UDC 621.822.6

EXPERIMENTAL RESEARCH OF ROLLING ELEMENT BEARING ARRANGEMENT OF AXIAL-PISTON HYDRO-MACHINE

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The methods were grounded and described as for experimental research of a bearing arrangement of axial-piston hydro-machine, with adducing general results obtained in example of hydro-machine 210.25. Loads acting on bearings' rolling elements have been measured directly at working conditions of the hydro-machine mentioned. The concept offered can be applied when researching rolling element bearing arrangements of other machines and units as well.

Key words: rolling element bearing, hydro-machine, experiment, load distribution between rolling elements, angular contact bearing, radial bearing, oscillogram, strain gauge resistor, recorder, fatigue life.

Introduction. Calculation of loads acting on elements of any bearing arrangement is a basic stage of its analysis, and the more accurate the load distribution between bearings rolling elements is found the more accurate the method of analysis is considered to be. The distribution mentioned can be found with needed accuracy if external forces and moments acting on every bearing in the arrangement are known. However, the principal criterion to serve for their determination reliability has to undoubtedly be connected with experimental measurements made on real machines, especially when the arrangement is not simple and consists of a number of bearings more than that of statically definable system.

The problem. It's impossible to find a bearing life accurately if a load distribution between its elements is unknown, analytic computations sometimes give inaccurate results and need experimental verification though.

Analysis of recent researches and publications. Experimental researches of bearing arrangements can be either bound up with bearings life tests or carried out to find a load distribution between their rolling elements. Life tests conducted at working conditions corresponding to those taking place during real operation take a long time to complete and that's why they are usually replaced by accelerated ones when an arrangement's life is defined at more tough conditions comparatively to those of exploitation. Tests time is shortened by increase of acting forces (working pressure) and the shaft rotation speed. Such an approach in most cases results in wrong estimates of bearings fatigue strength since forced loading conditions are accompanied by additional misalignments and deformations in a bearing arrangement to cause its destruction ahead of time [1]. More informative tests take place when an arrangement is loaded with really acting forces i.e. those in line with a machine running in operation, and at a limited time to carry that out to find a load distribution mentioned. This distribution then gives a possibility to quite accurately estimate the arrangement expected life with given rate of reliability. There are ways to find it based upon a study of imprints left on bearing rings or rolling elements [2, 3]. These methods allow to find a load distribution but exclusively at a fixed shaft so disadvantages here are untaken into account oil films, inertia forces, bearings' permanent being in an oil bath, etc. The SKF bearing company has worked out the method of finding a load distribution by deformations of a bearing outer ring which has short-base strain gauge resistors put in its grooves. The ring has a deformation of tension when rolling element is over the resistor and inversely, is pressed when the element is between the resistors [2]. This way allows to record loads at a revolving shaft but requires many resistors to apply and gives no possibility to simultaneously fix the loads in multi-support bearing arrangements.

Research purpose: By means of direct strain measurement, find actual load distribution in a bearing arrangement of axial-piston hydro-machine.

Research results: To determine loads acting on rolling elements in a bearing arrangement and estimate accuracy of proposed computation method, a research has been carried out with use of experimental plant made on the base of axial-piston hydro-machine 210.25 shown in Fig. 1. The plant gives a possibility to record load oscillograms immediately when the hydro-machine is in operation and consists of the hydro-machine itself and a special device connected to its housing from the side of hydraulic control valve. Loads acting on the rolling elements are measured by short-base strain gauge transducers 1, 2 and 3 connected with movable contacts of current collector 5 by means of wires going through axial holes made in the shaft, central tenon and extension tube. Temperature compensation is put into effect by resistor 4 placed on a plate attached to immovable spacer ring which separates radial bearing from angular contact ones.

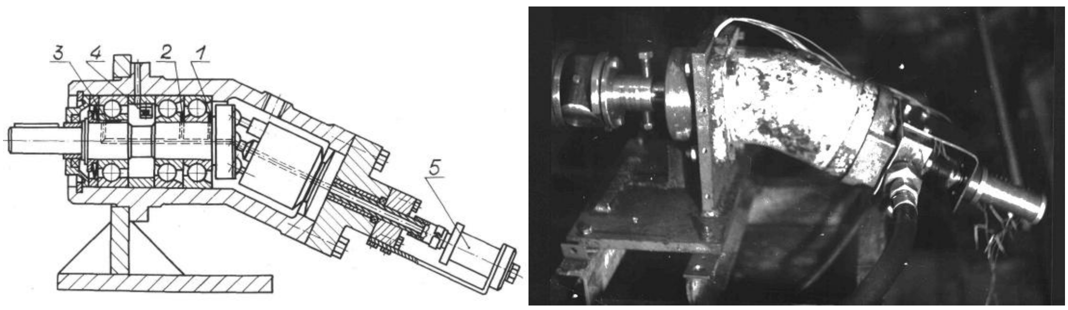

Fig. 1. Experimental plant. 1, 2, 3 – active transducers; 4 – compensation transducer;

5 – mercuric current collector.

Active transducers are disposed in grooves ground on wide butts of angular contact bearings' inner rings (Fig. 2a) and cylindrical surface of radial ball bearing (Fig. 2b). Active transducers for all bearings and the one for thermal compensation are connected within a half-bridge circuit.

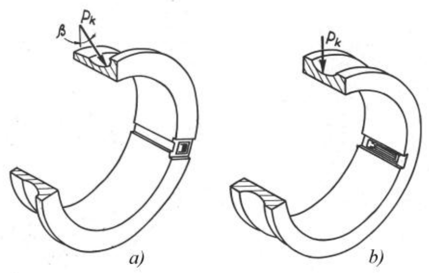

Fig. 2. Location of active transducers.

Signals sent from the transducers were fixed with the use of standard equipment – amplifier 8АНЧ-7М and four-channel fast-acting recorder Н338-4П. Location of active transducers was fixed by one of channels of the latter with connecting it to switch БВК-24 snapping into action every time the active transducers are in upper position. Calibration of active transducers was made when loading those very bearings supposed to then be mounted into the hydro-machine investigated. The transducer disposed in radial bearing was calibrated with one of the balls placed right on the line of radial force applied to secure taking the force by just one ball positioned over the transducer. Calibration of active transducer put in the angular contact bearing was made when loading it with solely axial force and under this conditions every ball took a normal load of

$$P_k = Q / (z_\omega \cdot \sin\beta_k) ,$$

where β_k – contact angle in the bearing.

The research was carried out on the test bench providing the experimental plant to work within conditions typical for hydro-motors – the shaft rotation frequency $n_в$ = 60 rev/min (pumping unit stable rotation lower limit) and pump pressure q = 10 MPa that corresponds to equivalent pressure for hydro-machine 210.25 mounted in a turn gear of excavator ЭО-4121. Oscillogram of loads acting on bearings of pumping unit is shown in Fig. 3. Its peaks fit the instants of the balls' going by the places of the transducers location. As follows from the kinematics, angle distance between the peaks corresponds to the shaft rotation angle

$$\gamma_в = 2\pi \cdot d_k / [z_\omega \cdot (d_k - D_\omega \cdot \cos\beta_k)] ,$$

where d_k , z_ω , D_ω , β_k – bearing middle diameter, number and diameter of rolling elements and contact angle correspondingly. Signal from transducer of rotation angle is shown there as well. Lines rounding curves obtained experimentally are presented in Fig. 4 and describe the load distribution between rolling elements in the arrangement researched. Table 1 is to present bearings radial and axial reactions calculated by realationships [4] :

$$F_{rxm} = \sum_k P_k \cdot \cos\beta_k \cdot \sin\psi_k ; \quad F_{rym} = \sum_k P_k \cdot \cos\beta_k \cdot \cos\psi_k ; \quad F_{am} = \sum_k P_k \cdot \sin\beta_k ;$$

$$F_{rm} = (F_{rxm}^2 + F_{rym}^2)^{1/2}, \quad m = 1, 2, 3.$$

This table also shows direction angles of bearings radial reactions as well as equivalent forces in balls contacts with outer and inner rings calculated basing upon the load distribution found experimentally. Values of bearings expected fatigue life are cited there as well.

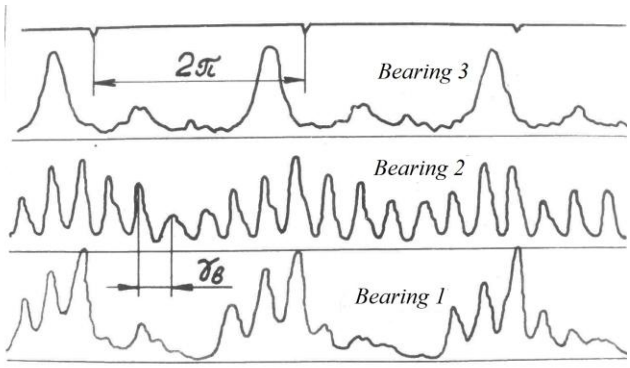

Fig. 3. Bearings loading oscillograms.

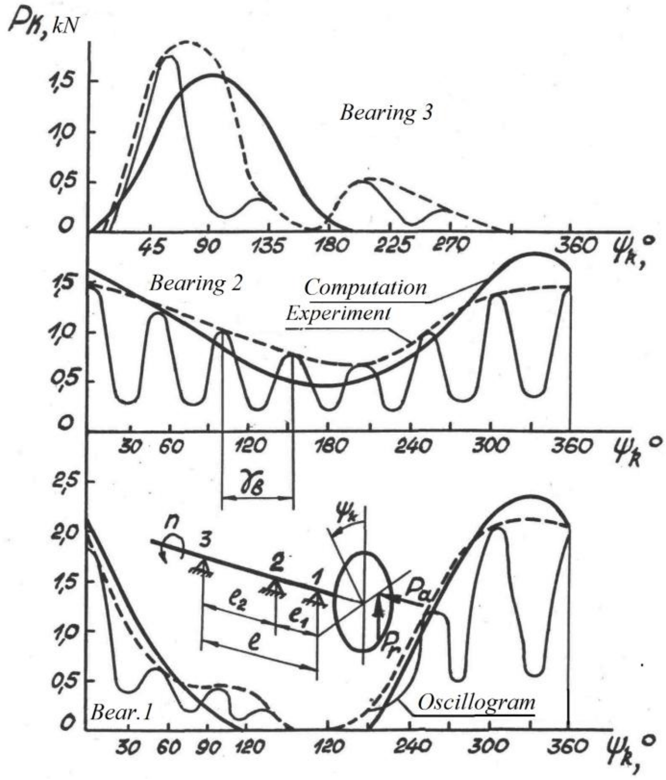

Fig. 4. Bearing arrangement load distribution.

For comparison, values obtained by computation [4] are also shown there. When computing, next initial data were taken into account – external forces at pressure q = 10 MPa are $F_x = 0$, F_y = 7.12 kN, F_z = 15.27 kN, M_x = 245 N·m, M_y = 368 N·m, the shaft cross-section inertia moment I_x = 45 cm^4. Measured clearance between outer rings and the housing Δ_k = 0.02 mm, duplicate clearance t_z = 5 µm. Bearing arrangement computation results are presented in Tables 1, 2 and Fig. 4. Differences between calculated values of maximum loads acting on rolling elements and that found experimentally are 12%, 18% and 20% for bearings 1, 2, 3 accordingly. One is to show it up that the highest difference takes place for bearing 3 which is the least loaded and doesn't determine the arrangement durability as a whole. For most loaded bearing 1 computation gives higher load values to secure some increase of the arrangement safety margin. This is also seen from Fig. 4 – the bearing disposed by the shaft flange has unloaded rolling elements.

Table 1. **Comparison of calculated data with experimental findings for hydro-machine 210.25**

Parameter	Bearing	Experiment	Computation
Radial reaction direction angle $\gamma_m{}^{o}$	m=1	- 57	-61
	m=2	-87	-84
	m=3	192	176
Axial reactions F_{am}, kN	m=1	7.014	7.112
	m=2	8.162	8.211
	m=3	-	-
Radial reactions F_{rm}, kN	m=1	4.672	5.376
	m=2	1.739	2.614
	m=3	2.595	2.842
Equivalent force Q_{eim}, kN	m=1	1.344	1.442
	m=2	1.118	1.197
	m=3	1.069	0.863
Equivalent force Q_{eo}, kN	m=1	1.393	1.491
	m=2	1.128	1.216
	m=3	1.118	0.903
Durability L_{hm}, thous. hrs	m=1	29.3	24.1
	m=2	51.4	42.5
	m=3	65.8	125.3

Table 2. **Misalignments and deformations in hydro-machine 210.25**

Deformations, μm			Misalignments, μrad		Misalignments after shaft bending, μrad		
f_x	f_y	f_z	φ_x	φ_y	$\varphi_{0x1}/\varphi_{0y1}$	$\varphi_{0x2}/\varphi_{0y2}$	$\varphi_{0x3}/\varphi_{0y3}$
5.1	17.6	17.6	259	221	$\frac{30.0}{120.0}$	$\frac{-3.8}{36.0}$	$\frac{-8.6}{-14.0}$

Conclusions: Comparison of experimental results with that of computation has shown good correlation to conclude that assumptions taken as a basis for computation method [4] are correct.

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ЭКСПЕРИМЕНТАЛЬНОЕ ИССЛЕДОВАНИЕ ПОДШИПНИКОВОГО УЗЛА ГИДРОМАШИНЫ

Елизаров С.П., Липин А.П.

Ключевые слова: подшипник качения, гидромашина, эксперимент, распределение нагрузок между телами качения, радиально-упорный подшипник, радиальный подшипник, осциллограмма, тензорезистор, самопишущий прибор, ресурс подшипника.

Резюме

Обоснована и приведена методика экспериментальных исследований подшипникового узла аксиально-поршневой гидромашины, а также приведены основные результаты исследования на примере гидромашины 210.25. Измерены нагрузки в контакте тел качения подшипников с кольцами непосредственно при работе гидромашины. Предложенная методика может быть использована для исследований опорных узлов других машин и агрегатов.

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Summary

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