800
9.	+1	0	0	30	3000
10.	+1	0	0	30	3000
11.	+1	0	0	30	3000
12.	+1	0	0	30	3000
13.	+1	0	0	30	3000

It is very important to know the dependence of time of operation of the bearing until failure from the change of temperature. Investigations were performed, trying to determine how changes of temperature influence the parameters of investigated bearings.

The results of investigations are given. They determine temperature levels of the rotor system. It was determined how the size of the gap between the rotor and the bearing is changed because of variations in temperature, the frequency of vibrations and the frequency of rotation of the rotor (Fig. 4, 5). Results for adaptive hydrodynamic bearings with strips that connect the segments (Fig. 1b) are presented. The results for other bearings of similar structures are very similar and differ very little. Thus they are not presented in this paper.

Dependencies of temperature changes from rotation frequencies of the rotor and from operating times are presented in Fig. 4. The primary temperature is temperature of the room 18 ^{0}C. After 60 minutes of operation the rotor was turning at 500 rev/min, the temperature increased only up to 27 ^{0}C. When the rotor was turning at 2000 rev/min, the temperature over the same period of time nearly doubled and reached 34 ^{0}C. The dependencies of the gap between the rotor and the segments of the bearing on the values of the frequency of rotation of the rotor and on the changes of temperature are shown in Fig. 5. The primary gaps between the rotor and the bearing in horizontal and vertical directions are different, because gravity operates on the bearing in the vertical direction. Then while increasing the temperature, the gap is decreasing for both directions.

For the rotor rotating for one hour at the speed of 2000 rev/min, temperature of the system increases up to 40 ^{0}C, and after this the temperature does not increase further. At this temperature the gap between the rotor and the bearing is bigger than a 10 μm threshold. The grease "Velosit 5" was used for lubrication of hydrodynamic bearings at 40 ^{0}C, with viscosity equal to 4.1 – 5.1 mm^2/s and with density equal to 0.850 g/cm^3. It is recommended to use grease of higher viscosity and density "Velosit 10" or "Velosit 22" for the rotor rotating at a speed higher than 2000 rev/min.

Increase of temperature is decreasing the size of the gap which is between the rotor and the bearing. This can be seen from the experimental results. When the rotation frequency of the rotor is higher, the bearing is heating up more, but the size of the gap between the rotor and the bearing does not have a direct dependence on the change of the rotor speed.

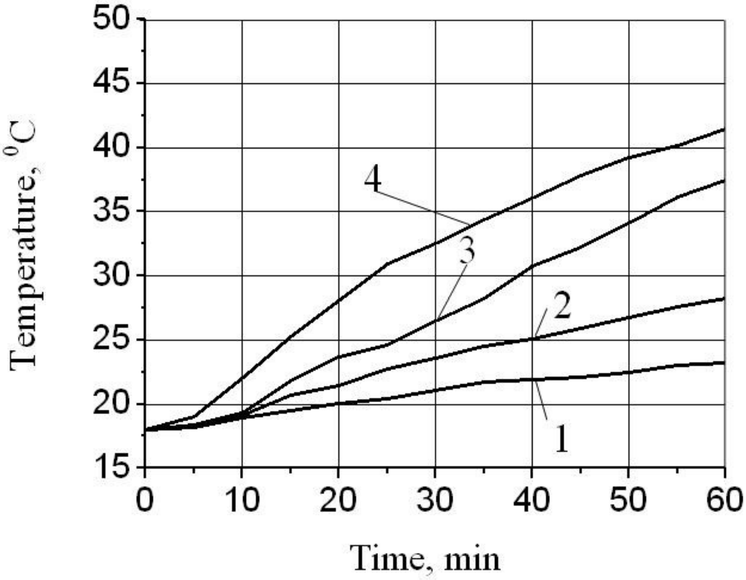


Fig. 4. Dependence of the temperature change on the operating time till failure of the rotor (1 – rotor revolution 500 rev/min, 2 – rotor revolution 1000 rev/min, 3 – rotor revolution 1500 rev/min, 4 – rotor revolution 2000 rev/min)

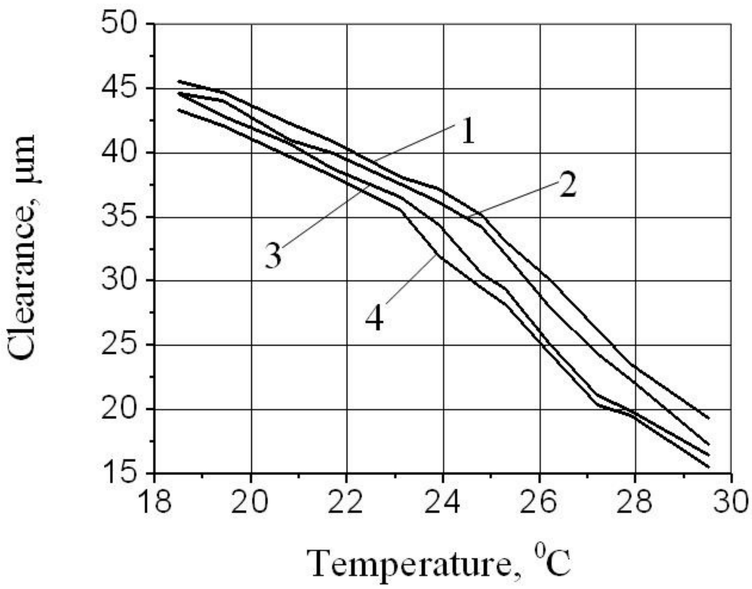


Fig. 5. Dependence of the gap between the rotor and the bearing on the change of temperature, when the rotor is rotated at a frequency of 1000 rev/min (1 – measurement results in the horizontal direction, 2 – measurement results in the vertical direction) and when the rotor is rotated at a frequency of 2000 rev/min (3 – measurement results in the horizontal direction, 4 – measurement results in the vertical direction)

Conclusions

Performed investigations of mechanical dynamic systems with adaptive elements and the obtained results indicate that only complex assessment and analysis of factors that operate on the system (such as temperature, vibration, force, hydrodynamic effects) enable to determine the instantaneous state of the investigated system accurately and to predict breakdowns and failures.

When the frequency of rotation of the rotor was doubled (from 1000 rev/min to 2000 rev/min), the temperature of the bearing increased approximately 1.5-fold during the same time period.

There is very small quantity of investigations in which vibration activity of adaptive hydrodynamic segmental bearings with sectional segments having connecting strips is compared with the other structures of investigated adaptive hydrodynamic segmental bearings.

Increase of temperature is decreasing the size of the gap which is between the rotor and the bearing. When the frequency of rotation of the rotor is higher, the bearing is heating up more, but the size of the gap between the rotor and the bearing does not have a direct dependence on the change of the rotor speed.

The results obtained in the paper find application in the design of hydrodynamic bearings for industrial devices.

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