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VIBRATION-ISOLATING BALL-TYPE CLUTCH AS A SOURCE OF INCREASED VIBRATION ACTIVITY IN THE SYSTEM: AMORTIZED MECHANISM – SHAFT LINE

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***Abstract**: The results of experimental studies of vibration activity of a vibration-insulating ball-type clutch installed on a shaft line are presented. These studies were made on the test stand. It is shown that, depending on the value of the shaft line fracture, the vibration activity of the ball-type clutch can increase in a wide frequency range up to 30 dB. The diagnostic features of the ball-type clutch which allow estimating its vibration activity have been identified on the basis of the analysis of experimental data.*

***Keywords:** diagnostics, vibration, vibration activity, ball-type clutch, vibration-isolating clutch, frequency*

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1. INTRODUCTION

A significant amount of research is devoted to the study of increased vibration of equipment and identification of the causes leading to an increase in vibration activity (VA) of equipment. Thus, in [1, 2, 3, 4], the vibration activity of electrical machines was investigated, in [3, 4, 5], bearing units on rolling and sliding bearings was also studied. The VA of reciprocating compressors were studied in [5], the VA of fans and ventilation systems, etc. were studied in [5, 6].

A large number of papers are devoted to diagnostics of devices and mechanisms using primary and secondary vibration spectra [7, 8, 9, 10, 11, 12, 13]. In [1, 4], the diagnostics of defects in electrical machines and their elements such as magnetic systems [3, 14], bearing assemblies [15, 16, 17, 18, 19] is considered. However, experimental studies of amortized mechanisms with shaft lines on which vibration-isolating ball-type clutches are installed have shown that under certain installation conditions, increased vibration levels on the bearing units of the shaft line attachment are observed.

This article is devoted to an experimental study of the causes that lead to increased vibration activity of the system: amortized mechanism – shaft line with a vibration-isolating ball-type clutch installed on it and is also devoted to identification of diagnostic signs that allow them to be identified.

2. EXPERIMENTAL RESEARCH

The use of vibration-isolating means that prevent the passage of vibration energy from the mechanism to the bearing structure [20] required the use of clutches of various designs, for example, ball-type clutch, which compensate for the fracture and displacement of the shaft relative to the oscillating amortized mechanism.

Structurally, the ball-type clutch is a bushing 2, connected through balls – 1 with two half-clutches – 4. Retaining ring – 3 preventing the balls from falling out of the groove (Fig. 1).

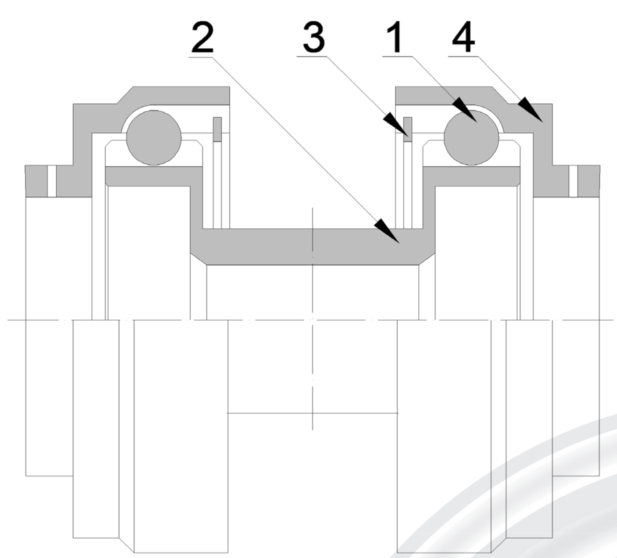

Fig. 1: Ball-type clutch design

Experimental research was carried out on a test stand (Fig. 2) consisting of a base 5 in the form of a cast iron plate installed through shock absorbers 6 on a foundation 7. On the base, four bearings with rolling bearings 1–4 with the possibility of vertical movement are installed. Two half-shafts 8, 9 connected by a ball-type clutch – 12 are installed in the bearings. Rotation of the shaft line is carried out by a direct current motor 11 connected to the half-shaft 9 through a vibration-isolating clutch 10. The vibration signal was recorded by accelerometers 13 mounted on the ends of the bearings 1–3 in the vertical plane.

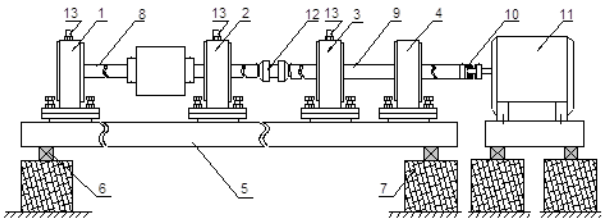

Fig. 2: The test stand design:
1–4 – shaft bearing with rolling bearings; 5 – base; 6 – shock absorbers; 7 – foundation; 8, 9 – half-shaft; 10 – vibration–isolating clutch; 11 – DC motor; 12 – ball-type clutch; 13 – accelerometers

Investigations of the vibration activity of the ball-type clutch were carried out by measuring the vibration levels when the bearings were displaced in the vertical plane. Tab. 1 shows ten options for shifting the bearings.

Figs. 3 and 4 show the spectrograms of normalized vibration levels ΔL_ξ in 1/3-octave frequency bands measured on the second and third bearings of the stand with different options for shifting the bearings. In Figs. 3 and 4, curve I shows the measurement results ΔL_ξ when shaft 2 bearing is displaced by ***h2*** = 3 mm, and shaft 3 bearing is displaced by ***h3*** = 1.5 mm (option 1); curve 2 shows the results when shaft 2 bearing is displaced by ***h2*** = 0.65 mm (option 3); curve 3 shows the results when shaft 1 bearing is displaced by ***h1*** = 0.73 mm, shaft 2 bearing is displaced by ***h2*** = 1.4 mm, shaft 3 bearing is displaced by ***h3*** = 1.08 mm (option 7); curve 4 shows the results when shaft 2 bearing is displaced by ***h2*** = 1.39 mm (option 9); curve 5 shows the results when shaft 2 bearing is displaced by ***h2*** = 3 mm (option 10). The vibration levels ΔL_ξ shown in the spectrograms (Figs. 3 and 4) are normalized to the vibration levels measured in the absence of displacement of the bearings (displacement and fracture of the elements of the clutch $\gamma = 0$, $\psi = 0$, see Fig. 5).

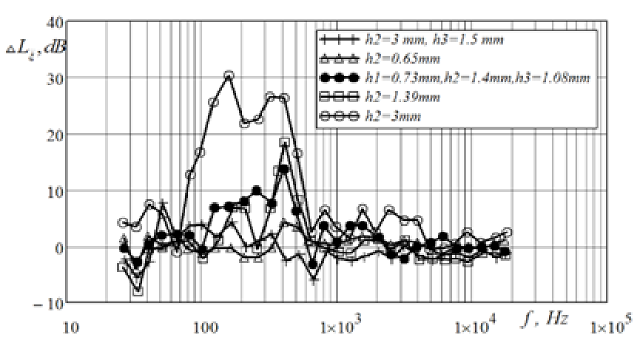

Fig. 3: Normalized vibration levels on bearing 2 of the stand in case of the shaft line fracture

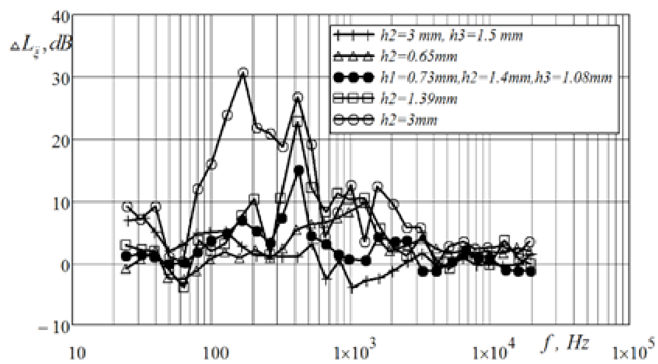

Fig. 4: Normalized vibration levels on bearing 3 of the stand in case of the shaft line fracture

Figs. 3 and 4 show that with some variants of displacement of bearings, a significant increase in vibration levels (up to 20–30 dB) in the frequency range 80–500 Hz takes place.

Two reasons leading to this effect are possible:
– fracture and displacement of the ball-type clutch elements;
– displacement of elements of rolling bearings installed in bearings.

Let's consider both options.

Option 1. The source of vibration activity increased is a ball-type clutch.

Fracture and relative displacement of the half-shafts rigidly connected with the half- clutches by a ball-type clutch is compensated for by the corresponding displacement of the bushing and the clutch balls. When the shaft assembled and installed without skewing rotates, the driving forces acting in the clutch are small. In this case, the clutch does not affect the overall vibration activity of the operating system: the mechanism is a shaft line with a ball-type clutch. When one or two half-shafts are skewed, the elements of the ball-type clutch are mutually displaced relative to each other compensating for the shaft fracture. In this case, the planes of the balls in contact with each of the two half-clutches form angles in relation to the diametric plane of the half-clutches γ to ψ (on the left and right half-clutches respectively) (Figs. 5 and 6). In one revolution of the shaft, each ball in the clutch produces a reciprocating movement along the axis of the bushing along the recess in the half-clutch, equal to $l_1 = (r+d)\mathrm{tg}(\gamma)$ or $l_2 = (r+d)\mathrm{tg}(\psi)$, where r is the radius of the bushing, d is the diameter of the ball (Fig. 5). It can be seen that the greater the skewness of the clutch elements, the greater the values of l_1 and l_2. In its turn, this leads to an increase in slippage speed of the ball along the recess in the half-clutch: $V_{1,2} = l_{1,2}\, n/30$ m/s, where ***n*** is the number of shaft revolutions per minute. An increase in the skewness angles leads to an increase in the ball pressure on the recess in the half-clutch. The action of these factors contributes to a sharp increase in the friction forces in the clutch, i.e. in case of fractures and displacements, the clutch can be an additional source that increases the vibration activity of the operating system: mechanism-shaft line with a ball clutch.

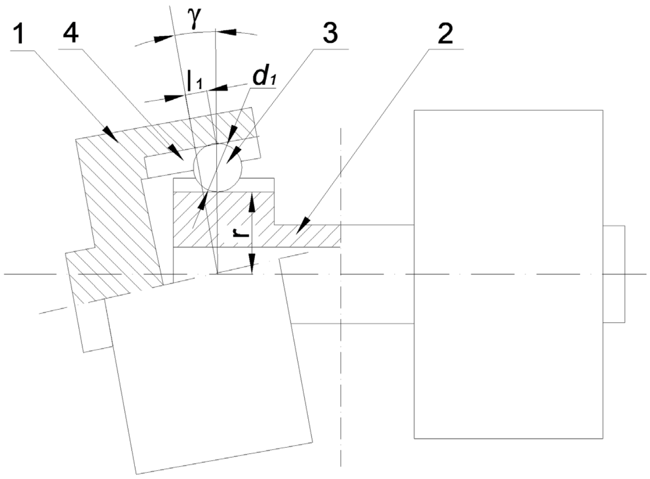

Fig. 5: A ball-type clutch with a fracture of one half-clutch: 1 – half-clutch; 2 – bushing; 3 – ball; 4 – recess

Thus, the value $\delta = \gamma + \psi$ can serve as a generalized characteristic of the mutual displacement of the ball-type clutch elements. Tab. 1 shows a comparison of the total fracture angles of the elements of the clutch $\delta = \gamma + \psi$ and the displacement of the bearings.

Analysis of the spectrogram shows that with an increase in the total fracture angle of the elements of the ball-type clutch δ (Figs. 5, 6), an increase in three-octave vibration levels is observed in the middle frequency range. In a wide frequency range, a significant increase in vibration levels (curve 5 in Figs. 3 and 4) is observed when the total fracture angle of the elements of the clutch δ approaches the critical value δ_{cr}. The value of the critical fracture angle is determined by the angles γ_{cr} and ψ_{cr} which are the maximum rotation angles of the half-clutches relative to the bushing. Calculations show that for a given clutch design, the angles are $\gamma_{cr} = \psi_{cr} = 6 \cdot 10^{-2}$ rad.

When the angles δ of the clutch are close to the maximum, slippage of the balls along the recess in the half-clutch can be accompanied by the phenomenon of self-oscillations, which are known to have subharmonic character and sharply expand the frequency spectrum and amplitude of the driving force in the ball-type clutch elements An increase in friction forces in the ball-type clutch elements, which leads to an increase in vibration of the stand bearings is accompanied by a significant increase in the temperature in the rubbing elements.

During the operation of the stand in option 9 $\delta = 5.65 \cdot 10^{-2}$ rad, the temperature of the ball-type clutch increased to 30 °C, and in variant 10 $\delta = 11.58 \cdot 10^{-2}$ rad, the temperature increased to 80 °C. Once again, this confirms that with significant skewnesses in the clutch elements, large values of driving forces arise due to the friction of the clutch elements.

The reciprocating motion of the ball along the recess in the half-clutch occurs with a period equal to one revolution of the shaft. Therefore, the spectrum of the driving forces acting in the clutch has to contain a component with the frequency $f_{bc} = z \cdot f_{fr}$, where f_{fr} is the rotating frequency of the shaft (Hz), z is the number of balls in one half-clutch. At this frequency, the amplitude of the vibration signal measured on the bearing should increase with the growth of the total angle of skewness of the clutch elements δ.

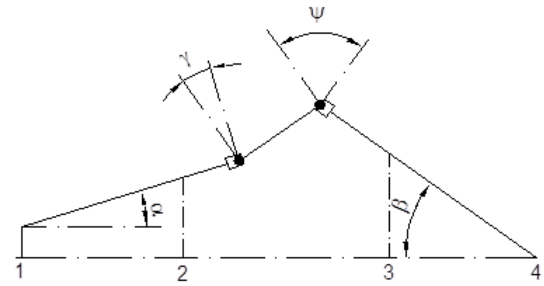

Fig. 6: Scheme of the mutual arrangement of the shaft elements with a ball-type clutch in case of fracture; 1, 2, 3, 4 – bearings of half-shafts

Option No	Displacement of bearings [mm] (Bearing number – displacement)	$\delta \cdot 10^2$ [rad]	$\delta_1 \cdot 10^3$ [rad]
1	I – 3, III – 1.5	1.35	13.55
2	I – 0.68, II – 0.65	1.94	0.06
3	II – 0.65	2.65	1.35
4	II – 0.7, III – 0.73	3.54	5.55
5	II – 1.4, III – 1.08	3.75	8.99
6	I – 1.42, II – 1.39	4.18	0.06
7	I – 0.73, II – 1.4, III – 1.08	4.52	7.46
8	I – 0.68, II – 1.39	4.95	1.48
9	II – 1.39	5.65	2.9
10	II – 3	11.58	6.25

Tab. 1: Connection of test modes with the total fracture angle of the elements of the ball-type clutch $\delta = \gamma + \psi$ and the total fracture angle of the half-shafts $\delta_1 = \alpha + \beta$

Fig. 7 shows the dependences of the change in the amplitude of the vibration signal measured at the frequency f_{bc}, depending on the normalized total skewness angle of the elements of the ball-type clutch δ/δ_{cr} obtained from the results of 10 tests (see Tab. 1). Points on curve 1 correspond to the levels of discrete vibration components at the frequency f_{bc} on the second shaft line bearing, and on curve 2 – on the first shaft line bearing. As it can be seen from Fig. 7, an increase in the total fracture angle of the ball-type clutch elements leads to an increase in the level of the discrete component of vibration at the frequency f_{bc}. At normalized angles of skewness of the elements of the ball-type clutch $\delta/\delta_{cr}<0{,}35$, this increase is hardly noticeable, but at $\delta/\delta_{cr}>0{,}35$, the levels of the discrete component increase significantly. Comparison of the spectrograms in Figs. 3 and 4 with the curves in Fig. 7 shows that the minimum vibration levels measured in the options for installing bearings 1–5 (see Tab. 1) correspond to the normalized angles δ/δ_{cr} lying in the range 0–0.35. At the same time, one-third-octave vibration spectra practically do not exceed the results of measurements of vibration levels obtained at zero ($\gamma=0$, $\psi=0$) skewness angles of the elements of the ball-type clutch.

Vibration levels measured in the options for installing supports 9, 10 at normalized angles δ/δ_{cr} lying in the range $\delta/\delta_{cr}>0{,}35$. These curves are characterized by an increase in one-third octave vibration levels in the middle frequency range. It should also be noted that the vibration levels measured on the first bearing of the stand (Fig. 7) repeat the nature of the change depending on δ/δ_{cr}, but by about 10 dB less, which indicates that the source of vibration activity is between supports 2 and 3.

Thus, it can be concluded that at the frequency f_{bc}, the change of the level of the discrete component of vibration can be uniquely associated with a change in one-third octave vibration levels in the middle frequency range, and this change has an undoubted connection with a change in the total angles δ and can be used as a diagnostic sign of improper functioning of the ball-type clutch.

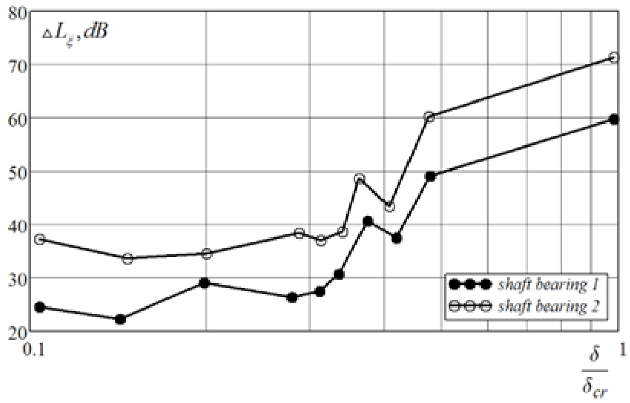

Fig. 7: At frequency f_{bc}, dependence of the vibration signal amplitude on the normalized total angle of fracture of the clutch elements

Option 2. A source of increased vibration activity – ball bearings in the shaft line bearings.

In principle, it is possible that an increase in the vibration activity of the system mechanism-shaft line with a ball-type clutch is associated with the mutual skewness of the elements of ball bearings installed in the shaft line bearings.

In this case, the source of the driving force is located in the support and not in the ball-type clutch. An increase in the fracture of half-shafts, which can be characterized by the total angle of fracture $\delta_1 = \alpha + \beta$ should be accompanied by an increase in the level of the measured vibration signal at the frequency f_{bc}.

Fig. 8 shows the function dependency . From the graph in Fig. 8, it is impossible to identify any regularity that determines the relationship between the angles and δ_1.

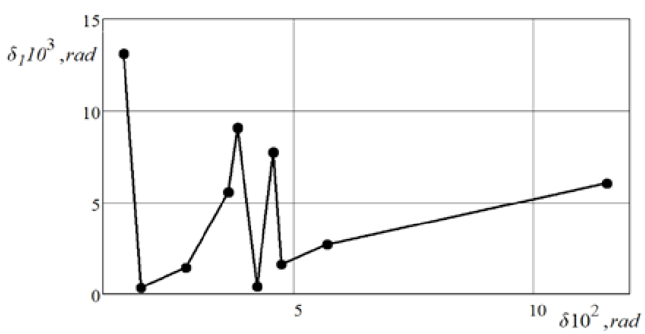

*Fig. 8: Relationship of changes in the total fracture angles of the elements of the clutch – **δ** and the half-shafts δ_1 obtained during experimental studies on the stand*

Fig. 9 presents the dependences showing the change in the vibration level for the discrete component at the frequency from the angle δ_1 that determines the fracture of the shaft line ($\delta_1 = \alpha + \beta$) for the first bearing (curve 1) and the second bearing (curve 2).

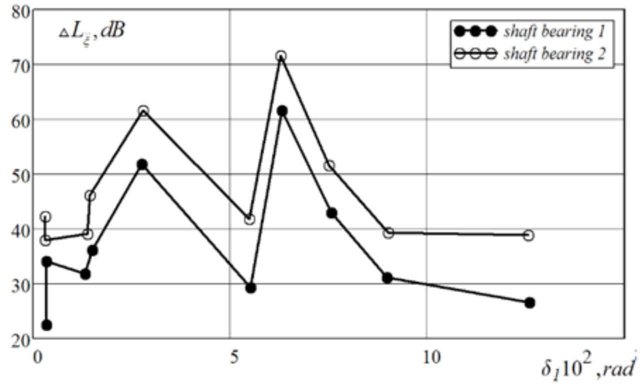

Fig. 9: Dependence of the vibration signal amplitude on f_{bc} on the total angle of the shaft fracture δ_1

As it can be seen from the graphs, any pronounced dependence of the vibration level on the angle δ_1 is absent. This position indicates the absence of a certain dependence of the discrete component at the frequency f_{bc} on the total skewness angle of the shafting line δ_1. The maximum angle δ_1 corresponds to the actual minimum value of the discrete component of the vibration level at the frequency f_{bc}. The maxima of the amplitude correspond to the angles located at the beginning and in the middle of the change interval of the angle δ_1 ($\delta_1 = 2.9 \cdot 10^{-3}$ rad; $\delta_1 = 6.25 \cdot 10^{-3}$ rad). In addition, if the vibration source is a bearing, the frequency should be present in the envelope of the high-frequency spectrum of the vibration signal [9, 10, 19]. However, the analysis of the frequency spectrum of the envelope shows the practical absence of a discrete component with this frequency. In addition, the vibration levels measured on bearing 1 at the frequency f_{bc} is about 10 dB less for all angles δ_1 than those measured on bearing 2 although the mutual skewness of the elements in the rolling bearings in supports 1 and 2 is the same. Consequently, the source of the driving force is the clutch, and the increase in vibration levels in the one-third-octave spectra at medium frequencies is associated with the mode of operation of the ball clutch.

3. CONCLUSION

The ball clutch can be a source of increased vibration activity over a wide frequency range. In this case, one-third-octave vibration levels can increase up to 20–30 dB in the medium frequency range 80–500 Hz.

It is necessary to diagnose the vibration activity of the ball-type clutch and, therefore, the correctness of its functioning by registering the amplitude of the vibration signal at the frequency f_{bc} measured at the shaft bearing closest to the clutch.

When designing the system: an amortized mechanism-a shaft line with a ball-type clutch, it is necessary to prevent the maximum skewness of the elements of the ball-type clutch more than $\delta > 0.35 \cdot \delta_{cr}$ during its operation.

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