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THE EFFECT OF THE RADIAL CLEARANCES ON THE DYNAMIC PROCESSES IN ROLLING BEARINGS

A method for calculating the parameters of shock pulses generated in rolling bearings due to the presence of radial clearance in them is proposed. The method is a further development of theoretical research in the scientific direction of vibration-pulse diagnostics in relation to the operational assessment of the technical condition of bearing units as components of the gear drive mechanism, which largely determine the drive resource as a whole. The calculation is based on the consideration of the precession movement of the bearing axis, which it makes during the rotation of its shaft, and the theory of shock interaction of elastic bodies by G. Hertz. The mechanism of occurrence of shock pulses has been studied with an additional introduction to the calculation of the radial force acting on the shaft. It is shown that wear, which causes an increase in the working radial clearance between the races and the rolling bodies in the bearing, increases the speed and amplitude of the impact interaction of the inner ring with the rolling bodies. An example of calculating the parameters of shock pulses in ball radial single-row bearings of the 307 series is given. Conclusions are made about the prospects of using the developed method for diagnostics and forecasting the remaining life of bearings of transmission units of mobile machines.

Keywords: rolling bearing, dynamic process, technical condition, diagnostics, radial clearance, shock pulse, shock pulse parameters.

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ВПЛИВ РАДІАЛЬНИХ ЗАЗОРІВ НА ДИНАМІЧНІ ПРОЦЕСИ В ПІДШИПНИКАХ КОЧЕННЯ

Запропоновано метод розрахунку параметрів ударних імпульсів, що генеруються в підшипниках кочення через наявність в них радіальних зазорів. Метод є подальшим розвитком теоретичних досліджень з наукового напрямку вібраційно-імпульсного діагностування стосовно експлуатаційної оцінки технічного стану підшипникових вузлів як складових елементів зубчастого приводного механізму, значною мірою визначають ресурс приводу в цілому. В основу розрахунку покладено розгляд прецесійного руху осі підшипника, яке вона здійснює в процесі обертання його вала, і теорія ударної взаємодії пружних тіл Г. Герца. Механізм виникнення ударних імпульсів досліджений з урахуванням додаткового введення в розрахунок величини радіальної сили, що діє на вал. Показано, що зношування, будучи причиною збільшення робочого радіального зазору між обоймами і тілами кочення в підшипнику, збільшує швидкість і амплітуду ударної взаємодії внутрішнього кільця з тілами кочення. Наведено приклад розрахунку параметрів ударних імпульсів в кулькових радіальних однорядних підшипниках серії 307. Зроблено висновки про перспективи застосування розробленого методу для діагностики та прогнозування залишкового ресурсу підшипників трансмісійних вузлів мобільних машин.

Ключові слова: підшипник кочення, динамічні процеси, технічний стан, діагностика, радіальний проміжок,ударний імпульс, параметри ударного імпульсу.

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ВЛИЯНИЕ РАДИАЛЬНЫХ ЗАЗОРОВ НА ДИНАМИЧЕСКИЕ ПРОЦЕССЫ В ПОДШИПНИКАХ КАЧЕНИЯ

Предложен метод расчета параметров ударных импульсов, генерируемых в подшипниках качения из-за наличия в них радиальных зазоров. Метод является дальнейшим развитием теоретических исследований по научному направлению вибрационно-импульсного диагностирования применительно к эксплуатационной оценке технического состояния подшипниковых узлов как составных элементов зубчатого приводного механизма, в значительной мере определяющих ресурс привода в целом. В основу расчета положено рассмотрение прецессионного движения оси подшипника, которое она совершает в процессе вращения его вала, и теория ударного взаимодействия упругих тел Г. Герца. Механизм возникновения ударных импульсов исследован с учетом дополнительного введения в расчет величины радиальной силы, действующей на вал. Показано, что изнашивание, являясь причиной увеличения рабочего радиального зазора между обоймами и телами качения в подшипнике, увеличивает скорость и амплитуду ударного взаимодействия внутреннего кольца с телами качения. Приведен пример расчета параметров ударных импульсов в шариковых радиальных однорядных подшипниках серии 307. Сделаны выводы о перспективах применения разработанного метода для диагностики и прогнозирования остаточного ресурса подшипников трансмиссионных узлов мобильных машин.

Ключевые слова: подшипник качения, динамические процессы, техническое состояние, диагностика, радиальный зазор,ударный импульс, параметры ударного импульса.

Introduction. The presence of structural radial clearance in rolling bearings causes the formation of shock pulses. During the rotation of the bearing shaft, its axis makes a precession movement, as a result of which the shaft collides with rolling bodies and there is a shock action that causes vibration in the mechanism. The resulting shock forces are characterized by a short duration (about 10^{-3} seconds) and a significant value.

During operation, due to wear of bodies and raceways, the radial clearance of the rolling bearing increases and, as a result, its vibration activity increases. At the same time, even a slight increase in the clearance in some cases may cause the bearing to lose working capacity.

Therefore, vibration characteristics can serve as effective diagnostic criteria that allow evaluating the technical condition of rolling bearings and is fixedthe processes of defect development without disassembling the mechanism.

In this work, developed in the Joint Institute of Mechanical Engineering of the National Academy of Sciences of Belarus, vibration-pulse method of diagnostics of gear drives complemented by the development of calculation method of parameters of the impulse roller bearing (peak value of the shock pulse, speed and duration of impact). The combined application of these methods makes it possible to evaluate the vibration of rolling bearings depending on the radial clearance value during the development of wear processes on rolling surfaces.

Problem statement. Clearances in rolling bearings are necessary to ensure assembly and prevent rolling bodies with rings from jamming during operation. They compensate for the reduction of the distance between the inner and outer rings of the bearing when the bearing assembly is significantly heated, interference fit, and the rings are displaced. Thus, clearances are one of the most important factors affecting the performance of bearings [1].

The initial geometric clearance in bearings is regulated by the relevant standards, in particular GOST 24810. An increase in radial clearances relative to those correctly selected for specific operating conditions increases the unevenness of the load distribution between the rolling bodies, shortens the service life of bearings, and increases vibrations [2]. On the other hand, clearance reduction impairs the ability of ball bearings to accept axial load, leads to increase in temperature, and reduces the maximum allowable rotational speeds.

If the operating conditions are maintained, there is usually no noticeable wear of rolling bearings. Wear occurs most often when fouling particles penetrate the bearing or there is insufficient lubrication. Many machines operating in an abrasive environment, such as automobiles, agricultural, mining, construction and road vehicles, etc., are susceptible to this, despite oil seals and filtration. Thus, it is known [3] that according to the results of a statistical survey, 2.5 times more bearings were rejected due to wear of rings and rolling bodies of tractor bearings than due to pitting.

It should also be taken into account that if the radial clearance is too large, the shaft beat, the smoothness of rotation decreases, the load zone of the rolling elements decreases (the smaller the clearance, the more evenly and on a larger number of rolling elements the load is distributed), and the bearing life decreases [4].

One of the variants for determining the maximum allowable value of the radial clearance due to wear is proposed in the form of a wear coefficient [5]:

$$f_v = \frac{V}{e_0} = \frac{V}{0.46 d^{2/3}} .$$

where V – the total wear in micrometers that comes as the increased radial clearance measured in an unmounted bearing; d – the inner diameter of the bearing.

Hence the overall wear in micrometers

$$V = 0.46 f_v d^{2/3} .$$

A number of wear factors for various applications are shown in Table 1.

Table 1 – Representative limits for wear factor [5]

Application	Limiting value of wear factor
Agricultural machines	8–25
Construction machinery	6–12
Crushers	8–12
Electric motors	3–5
Gears, general engineering	3–8
Machine tool spindles	0.5–1.5
Motor vehicles	3–5
Rail vehicles	6–12

Higher values of the factor would be expected with more difficult conditions involving such factors as high speed, rapid speed changes, decreased lubricant efficiency, decreased seal efficiency, and environmental contamination.

A method is known for diagnosing the technical condition of the gear motor-wheel of mining dump trucks, based on tracking the change of the clearance in the bearing (backlash is controlled by a special sensor in the thrust ring of the bearing) to prevent wear on gears. During the time between scheduled disassembly of the specified gearbox (3 years), wear is 0.5…1 mm, and at a value of 1.03 mm, repairs are carried out [6].

It is also possible to prevent premature repairs and significantly reduce maintenance costs by carrying out the necessary amount of work by monitoring the output of vibration parameters beyond the nominal values that are caused in the unit by wear and aging of bearings. In this case, the technical condition of the bearing assembly is normalized through the radial clearance [7], and the change in the value of the radial clearance is predicted depending on the vibration parameters.

At the same time, it is known that during the rotation of the bearing shaft, its axis makes a precession movement, as a result of which the shaft collides with rolling bodies and there is a shock interaction that causes vibration in the mechanism [8]. The resulting shock forces are characterized by a short duration (about 10^{-3} seconds) and a significant amount.

Based on this, in this paper, the task is to supplement the vibration-pulse method of diagnostics of gear drives developed in the Joint Institute of Mechanical Engineering of the National Academy of Sciences of Belarus [9, 10] with the development of a method for calculating the parameters of the shock pulse for rolling bearings. The combined application of these methods will make it possible to evaluate the vibration of bearings depending on the size of the radial clearance in the development of wear processes of rolling surfaces.

The process of occurrence of shock pulses in rolling bearings with a radial clearance. In this paper, the shock pulse is understood as the law of change of the shock force over time. The process of occurrence of shock pulses when the shaft collides with rolling bodies (Fig. 1) occurs as follows [8, 11]:

– Due to the radial clearance Δ_p, the position of the shaft during rotation is stable for a period of time when it is supported by two rolling bodies.

– At the moment when one of the rolling bodies reaches the lower position, the inner ring of the bearing together with the shaft is tipped over on the rolling body following it.

The radius R_B of the inner ring raceway (Fig. 1, a) is less than the radius of the circle R_{1B}, touching all the balls, by an amount $\Delta = 0{,}5\Delta_p$

$$R_{1B} = R_B + \Delta .$$

When falling the inner ring of the bearing passes the distance h (Fig. 1, b)

$$h = \Delta(1 - \cos\varphi_0) , \tag{1}$$

where $\varphi_0 = 2\pi/N$ – the angular distance between neighboring rolling bodies; N – number of rolling bodies.

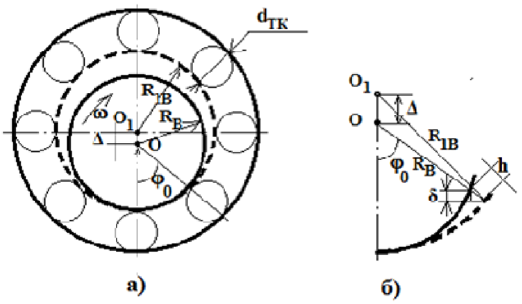

Fig. 1 – The process of occurrence of shock pulses when the shaft collides with rolling bodies

Methodforcalculatingthe parameters of shock pulses. The main parameters of the shock pulse are the amplitude and duration of the impact force. The calculation of these parameters is based on the theory of shock interaction of elastic bodies developed by G. Hertz. In this theory, the following assumptions are made: the linear dimensions of the contact zone of touching bodies are small compared to the radius of curvature of their contact surfaces; the material under deformation obeys Hooke's law; the force compressing the bodies is normal to the contact surface.

The process of shock interaction of the bearing shaft with rolling bodies is shown in Fig. 2. The elastic impact of two bodies is considered. The first body of mass M (the shaft with the bodies attached to it), which is affected by the radial force F, is struck at a speed of V_0 into the second body of mass m (the rolling body). In this case, there is a central impact of two bodies. It is assumed that the mass of the rolling body is significantly less than the mass of the shaft: $m << M$.

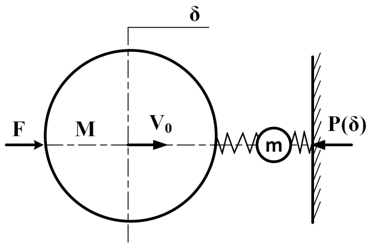

Fig. 2 – Impact interaction of the shaft with the rolling body

The speed of the shaft V_0, acquired at the moment of impact, is determined from the equality of the kinetic energy of the shaft with the bodies attached to it and the radial force F. The kinetic energy of the shaft at the moment of collision is equal to

$$\frac{MV_0^2}{2} = Fh_1 . \quad (2)$$

From Fig. 1, b follows

$$h_1 = h\cos\varphi_0 . \quad (3)$$

For small anglesφ_0 we can assume that `

$$\cos\varphi_0 \approx 1, \quad 1-\cos\varphi_0 \approx \frac{\varphi_0^2}{2} . \quad (4)$$

After substituting equality (2) to expressions (1), (3), (4) the speed of impact is found

$$V_0 = \varphi_0\sqrt{\frac{F\Delta}{M}} = \frac{2\pi}{N}\sqrt{\frac{F\Delta}{M}} . \quad (5)$$

The differential equation of the shaft movement has the form

$$M\ddot{\delta} = F - P(\delta) , \quad (6)$$

where $P(\delta)$ – contact force; $\delta = \delta_1 + \delta_2$ – convergence of bodies as a result of local compression; δ_1 – convergence of the bearing inner ring and the rolling body, δ_2 – convergence of the rolling elements and outer ring.

For ball bearings that have a contact interaction of bodies bounded by curved surfaces and touching until deformed at a single point, the relationship between the contact force and the convergence of bodies is described by the expression [12]

$$P(\delta) = \lambda\delta^{\frac{3}{2}} , \quad (7)$$

where

$$\lambda = \frac{2^{\frac{3}{2}}}{\left(\sum_{i=1}^{2} n_{\delta i}\sqrt[3]{\frac{9}{4}\eta_i^2\sum k_i}\right)^{\frac{3}{2}}} . \quad (8)$$

In expression (8), $n_{\delta i}$ – the coefficients that characterize the convergence of touching bodies; η_i – the elastic constants of touching bodies, which are equal for the same materials $\eta = 2(1-\nu^2)/E$; E – the modulus of elasticity; ν – the Poisson's coefficient; $\sum k_i$ – the sum of the principal curvatures of the surfaces of touching bodies.

The values of the coefficients $n_{\delta i}$ depend on the value of the parameter Ω by interpolation.

Geometric parameter Ω that characterizes the curvature of touching bodies [12]:

$$\Omega = \frac{k_{21} - k_{22}}{\sum k_{ij}} . \quad (9)$$

Integration of equation (6) under the condition of constant radial force F allows us to determine the maximum convergence of bodies δ_{max} due to local deformation,

$$\delta_{\max}^{\frac{5}{2}} - \frac{5}{2}\frac{F}{\lambda}\delta_{\max} - \frac{5}{4}\frac{MV_0^2}{\lambda} = 0 . \quad (10)$$

After finding the δ_{max} from (10), the amplitude of the shock pulse is calculated

$$P_{\max} = \lambda \delta_{\max}^{\frac{3}{2}} . \qquad (11)$$

The duration of the shock pulse T is found from the integration of the expression

$$T = 2\frac{\delta_{\max}}{V_0}\int_0^1 \frac{d\xi}{\sqrt{1+2\alpha\xi-\beta\xi^{\frac{5}{2}}}} , \qquad (12)$$

where $\xi = \delta/\delta_{\max}$, α, β are dimensionless quantities.

The value ξ varies within $0 \le \xi \le 1$. The values α, β are calculated using the formulas

$$\alpha = \frac{F \cdot \delta_{\max}}{m \cdot V_0^2}; \quad \beta = \left(\frac{\delta_{\max}}{\delta_{0\max}}\right)^{\frac{5}{2}} , \qquad (13)$$

where $\delta_{0\max}$ – the maximum compression of bodies caused by the impact velocity V_0 in the absence of a radial force ($F = 0$).

The value $\delta_{0\max}$ is determined by the formula [13]

$$\delta_{0\max} = \left(\frac{5}{4}\cdot\frac{M \cdot V_0^2}{\lambda}\right)^{\frac{2}{5}} . \qquad (14)$$

Example of calculating the parameters of shock pulses. We will calculate the parameters of the shock pulse in ball radial single-row bearings of the 307 series (Fig. 3). The reduced weight of the shaft $M = 6.5$ kg. Radial force F =1500 N. the characteristics of the bearing 307 are shown in Table 2.

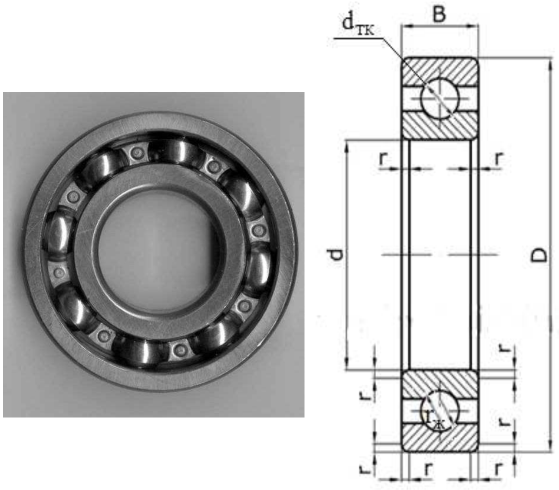

Fig. 3 – Deep groove ball bearing series 307

Table 3 shows the values of the speed of impact of the shaft V_0, the maximum convergence of bodies $\delta_{\max}$ due to local deformation, the duration of the impact T, the amplitude of the impact pulse $P_{\max}$, depending on the values of the radial clearance Δ_p.

The dependence of the amplitude of the shock pulse and the speed of the shaft impact on the radial clearance, taking into account wear, is shown in Fig. 4.

Table 2 – Deep groove ball bearing series 307 characteristics

Parameter		Designation	Value	Units
The inner diameter of the bearing		d	35	mm
The outer diameter of the bearing		D	80	mm
Bearing width		B	21	mm
Diameter of the rolling body		d_{TK}	12,7	mm
Number of rolling bodies		N	8	–
Rolling body weight		m_{TK}	$8{,}42\cdot10^{-3}$	kg
The value of the initial radial clearance		Δ_p	$6\ldots20\cdot10^{-6}$	m
Diameter of the raceway [6]	Bearing inner ring	D_B	44,8	mm
	Bearing outer ring	D_H	70,2	mm
Sum of main-curvatures[6] for the ball contacting	the inner raceway	$(\sum k_{ij})_1$	208,2	m^{-1}
	the outer raceway	$(\sum k_{ij})_2$	135,5	m^{-1}

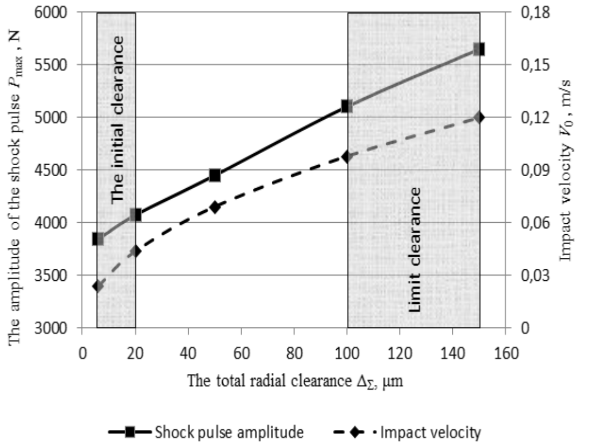

Fig. 4 – The dependence of the amplitude of the shock pulse and the speed of the shaft impact on the radial clearance

Table 3 – Table of values V_0, $\delta_{\max}$, $\delta_{0\max}$, T, P_{max}

Parameter	Radial clearance, μm				
	6	20	50	100	150
V_0, m/s	0,0207	0,0377	0,0597	0,0844	0,1033
$\delta_{\max}$, μm	47,56	49,41	52,68	57,42	61,44
$\delta_{0\max}$, μm	10,56	17,74	25,59	37,77	39,72
T, ms	1,48	1,42	1,32	1,19	1,13
P_{max}, N	3844	4071	4448	5099	5644

Conclusions. The mechanism of occurrence of shock pulses and the processes of interaction of the shaft with the rolling bodies of the bearing are studied, the dependence of the shock pulses parameters on the speed of rotation of the shaft and the load is analyzed. The proposed calculation method allows us to evaluate the dynamic processes in new bearings depending on the radial clearance value, as well as the development of wear processes on the rolling surfaces of the bearing.

To establish an analytical relationship between the amplitudes of shock pulses in rolling bearings and the corresponding amplitudes of vibration pulses perceived by the vibration sensor, a vibration-pulse diagnostic method can be used, developed to solve an analogous problem in relation to toothed gears.

The use of the developed method allows to significantly increase the efficiency and reliability of diagnostics and forecasting of the residual life of elements of gear drives of transmission units of mobile machines by expanding the scope of vibration-pulse diagnostics.

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