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DETERMINATION OF RIGIDITY AND DAMPING PARAMETERS OF HYDRODYNAMIC BEARINGS

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Abstract: *The article deals with the problems of identify of rigidity parameters k and damping parameters c of bearings of hydrodynamic two-stage mixed flow pumps. Bearings with diameters smaller than those used in reality were tested with the use of the principle of geometric and hydrodynamic similarity. The tests were conducted on a specially constructed test station. Parameters of bearings were identified by the method of amplitude-frequency characteristics.*

Keywords: Mixed flow pumps, Hydrodynamic bearings, Identification of bearing parameters.

1. Introduction

Tests of a dynamic system are carried out in a few steps. A physical model (commonly referred to as a structural model) is developed on the basis of a real object which, simply speaking, reflects a real system and preserves its significant characteristics. It provides the basis for development of a mathematical model describing analytically the phenomena that occur in the tested dynamic system (motion similar to a real system). Solution of motion equations enables exploring free motion, forced motion or stability (Holka, 2011 and Giergiel et al., 1990). It is possible on condition that all parameters of the analyzed physical model are known. Otherwise, it is necessary to carry out tests to identify the object parameters.

One of ten pump aggregates operating in a serial system was tested (Fig. 2) its capacity was $Q = 5000\ m^3/h$ and it consisted of a two-stage mixed flow pump (Fig. 3), electric motor with power $N = 1250$ kW (Fig. 1) and an discharge system. While performing tests it was necessary to identify rigidity parameter k and damping parameter c of a pump hydrodynamic bearings.

Fig. 1: Electric motor of a pump aggregate.

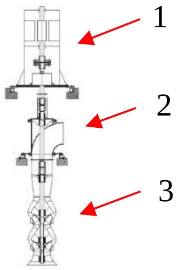

Fig. 2: Scheme of a pump aggregate: 1 – electric motor, 2 – discharge system, 3 – pump.

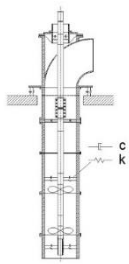

Fig. 3: Structural model of a pump.

There is abundant literature available on the subject of identification. Apart from the above listed publications (Holka, 2011 and Giergiel el al., 1990), works (Nizioł, 2005 and Skalmierski, 2001) are worth attention, as well. Due to the fact that a linear system of equations with constant coefficients was used for the description of the rotor motion, a method based on amplitude-frequency characteristics was used for identification of the model. Acceptance of a linear system with constant coefficients k and c makes calculations less complicated. Bearings are a frequent subject of study. Scientific papers cover different areas of this issue. In this field, positions (Avramov, 2009, Jiang et al., 1999, Laktyushin et al., 2003, Papadopoulos, 2008 and Temis el al., 2007) are considered to be interesting.

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The main goal of this work is to provide methodology and tests results in order to determine parameters of rigidity k and damping c for hydrodynamic bearings of pumps.

2. The research object

Hydrodynamic bearings with internal diameters equal to ϕ150 mm were used in the pump. Carrying out experimental tests of such big bearings would be very difficult, so it was decided to test bearings with three different, smaller diameters (Fig. 4), with application of the principle of geometric and hydrodynamic similarity. Toughness of the tested bearings was consistent with toughness of the real bearing and it was 65 ShA. Dimensions of the tested bearings are presented in Tab. 1.

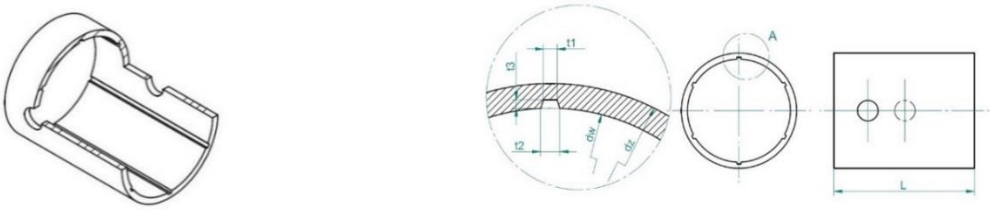

Fig. 4: Tested hydrodynamic bearings.

Apart of bearings with dimensions specified in Tab. 1, also bearings with analogical dimensions, though different toughness, were tested: 52 ShA, 78 ShA. The research was supposed to evaluate the influence of toughness of rubber on the tests results, however this was not included in the presented materials.

Tab. 1: Dimensions of the tested bearings.

Number of bearing	Dimensions in [mm]						Toughness [ShA]
	d_w	d_z	t_1	t_2	t_3	L	
1	30	33.6	1	1.4	0.6	41	65
2	50	56	1.65	2.3	1	683	
3	70	78.4	2.3	3.2	1.4	95.62	

A test station was constructed for the tests (Fig. 5). Its construction was inspired by work (Jiang et al., 2000). The major element of the station was rotor 1 with disc 2. The rotor was fixed in two bearings: self-aligning rolling bearing 3 and the tested hydrodynamic bearing set in a mounting (Fig. 6). Electric motor 5 was used to drive the rotor, whereas water to be used in bearings was pumped by means of a multi-stage mixed flow pump 6. Constructional elements, which are loaded by variable force, were checked for fatigue life by using the method presented in (Strzelecki et al., 2016).

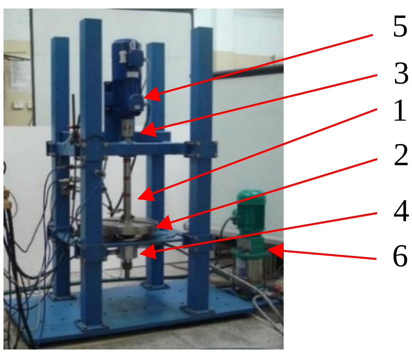

Fig. 5: A test station.

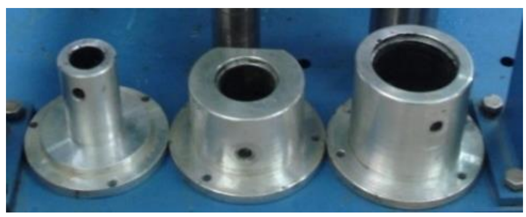

Fig. 6: Tested hydrodynamic bearings with different diameters in mountings.

3. Experimental tests

Bearings with smaller diameters, as compared to real bearings, were tested. It was possible thanks to preservation of constructional similarity, that is, ratio L/d_w, of the journal wrapping angle by a bearing and the Sommerfeld number. At this stage two first conditions were taken into account, whereas the Sommerfeld number is a criterion of hydrodynamic similarity:

$$S = \frac{\eta \cdot n}{p_{śr} \cdot \psi^2} \tag{1}$$

where: η – dynamic viscosity of water [Pa·s], n – the shaft rotational speed [rev/s], $p_{śr}$ – average surface pressure [Pa], ψ – relative clearance in the bearing.

The tests were performed with the use of the most loaded bearing in the pump (value of reaction on the bearing R = 8158.27 N) with dimensions: d_w = 150 mm, L = 220 mm. Substituting data: η = 0.001030499 Pa·s (for 19 °C), n = 740 rev/min = 12.33 rev/s, ψ = 0.00266, $p_{śr}$ = 265309.5 calculated from the dependence:

$$p_{śr} = \frac{R}{L \cdot d_w} \tag{2}$$

The Sommerfeld number was calculated S = 0.006737. To preserve similarity, model bearings should have the same value S.

It was decided that beariungs with diameters ϕ30, ϕ50, ϕ70 mm. would be tested. The lengths of bearings were determined maintaining ratio L/d_w. Average surface pressures (1) $p_{śr}$ were calculated from formula (1) after transformation. Having data of $p_{śr}$ and S, p_{max} was provided from literature. On this basis it was possible to define pressure at which water should be fed to the bearning:

$$p = p_{max} + \rho_w \frac{v^2}{2} \tag{3}$$

where: v – circumferential speed of the shaft [m/s], ρ_w – density of water [kg/m^3].

Whereas, centrifugal force loadning the bearing was calculated from dependence:

$$R = p_{śr} \cdot L \cdot d_w \tag{4}$$

Selected parameters according to which tests were conducted are presented in Tab. 2.

Tab. 2: Presentation of test parameters S = 0.006737.

n rev/ min]	bearing ϕ30 mm			bearing ϕ50 mm			bearing ϕ70 mm		
	$p_{śr}$ [Pa]	p [Pa]	R [N]	$p_{śr}$ [Pa]	p [Pa]	R [N]	$p_{śr}$ [Pa]	p [Pa]	R [N]
200	2868	16862	3,53	7967	46838	27,208	15616	91803	104.52
350	5019	29573	6.17	13943	82147	47.614	27328	161008	182.91

Amplitudes of the shaft relative motion were measured during tests. The results are presented in Fig. 7. Due to high vibrations, higher than rotational speed equal to 950 rev/min, accompanying tests of a bearing with diameter ϕ70 mm, at this speed the measurements were finished.

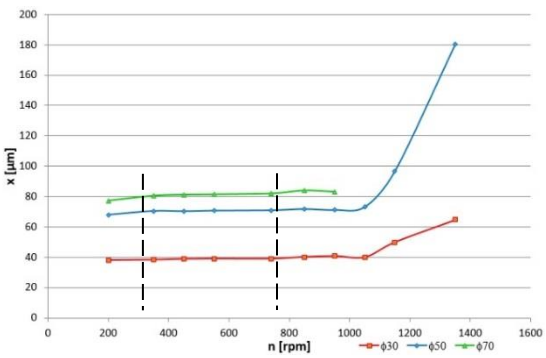

Fig. 7: Measurement of a shaft vibration amplitude for bearings with diameters ϕ30, ϕ50, ϕ70 mm.

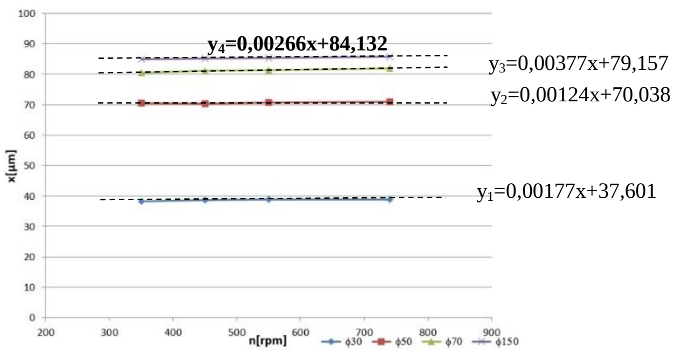

Fig. 8: Equations of straight lines running through two points of the diagrams.

While studying characteristics of the test station (results have not been presented in this work) it was found that the period of its stable operation was within the range of 350 – 740 rev/min, (Fig. 7). Above these speeds resonanse states were observed in two orthogonal directions. For the speeds below

350 rev/min vibrations were also increased. For rotational speeds 350 – 740 rev/min, the diagrams are similar to linear ones. Therefore, this feature was used and straight lines were run through the first and the last points of the diagrams (Fig. 8). An equation of straight line was determined for a bearing with diameter ϕ150 mm (Fig. 8). Coefficients k and c of bearings were determined on the basis of an equation of dynamic susceptibility:

$$\left|\frac{x}{P}\right| = \frac{1}{\sqrt{(k-m\omega^2)^2+(\omega c)^2}} \tag{5}$$

where: m – mass of shaft [kg], P – force of unbalance [N], x – radial displacement of shaft in the bearing [m]. Diagram of dynamic susceptibility for bearing ϕ150 mm is presented in Fig. 9.

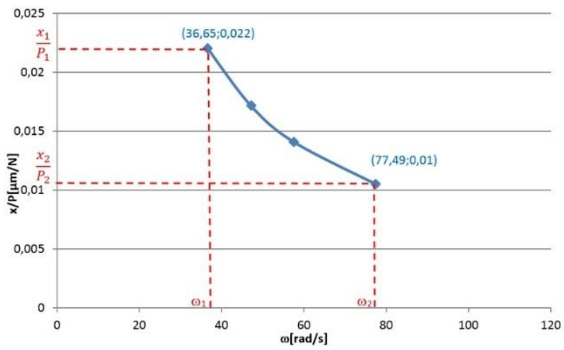

Fig. 9: Diagram of dynamic susceptibility of bearing ϕ150 mm.

By using a linear simplification of the diagram and substituting coordinates of two points a system of equations with unknowns k and c is provided. By substituting data: m = 26.517 kg, x_1 = 0.0000849224 m, P_1 = 3858.64 N, ω_1 = 36.65 rad/s, x_2 = 0.0000858032 m, P_2 = 8158.27 N, ω_2 = 77.49 rad/s, the values of coefficients: k = 7370746.313 N/m, c = 1223436.493 Ns/m were determined.

4. Conclusions

1. In order to identify parameters of bearings k and c a method that involves using amplitude-frequency characteristics was used.
2. Vibrations of bearings are linear for the range of rotational speeds used in the pump.
3. It is planned that further studies of dynamics will take into account the pump mounting, as shown in Fig. 3.

References

Holka, H. (2011) Vibrations and dynamics of machines. Wydawnictwa Uczelniane UTP, Bydgoszcz, Poland, (in Polish).

Giergiel, J. and Uhl, T. (1990) Identification of mechanical systems, PWN, Warszawa, Poland, (in Polish).

Nizioł, J. (2005) Dynamics of mechanical systems, Komitet Mechaniki PAN, Warszawa, Poland, (in Polish).

Skalmierski, B. (2001) Analytical mechanics and vibration theory. T.3, Politechnika Częstochowska, Poland, (in Polish).

Avramov, K.V. (2009) Vibrations of a single-disk rotor on nonlinear supports. International Applied Mechanics, vol. 45, no. 10, pp. 1112-1119.

Jiang, P. L. and Yu, L.(1999) Dynamics of a rotor-bearing system equipped with a hydrodynamic thrust bearing. Journal of Sound and Vibration, vol. 227, no. 4, pp. 833-872.

Laktyushin, A.N., Smilovenko, O.O. and Lakktyushina, T.V. (2003) Dynamics of a rotor in film lubrication bearings. Journal of Engineering Physics and Thermophysics, vol. 76, no. 5, pp. 1131-1138.

Papadopoulos, C.A., Nikolakopoulos, P.G. and Gounaris G.D. (2008) Identification of clearances and stability analysis for a rotor-journal bearing system, Mechanism and Machine Theory, vol. 43, pp. 411-426.

Temis, Yu.M, Temis and M.Yu. (2007) Rigidity and damping characteristics of hydrodynamic sliding bearing with deformable working surfaces. Journal of Friction and Wear, vol. 28, no. 2, pp. 128-138.

Jiang, P.L. and Yu, L. (2000) Identification of the oil-film dynamic coefficients in a rotor-bearing system with a hydrodynamic thrust bearing, Journal of Sound and Vibration, vol. 236, no. 4, pp. 733-740.

Strzelecki, P. and Tomaszewski T. (2016) Aplication of Weibull distribution to describe S-N curve with using small number specimens, AIP Conference Proceedings 1780, vol. 020007, pp. 1-8.