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DIAGNOSTICS OF ROLLING BEARINGS OF MACHINES FOR EXTRACTION AND PROCESSION OF ROCK MATERIALS

Viara Pojidaeva

University of Mining and Geology
St. Iv. Rilski
Sofia 1700, Bulgaria

RESUME
The article deals with the matter of rolling bearings wearing effect, expressed by increase of their radial clearance, on the dynamic characteristics of diamond discs of disc and cutting machines for processing of lining materials of rock origin. A method is proposed for establishing the size of the radial clearance by means of non-dismantling control of vibrations.

The main problem in the maintenance of machines for extraction and processing of lining materials of rock origin is how to control the condition of the separate units and architectonic components during the period of operation. This control is directly relating to the quality of the products and is implemented by means of labor and time consuming mounting and dismounting operations.

The methods of technical diagnostics, in particular the non-dismantling control of vibrations, are a means of making the maintenance of machines in this industry more precise and up-to-date. The machines themselves react to defects and wearing of their components and units, bad assembly, etc. by changing the conduct of their vibrations. The implementation of the non-dismantling control of vibrations requires normalizing of the vibration indicators (by frequency and amplitude) beforehand appropriate to the design characteristics of the particular machine.

In the particular case is treated the matter of the rolling bearings wearing effect (increase of the radial clearance) on the vibration characteristics of the diamond disc of disc and cutting machines. As it is known, the radial clearance in bearings is established by dynamic eccentricity, which is equivalent to unbalance [1,2]. If we assume that the disc is perfectly balanced, then the radial clearance in the bearing preconditions the shift of its geometric center from the theoretical rotation axis (Fig.1).

The following assumption were made in research of this problem: the shaft is non-deforming, its mass is insignificantly small as compared with the disc, the bearing shields are perfectly hard in radial direction and play the function of vibro-acoustic bridges, the damping in the system is ignored. The shaft and the disc are coaxial and rotate at one and the same angular speed. The shift of the geometrical center of the disc from the theoretical rotation axis in result of the radial clearance in the bearing creates a centrifugal force, which could be presented by means a vertical component F and moment M [4] (Fig.1).

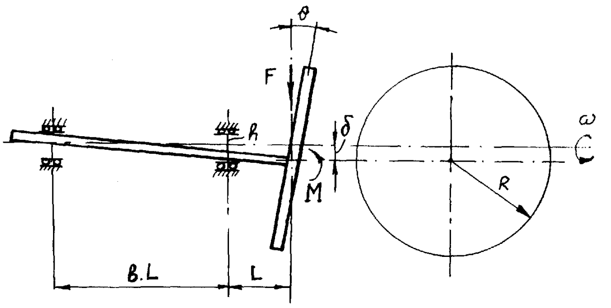

Figure 1.

$$\begin{cases} F = m\omega^2\delta \\ M = I_g\omega^2\theta \end{cases} \qquad (1)$$

where:

- m - mass of the disc;
- ω - angular speed;
- $I_g = m\frac{R^2}{4}$ - radius of the disc inertia towards its diameter
- δ and θ - parameters, which determine the shift of the geometrical center of the disc towards the rotation axis.

Using the method of the influence factors and the Maxwell's theorem of reciprocity, δ and θ are as follows:

$$\begin{cases} \delta = F.\delta_F + M\delta_M \\ \theta = F.\theta_F + M\theta_M \end{cases} \qquad (2)$$

where: $\delta_F, \delta_M, \theta_F, \theta_M$ are influence factors reflecting the linear and the angular shift of the shaft axis from the theoretical rotation axis as follows:

$$\delta_F = \left(\frac{L^3}{3EI}\right)(1+b)$$

Shift of the shaft axis caused by a force unit applied.

$$\delta_M = \left(\frac{L^2}{6EI}\right)(3+2b)$$

Shift of the shaft axis caused by a momentum unit applied.

$$\theta_F = \left(\frac{L^2}{6EI}\right)(3+2b)$$

Angle of shaft inclination caused by a force unit applied $(\delta_M = \theta_F)$

$$\theta_M = \left(\frac{L}{3EI}\right)(3+b)$$

Angle of shaft inclination caused by a momentum unit applied.

After substitution of the influence factors and transformation of equations (2), for the frequency equation of the system is obtained:

$$\left|\begin{array}{l}\left(1-m\omega^2\delta_F\right)\delta+\left(I_g\omega^2\delta_M\right)\theta=0\\ \left(m\omega^2\theta_F\right)\delta-\left(1+I_g\omega^2\theta_M\right)\theta=0\end{array}\right. \qquad (3)$$

From equations (3) can be established the frequency and the amplitude of the vibrations due to radial clearance in the system and its size. After simple geometrical transformations is obtained the size of the radial clearance h:

$$h \approx 2(\delta - tg\theta) \qquad (4)$$

Formula (4) is an approximate solution of the radial clearance size because of the assumptions made to simplify the analysis. It can be used with sufficient practical precision only with respect to discs, which are designed in accordance with the diagram on Fig.1 and operate within the optimum range of cutting speeds.

The adopted approach to analyzing the frequency characteristics of the system as a function of the radial clearance well illustrates the necessity for setting standards of the vibration indicators appropriate to the structure of the particular machine and its specific features such as: balance quality, dynamic stresses, deformation and strength characteristics, stability, suspension type, pliability of the foundations, etc.

The technical implementation of the non-dismantling vibrations control provides an opportunity for early diagnostics of pending damages in the machines and timely repair works.

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