Advances in Science and Technology Research Journal
Volume 8, No. 23, Sept. 2014, pages 68–72
DOI: 10.12913/22998624.1120325

Research Article

MECHANISM OF ORIGIN OF STRUCTURAL VIBRATIONS IN CONICAL ROLLER BEARINGS

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Received: 2014.07.22
Accepted: 2014.08.18
Published: 2014.09.09

ABSTRACT
This paper investigates the mechanism of origin of structural, structural and technological defects of rollers. The technique for integrated indicator of vibration working surfaces of the rings to determine the level of life of the finished part to the operation as part of the bearing and predict the vibroacoustic characteristics of rolling bearings. It was established that technological defects cause low-frequency and high-frequency vibrations. The question about the extent to which it is necessary to strengthen the tolerances on the parameters of bearings on which vibration level is determined not errors bearing parts and their structural properties. Calculated values of the amplitudes vibroacceleration due to the rigidity of the bearing vibrations are so small that in some cases adopted precision calculations turned enough to detect such vibrations. Thus, when tested on the vibro-acoustic installations structural vibration does not play an important role.

Keywords: vibroactivity, geometric undulation, structural vibration microtopography, design and technological factors.

INTRODUCTION

One of the main tasks that domestic bearing industry faces nowadays is to reduce the vibration and noise of rolling bearings. The main direction of solving this problem is to improve the technology of bearings manufacture: improving geometric accuracy of rings and rolling bodies (balls and rollers), reducing the roughness of the working surface, using isotropic to elastic properties of the starting materials, ensuring highly refined lubricants and parts and assembly cleanliness, proper conditions of installation and operation. Obviously, the implementation of the above mentioned measures is expensive and must be economically justified. Details of the modern high-precision and low-noise bearings must meet high requirements. To implement them in terms of mass and, moreover, serial production is extremely difficult and very expensive. Regarding this fact the question arises to what extent it is necessary to strengthen the tolerances on the parameters of bearings and where lower limit by which vibration level is determined not by errors of bearing parts and their structural properties.

It is known that even in cases when the bearing can be considered ideal in terms of the absence of errors in geometry and isotropy of parts' material, the presence of indissoluble elastohydrodynamic lubricant film between rings and rolling bodies completely clean from contamination of the lubricant, the bearing nevertheless vibrates. This vibration is common not only to perfect but also to ordinary bearings. The vibration is due to their structure and is called structural vibration, mainly due to two reasons [5, 8, 11].

The first reason is the bending deformation of the bearing rings forces acting on them from rolling bodies. Rings of bearing are folded under contact loads from rolling bodies, taking a form of a polyhedron that rotates with them. Such bending deformations are transferred to other parts of machines or distributed in the environment (for example air) in a form of acoustic waves, i.e. noise. The frequency of this vibration is the frequency of rolling bodies passing through the ring and multiple frequencies: qzf_c, where $q = 1, 2, ...$; z – number of rolling bodies; f_c – rotational speed of the separator. Most machines and devices of practical importance may have primarily bending deformation of the outer bearing rings [3, 5, 9, 11].

The second reason of the structural vibration is oscillation of stiffness of the bearing during rotation as a result of the radial component of the load. As the rolling bodies move to the line of the radial load, the stiffness of the bearing varies periodically with frequencies multiple to frequency of balls passing on non-rotating ring. Considering the most common scheme of rotating of inner and non-rotating outer rings we observe that as a result of fluctuations of stiffness relative displacement of the bearing rings occurs on the same frequencies as in the first case [5, 6].

It should be noted that despite the obviousness of these mentioned causes of structural vibration of bearings in scientific and technical literature there are no systematic quantitative evaluations of its significance. It is clear that the level of this vibration is relatively small because of the high rigidity of bearing rings, small values of the ratio of radial to axial load under conditions specific to the operation of most sizes of bearings. In bearings of low precision this vibration is not noticeable. At the same time, with increasing of bearings precision, structural vibration plays a much more visible role, and therefore, it requires scientific and reasonable evaluation of its significance [2, 4].

The assessment of the significance of structural vibration of bearings can be done in two ways. Firstly, comparing the calculated values with the values of the amplitudes, which are recorded practically. Second, comparing calculated values of the amplitudes of the structural vibration and the vibration caused by errors of bearings. In both cases the amplitudes of the harmonics vibration at frequencies qzf_c must be compared. There are no difficulties with the first way of assessment the causes if the problem of structural vibration of bearing is solved and amplitudes of vibratory acceleration of outer rings during practical measurements are known. The second method requires some explanation.

The theory of no ideal ball bearings with geometric errors of the working surfaces of the cones and balls allows to link errors spectrum with the spectrum of vibration [5, 11]. Since the structural vibration occurs only at frequencies qzf_c. To compare with the spectrum of vibrations of non-ideal ball bearing it should be highlighted that these harmonic components. The theory shows that the vibration at frequencies qzf_c is caused only by the geometry errors of the outer ring (as summing isotropy of the elastic properties of the material) and the harmonics of errors with numbers $\lambda = qz \pm 1$. Average value of amplitude of vibratory displacement of out erring as a rigid body in this case is [3]:

$$a_{f=qzf_c} = \sqrt{a^2_{\lambda=q-1} + a^2_{\lambda=q+1}}\,,$$

$$A_{f=qzf_c} = \sqrt{a^2_{\lambda=q-1} + a^2_{\lambda=q+1}} \quad (1)$$

first harmonic of vibratory displacement that is the most important (at a frequency of passing rolling bodies in the outer ring zf_c) is specified by harmonic geometry errors of the outer ring with numbers $\lambda = z + 1$ and $\lambda = z - 1$ that regardless of phase emerge in (1) mean-square.

Thus, to assess the importance of structural vibration of roller bearing with another method we should calculate the amplitude of the first harmonic of the structural vibration of the two considered sources and compare them directly with amplitudes of harmonic waviness of the outer ring of numbers $\lambda = z + 1$ and $\lambda = z - 1$.

We solve the problem of structural vibration of roller bearing, specified by two above-mentioned reasons based on that decision and calculate the amplitude of vibratory accelerations and vibratory displacements of surfaces of the outer rings, the most commonly used in machinery and tool engineering of roller bearings with their characteristic operating conditions at frequencies qzf_c and make comparison of calculation results with actual according to the accuracy of the rings and vibration levels for the same bearings. Experimental studies of the vibration level were conducted on automated measuring complex DVK [4].

VIBRATION FROM CHANGES OF STIFFNESS OF ROLLER BEARING

To solve the problem of structural vibration of roller bearing, caused by variability of its rigidity during rotation under combined loads we should solve the problem of static of perfect bearing, which involves determining the position of the inner ring as to the outer under the applied load. A characteristic feature of a roller bearing as no rigid deformation system is static uncertainty, and therefore, the direct analytical solution of the direct problem of statics in general case is impossible [1, 11]. Thus, in the calculation of the roller bearing initially the inverse problem is solved: the reaction vector is determined $\overline{R}=[F_x, F_y, F_z, m_y, m_z]^T$ by responses to the vector of relative displacement of the rings $\overline{\Delta}=[x, y, z, Q_y, Q_z]^T$ then by approximate analytical methods [11] or numerically [1] the direct problem of statics is solved.

Nonlinearity of dependence $\overline{R}=\overline{R}(\overline{\Delta})$ makes it impossible to exact the solution of the direct problem of static roller bearings under conditions of combined loading. During the solution of the inverse problem the bearing is divided into z mutually independent contact groups, where z is a number of rollers, each of which consists of a roller and two rings. For each roller on the relative position of the rings contact force is calculated if defined model of the contact group, and then by summing over all the rollers the vector of responses is calculated.

We use a generally accepted Hertz model of contact group roller [10] – the ring that connects the contact force Q with the contact deformation ω with following dependence:

$$Q(\omega)=K_H\omega^{1,5},$$

where: K_H – Hertz coefficient [11].

The placement of j-th roller as to the fixed point of the outer ring is defined by angle $\varphi_j=\gamma+2\pi(j-1)/n$, where φ_j – angle of the separator bending on the outer ring. We introduce the denotations and also basic rule of signs [10]. In this case, the deformation of the j-th contact group ω_j is determined by the vector relatively to displacement of rollers in the following way:

$$\omega(\overline{\Delta},\varphi_j)=\begin{cases}\left\|L(\varphi_j)\overline{\Delta}-\overline{l}'\right\|, f & \left\|L(\varphi_j\ \overline{\Delta}-\overline{l}')\right\|\succ 0\\ 0, f & \left\|L(\varphi_j\ \overline{\Delta}-\overline{l}')\right\|\leq 0\end{cases},$$

where:

$$L(\varphi_j)=\begin{bmatrix}1 & 0 & 0 & -R_B\cos\varphi_j & -R_B\sin\varphi_j\\ 0 & \sin\varphi_j & \cos\varphi_j & 0 & 0\end{bmatrix}$$

$$\overline{l}'=l'\begin{bmatrix}\sin\tau_0\\ \cos\tau_0\end{bmatrix}$$

l' – initial distance between the inner and outer rings is defined in the radial cutting plane passing through the axis of the j- th and the center of the inner ring.

$$\cos\tau=(R_b+r_b)-(R_z+r_z)/l',$$

where:

$$l'=r_z+r_{\hat{a}}-d_m;$$

φ – contact angle;
r_z, r_b – in accordance the initial radius of the outer and inner rings;
R_z, R_b – finite radii of the outer and inner rings;
d_m – diameter of the roller.

Working contact of the angle of j-th roller of rings is calculated by the formula:

$$\tau(\overline{\Delta},\varphi_j)=arctg\left(\frac{L^1(\varphi_j)\overline{\Delta}}{L^2(\varphi_j)\overline{\Delta}}\right),$$

where: L^1– the first and L^2 – second row of the matrix $L(\varphi)$.

The component of the vector load on roller bearing $\overline{F}$ is caused by the contact forces from the j-th roller,

$$\overline{F}\varphi_j=[F_x, F_y, F_z, m_y, m_z]^T\ \varphi_j=\overline{E}(\tau)\,Q(\omega) \quad (2)$$

where:

$$\overline{E}(\tau)=[-\sin\tau, 0, -\cos\tau, R_B\sin\tau, R_B\cos\tau, -r_B]^T$$

Let us denote by $T(\varphi_j)$ matrix that relates the coordinates of the vector $\overline{F}\varphi_j$ in the basis OXZ with coordinates of the same vector in the basis $\varphi_j X\varphi_j Y\varphi_j\varphi$:

$$T(\varphi_j)=\begin{bmatrix}1 & 0 & 0 & 0 & 0\\ 0 & \cos\varphi_j & \sin\varphi_j & 0 & 0\\ 0 & -\sin\varphi_j & \cos\varphi_j & 0 & 0\\ 0 & 0 & 0 & \cos\varphi_j & \sin\varphi_j\\ 0 & 0 & 0 & -\sin\varphi_j & \cos\varphi_j\end{bmatrix}$$

Then the solution of the inverse problem of static roller bearing has the form:

$$\overline{R}(\overline{\Delta})=\sum_{\varphi_j}T(\varphi_j)\,\overline{F}(\varphi_j)=\sum_{\varphi_j}T(\varphi_j)\,\overline{E}(\tau)\,Q(\omega) \quad (3)$$

Direct static problem, which involves determining the vector of relative displacement of rings $\overline{\Delta}$ at angle of separator bending and vector of load $\overline{P}=[P_x, P_y, P_z, G_y, G_z]^T$ that should be considered as a problem of determining the root $\overline{\Delta}$ of the equation $\overline{R}(\overline{\Delta})=\overline{P}$. Numerical solution of this problem is convenient to carry out using the method of Newton-Raphson. To this end, for this purpose at each step of iterations as to the known rules of differentiation tangent stiffness matrix of the form is calculated:

$$\frac{\partial R}{\partial \overline{\Delta}} = \sum_{\varphi_j} T(\varphi_j) \left\{ \frac{\partial \overline{E}_1}{\partial \tau}(\tau) \frac{\partial \tau}{\partial \overline{\Delta}}(\varphi_j, \overline{\Delta}) Q(\omega) + \right. \left. + \overline{E}(\tau) \frac{\partial Q}{\partial \omega}(\omega) \frac{\partial \omega}{\partial \overline{\Delta}}(\varphi_j, \overline{\Delta}) \right\} \quad (4)$$

The calculations terminates when $\left\| \overline{R}(\overline{\Delta}) - \overline{P} \right\|$ becomes less than 0.01 (in the units of H, mm). The implementation of this algorithm for solution of the problem of static on the computer proved its high efficiency – the number of iterations required by the proper choice of the first approximation does not exceed 7. As calculations of many types of roller bearings have shown significant value is only the first harmonic vibration in the frequency spectrum zf_c, harmonics of higher or small amplitude and practice do not matter.

The comparison of the calculated values of the vibratory acceleration amplitudes of roller bearings due to bending vibrations of the outer ring in the axial load characteristic for testing of bearings for driven units [4] with the values established by standards for low-noise vibration of roller bearings shows that the calculated value is at 11–23 dB below established standards for the relevant frequency range. Calculated values of the vibratory acceleration amplitudes due to the rigidity of the bearing vibrations are so small that in some cases adopted precision calculations were not enough to detect such vibrations. Thus, when tested on the vibroacoustic installations structural vibration does not play an important role.

To assess the importance of structural vibration of roller bearing value of its amplitude should be compared with the tolerances for amplitude of harmonic undulation of bearings components. At present, the general tolerances for the undulation of tracks of rolling outer rings, for example 2-class accuracy, reaches 0,1–0,2 m, depending on the size of the rings. Thus, the structural vibration is significant if its amplitude is larger than 0,1–0,2 microns. As it follows from [11], parameters that define the structural vibration during measurement of vibration on the vibro-acoustic setup always at least one order less than specified undulation tolerances, and therefore, structural vibration in this case does not play any role. The amplitudes of the structural vibration of roller bearings caused by bending deformations of the outer ring under severe axial tension are equal or approximately equal to the tolerances for undulation of rolling ring tracks of bearing accuracy class 2. Structural vibration amplitudes caused by fluctuating rigidity under severe tension and a very high ratio of radial stress to axial in most cases have much lower tolerance for waviness.

The performed analysis shows that the vibration of roller bearings under the characteristic conditions of control and exploitation conditions is due not to the design (structure) peculiarities of the bearing but technological errors, lubrication conditions and other factors. The level of structural vibrations of modern precision roller bearings does not exceed the level of vibration caused by technological reasons, including errors of macro geometry of tracks of bearing rings, and the undulation of working surfaces.

The most important is the structural vibration of roller bearing due to bending vibrations of the outer rings as a result of their loading by rollers that move. Oscillation amplitude of surfaces of outer rings in these cases can be quite large; such fluctuations may have a significant source of acoustic noise.

The vibration of the bearing due to fluctuations in its rigidity during rotation under combined loads in real bearings is very small and is the source of vibrations in machines and devices that can be ignored [5, 10, 11].

CONCLUSIONS

The mechanism of origin of constructural, structure and process defects of roller bearings is investigated. The relationship between the defect and performance indicators is determined on the example of changes of vibro-acoustic characteristics of the bearing. This fact served as the basis for development of a strategy of quality of components prediction and performance indicators of bearing on the stage of the design process. To diagnose common causes of vibrations, general spectrum of fluctuations of bearing is advisable to divide into three bands – low frequency oscillations (frequency from 50 to 1500 Hz), medium (1500–3000 Hz) and high frequency vibration (1500–10 000 Hz). It was established that technological defects cause low-frequency and high-frequency vibrations. The proposed technique according to integrated indicator of vibration of working surfaces of the rings determines the level of life of the finished part for the exploitation as a part of the bearing and predicts the vibro-acoustic characteristics of roller bearing.

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