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Вибромониторинг технического состояния трансмиссионных систем мобильных машин

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Vibrating Monitoring Technical Condition of the Transmission Systems of Mobile Machines

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Изложены результаты комплексных теоретических и экспериментальных исследований ударного взаимодействия зубчатых профилей применительно к задачам вибродиагностики трансмиссионных систем мобильных машин. Разработаны методы аналитического и экспериментального определения резонансных режимов работы зубчатых механизмов, фактического коэффициента торцового перекрытия, а также остаточного ресурса зубчатых передач по результатам вибромониторинга при их испытаниях и в эксплуатации.

Ключевые слова: мобильная машина, трансмиссия, техническое состояние, вибромониторинг, методика, динамический анализ, ударный импульс, резонансные режимы работы, коэффициент торцового перекрытия, остаточный ресурс.

The results of a comprehensive theoretical and experimental studies of shock interaction gear profiles in relation to problems of vibration transmission systems of mobile machines. The methods of analytical and experimental determination of the resonant modes of gear mechanisms, the actual contact ratio and residual life of gears according to the results of vibration monitoring for their testing and operation are worked out.

Keywords: mobile machine, transmission, technical condition, vibrating monitoring, methods, dynamic analysis, shock pulse, resonant modes, contact ratio, residual life.

Introduction. As a result of executing a comprehensive theoretical and experimental research at the Joint Institute of mechanical engineering of the NAS of Belarus has developed a number of methods when they are used together to carry out the estimation of the residual life of the transmission systems of mobile machines under operating conditions. The basis for research on the development of methods for dynamic analysis of geared nodes on the basis of the evaluation of shock pulses in the toothed gearings. This development allows the allocation of an oscillatory process system components and criteria that correspond to a specific defect specific gear drive mechanism, and then to evaluate these components, technical condition of this gear [1].

The solution of this problem based on the fact that in gears of general engineering applications in the initial phase of the engagement of the teeth due to their deformation, to manufacturing errors and Assembly of gear wheels arise shock pulse, vibration generating processes in the mechanisms. With the development of damages in the teeth to change the amplitude and energy of the shock pulse, determined by its form, leading in turn to changes in the vibrational characteristics of the gears. Experimental studies allowed us to establish correlations between them [2]. On this basis was established method of estimating the residual life gear drive mechanisms and transmission systems of mobile machines on the results of their periodic vibration monitoring.

Studies of shock interaction gear profiles in relation to problems of vibration. To estimate the parameters of shock pulses in spur gears adopted a dynamic model (fig. 1), considering only torsional vibrations of wheels (generalized coordinates φ_1, φ_2) relative to the uniform rotation of gear wheels with a constant angular velocity, due to manufacturing errors and Assembly of gear wheels [3].

The differential equation that establishes the relationship between the impact force F and the acceleration of the teeth d^2x_3/dt^2 in their relative movement in the strike has the form

$$\frac{d^2x_3}{dt^2} = -\frac{F}{\mu_{\text{пр}}},$$

where x_3 is the convergence of teeth due to local compression in the contact zone at an arbitrary time t; $\mu_{\text{пр}}$ is the reduced mass of the wheel gear.

The calculations have shown [4] that with a sufficiently high degree of accuracy in the first approximation we may take the form of a shock impulse (law changes impact forces in time) interacting gears described by a sine wave

$$F = F_{\max} \sin\omega_3 t = V_0 \sqrt{\frac{\mu_{\text{пр}}}{\delta_{\text{кср}}}} \sin\omega_3 t,$$

where $\delta_{\text{кср}}$ is the average value of the contact compliance mating teeth to the extent that the impact force from zero to maximum, ω_3 — the natural frequency of the gear; $F_{\max}$ is the amplitude of the shock pulse, V_0 —edge shock speed.

The approximate values of the duration of the shock pulse is

$$T = \pi / \omega_3.$$

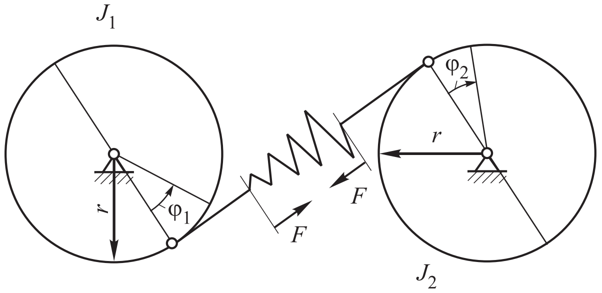

Fig. 1. Dynamic scheme of gearing

The results of theoretical studies were validated by conducting strain measuring gear bench testing of spur gears (fig. 2).

Fig. 3 shows the waveforms of the load of the tooth of the driven wheel. The first wave is the sum of the impact pulse appearing at the input of the tooth engages and district forces caused by an external torque. Subsequent changes to the loading of

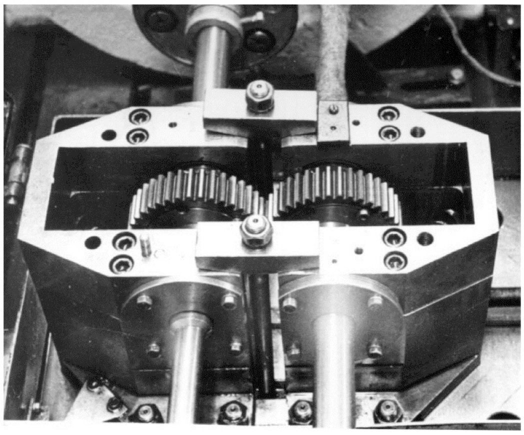

Fig. 2. Stand test

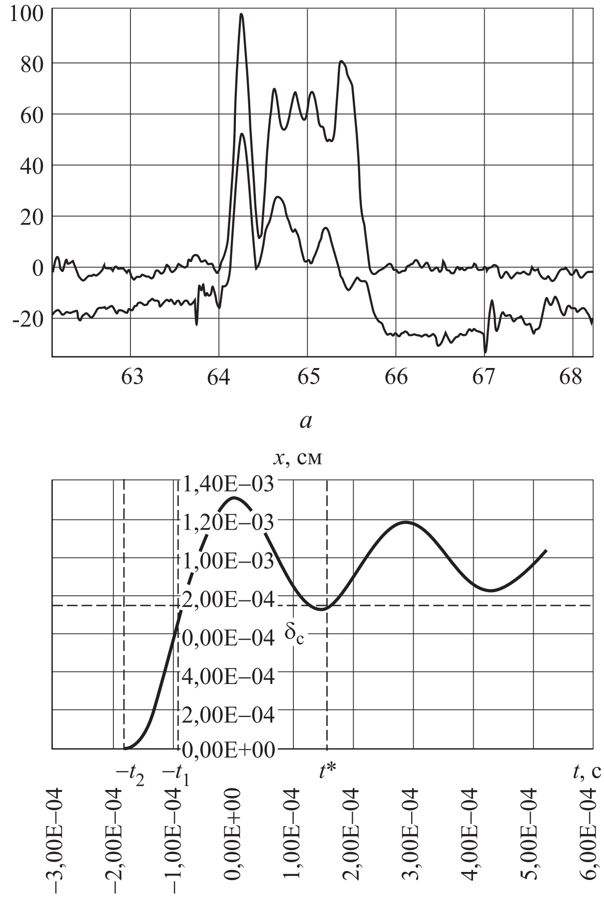

Fig. 3. Shock pulse waveform of teeth loading:
a — experiment; *b* — calculation

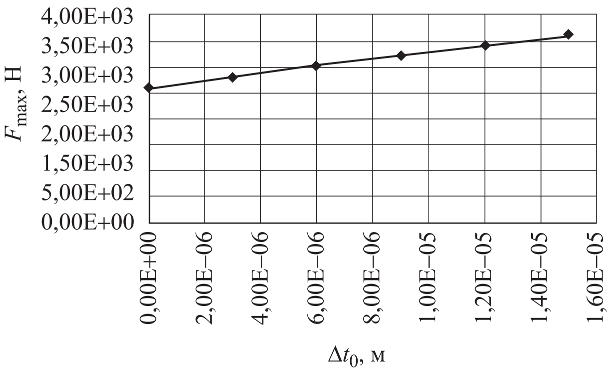

Fig. 4. Graph of the amplitude of the shock pulse depending on the change of error Δt_0 engagement

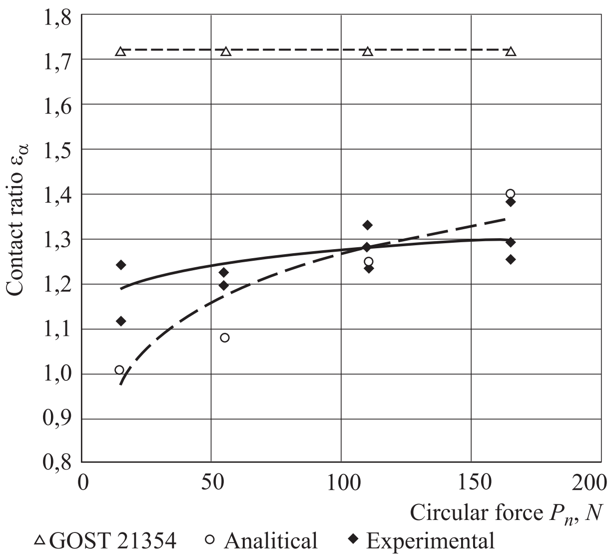

Fig. 5. Dependence of the calculated and experimentally found values of the contact ratio from the ring of power P_n

the tooth meet dynamic oscillatory process masses of gear-wheels on the stiffness of the mesh.

The results of calculations have shown that the amplitude of the shock pulse is dependent on the error of the step gear meshing, the magnitude of the torque applied to the leading gear and speed of pinion [5].

An example of the dependence of the amplitude of the shock pulse investigated toothed gear from changes in the meshing error Δt_0 shown in figure 4.

Determining of the actual contact ratio. To calculate the contact stresses in the gear mate, you must know the real value of the contact ratio, which may differ significantly from the theoretically calculated.

In gears of mobile machines the actual contact ratio ε_α, largely determines the resource, dynamic and vibration characteristics of the mechanism, substantially lower than the theoretically calculated by known dependencies (GOST 21354–87 or proposed by C.D. Andozhskiy and E.B. Bulgakov), because they do not take into account the influence of dynamic processes in the transmission, change the picture of the deformed state gear in time.

The proposed method of calculation ε_α, in contrast to the known, takes into account the internal dynamics of engagement, due to manufacturing errors and assembly of toothed wheels, the deformation of the teeth under load [6].

Depending on inertial stiffness parameters, the values of damping in the gear pair, the ratio of errors of steps, engagement Δ_0 and deformation of the front pair of teeth δ_c can be two cases in the course of dynamic processes in the gear mesh. In the first case the process of twodimensional engagement ends while the contact point of the considered pairs of teeth on the irregular section line gear $t = t_1$, the second — after the release of the teeth on the theoretical line of engagement at the time $t = t^*$. The residence time t2 of the teeth outside the theoretical line of engagement is determined by the conditions of edge engagement.

Estimated value of the contact ratio $\varepsilon^p_{\alpha 1}$ for the first and $\varepsilon^p_{\alpha 2}$ for the second course of dynamic processes in cylindrical spur transmission are respectively according to the formula

$$\varepsilon^p_{\alpha 1} = 1 + \frac{|t_2| - |t_1|}{T_z}, \quad \varepsilon^p_{\alpha 2} = 1 + \frac{|t_2| + t^*}{T_z}.$$

For experimental gear pair with parameters: module m = 3 mm, number of teeth gear wheel $z_1 = z_2 = 40$ were conducted analytical and experimental evaluation of the contact ratio depending on the size of the district power P_n of the current gear engaged. Graphs of the calculated and experimentally found contact ratio is shown in fig. 5.

The figure shows that the calculated value of ε_α is not more than 1.3. The coefficient of the contact ratio defined by the well-known dependency for the studied transmission is is $\varepsilon_\alpha = 1{,}72$.

Calculated values of the spectra of periodic shock pulses gear pair. In timing the transmission of a periodic action of shock pulses takes place with the period of the meshing frequency T_z (fig. 6)

$$T_z = \frac{60}{nz}.$$

Therefore, the function of the shock strength in time can be represented by a decomposition into Fourier series [7]:

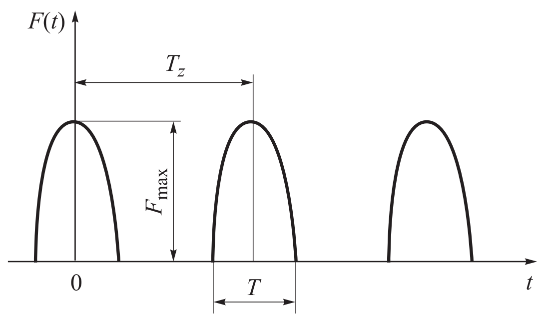

Fig. 6. Periodic shock pulses

$$F(t) = F_0 + \sum_{k=1}^{\infty} F_k \sin(\omega_k t + \varphi_\kappa),$$

where $\omega_k = k\omega_1$ (k = 2, 3, 4, … is a natural number sequence) — harmonics, whose frequencies are multiples of the fundamental frequency ω_1; $\omega_1 = 2\pi/T_z$ — basic frequency determined by the period of the pulses; φ_k is the phase of the oscillations of the k-th harmonic; T_z — period of meshing frequency. The amplitude spectrum of the shock pulse is a set of amplitudes F_k along the frequency axis in accordance with the frequencies ω_k. To simplify the calculation of the amplitude spectrum of the periodic shock pulse described by a sine wave with a duration T and a maximum value F_{max}, place the origin of coordinates at the point of maximum impulse (fig. 6).

In this case, the shape of the shock pulse is expressed cosinusoidally function

$$F(t) = F_{max} \cos\frac{\pi}{T} t \quad \left(-\frac{T}{2} \le t \le \frac{T}{2} \right).$$

Range of amplitudes F_k is determined by the expression

$$F_k = \frac{2}{T_z} \int_{-T_z/2}^{T_z/2} F_{max} \cos\frac{\pi}{T} t \cos\omega_k t \, dt,$$

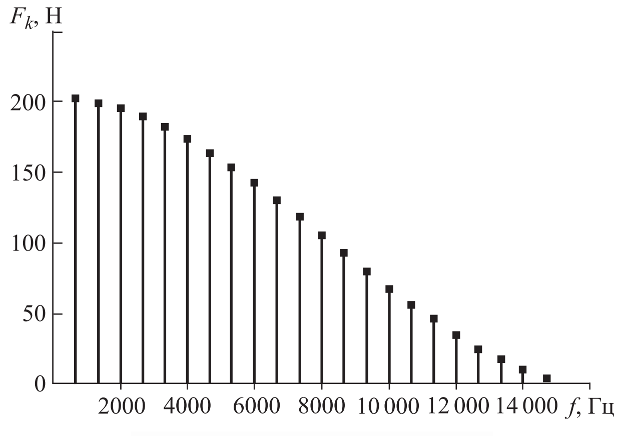

Fig. 7. Range periodic shock pulse

or because of the symmetry relative to the vertical axis and its changes within the period $\pm T_z/2$ from $-T/2$ to $T/2$

$$F_k = \frac{4T}{\pi T_z} F_{max} \frac{\cos(\omega_k T/2)}{1-(\omega_k T/\pi)^2}.$$

Spectrum of oscillations periodically acting shock pulse is discrete and represents the set of harmonic oscillations at the fundamental frequency ω_1 determines the discrete intervals (fig. 7).

Calculated values of the contact stresses in the pitch point. Calculation of contact stresses in the pitch point of the spur gear of the mobile machine is produced according to GOST 21354–87 [8].

The calculated contact stress σ_H

$$\sigma_H = Z_E Z_H Z_\varepsilon \sqrt{\frac{P_{ст}}{bd_1}\frac{u+1}{u}} \sqrt{K_A K_{HV} K_{H\beta} K_{H\alpha}}, \text{ MPa},$$

where d_1 is the pitch diameter of the gear; $P_{ст}$ — static circumferential force on a pitch diameter; b — width of the ring gear; u — gear ratio; K_A — coefficient taking into account the external dynamic load; K_{HV} — coefficient taking into account the dynamic loads arising in the toothing; $K_{H\beta}$ — coefficient taking into account the uneven distribution of load along the length of the contact lines; $K_{H\alpha}$ — coefficient taking into account the distribution of the load between the teeth; Z_E — coefficient taking into account the mechanical properties of materials paired gear wheels, MPa$^{1/2}$; Z_H — coefficient taking into account the shape of the mating surfaces of the pole teeth in gear; Z_ε — coefficient taking into account the total length of the contact lines. For spur gears

$$Z_\varepsilon = \sqrt{\frac{4-\varepsilon_\alpha}{3}},$$

i.e. largely is determined by the contact ratio of the gearing.

A dynamic model of propagation of elastic waves in a drive gear mechanism, caused by the shock pulse in a gear pair. When considering the question of the relationship of the parameters of the impulse in spur gears with their vibration characteristics were used the notion of technical diagnostics, acoustic channel [9]. The problem is formulated mathematically as follows: let the input of the system (fig. 8) has a shock pulse sequence corresponding to the punches in any gears (L — acous-

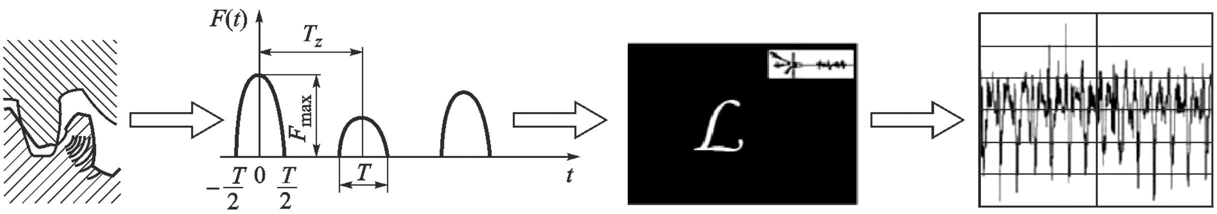

Fig. 8. Logic diagram for determining the relationship the parameters of the shock pulse and vibration signal

tic channel of gears), and the output of the vibration sensor system interprets the signal $w(t)$.

Thus, knowing the function $H(\omega)$ of the amplitude-frequency characteristics of the channel, the spectrum amplitude of the input signal (shock pulse) $S_P(\omega)$ one can determine the spectrum of the output signal $S_W(\omega)$, entering a vibration sensor

$$S_W(\omega) = H(\omega)S_W(\omega).$$

Based on the consideration of acoustic channels was carried out to develop methods for the assessment of existing efforts in a gear meshing based on calculation and experimental dependences of the internal dynamic component of the load in the mates and values vibrating impulses measured at the buildings of the high-speed gear units of the mobile machines [3].

The propagation of elastic waves in the mechanism described by the differential equations of the second order partial derivatives. The extreme difficulty of solving these equations is due to the inability to correctly formulate the boundary and initial conditions describing the state of the surface mechanism. In addition, the study is complicated by the fact that in elastic solids are excited elastic waves of different types: bending, surface, longitudinal, transverse, etc., as well as having problems with the propagation of elastic waves due to the existing mechanism joints, gaps, strips between the parts. Significant simplification of the solution of the problem is obtained by considering the propagation of elastic waves by a point source in an unbounded medium. The propagating wave is a spherical wave process and is determined by only one variable — the coordinate r of an arbitrary point of the mechanism from the origin. For a point source is taken mating pair of teeth, in which an impulse is generated.

The process of propagation of elastic waves is described by a velocity potential, represented in the form

$$\psi(r,t) = \frac{A}{r} e^{i(\omega t - kr)},$$

where $i = \sqrt{-1}$ is the imaginary unit; ω — circular frequency, the constant of integration A can be found from boundary conditions: equality of amplitude values of the expressions for the stresses in a spherical cavity $r = r_0$.

Wave process, caused by the shock pulse in the toothing, can be investigated as follows. The function of the shock pulse is decomposed into harmonic components, and discusses the vibrations of an elastic medium of the mechanism caused by each component. Thus methodically solves the problem of establishing a connection between the parameters of the impulse in gears spur gears and vibration signal perceived by the vibration sensor.

Speed V_r, acceleration a_r particles of an elastic medium, the stress σ_r acting in a plane perpendicular to the radius r, connected with the potential $\psi(r, t)$ by ratios [9]

$$V_r = -\frac{\partial\psi}{\partial r}, \quad a_r = -\frac{\partial^2\psi}{\partial t \partial r}, \quad \sigma_r = \rho\frac{\partial\psi}{\partial t}.$$

and using the value F_k-th harmonic component of the shock pulse corresponding to propagation in the gear housing of the shock wave with circular frequency $\omega = \omega_k$, defined by the formula [10],

$$F_k(t) = F_{k\max}\cos\omega_k t,$$

The amplitude values of the accelerations of points of reducer defined by the coordinate r and the corresponding harmonic components of the shock pulse are found from the expression

$$a_{rk\max} = \frac{F_{k\max} r_0 \sqrt{1 + (\omega_k / c)^2 (\gamma r)^2}}{S_b \rho \gamma r^2},$$

where γ is the coefficient taking into account the lengthening of wave propagation due to the presence of voids and joints in the mechanism; c — the speed of propagation of spherical waves in elastic media; S_b is the surface area of a spherical cavity r_0, is assumed equal to the area of contact between mating teeth, ρ is the density of the material is geared mechanism [11].

Knowing the amplitude values of the accelerations of the points of the gearbox, it is possible to theoretically estimate the value of acceleration perceived by the vibration sensor mounted on the gearbox housing at the point with coordinate r. The magnitude of the RMS value of acceleration is determined by the values of the accelerations $a_{rk\max}$ corresponding to the harmonics excited by the shock pulse in the toothing, the ratio [12]

$$a_{\text{скз}} = \sqrt{\frac{1}{2}\sum_{k=1}^{\infty} a_{rk\max}^2}.$$

Hence the amplitude of the shock pulse in a conjugate pair of teeth

$$F_{\max} = \frac{\pi T_z S_b \rho\gamma r^2 a_{\text{скз}}}{\beta_y r_0 T} \times$$
$$\times\left\{2\sum_{k=1}^{\infty}\left[1+\left(\frac{\omega_k}{c}\right)^2(\gamma r)^2\right]\left[\frac{\cos}{1-(\omega_k T/\pi)^2}\right]^2\right\}^{-1/2}.$$

The resulting research analytical presentation of the shock pulse to obtain its spectrum, which, in turn, allows the comparison with the spectrum of real vibration signal of mechanism to identify those harmonic components that are multiples of subcool frequency, which are within the scope of the mechanism resonance and excite in him an intense vibrations. On the changes of the values of these components in the process of operation can break damage gear.

The determination of residual life of gears according to the results of the vibration monitoring. The next step was to develop a method of predicting residual service life of critical elements of mobile cars drive in operating conditions based on the monitoring of the actual loads in the mesh of gears [5].

The amplitude of the shock pulse in a conjugate pair of teeth is determined experimentally according to the results of the vibration monitoring; then formed the discrete spectrum of oscillations periodically acting shock pulse and, consequently, the set of harmonic oscillations, determining the load in mesh; further calculated actual circumferential force and contact stress σ_{Hi}; the amount of consumption of the resource ΔQ_{Hi} is determined for each of the i-th interval achievements ΔSi mobile machine during its operation.

The resource consumption of gear on each interval operating time of the mobile machine is calculated by the formula

$$\Delta Q_{Hi} = \sigma_{Hi}^{q_H} N_i \ \text{МПа}^{q_H},$$

where $q_H = 6$ is the exponent of s–n curve when calculating the teeth on the contact endurance; N_i is the number of loading cycles of the Central wheel.

Residual life $R_{\text{ост}\, i}$ on the i-th interval is

$$R_{\text{ост}i} = R_H - \sum_{i=1}^{i}\Delta Q_{Hi}, \ \text{MPa}^{q_H},$$

where R_H is the measure of the carrying capacity of the gear on contact fatigue,

$$R_H = \sigma_{H\lim}^{q_H} N_{H0};$$

$\sigma_{H\lim}$ — limit contact fatigue; N_{H0} is the number of stress cycles corresponding to the inflection of the curve, when calculated on a contact endurance. Residual life in numbers of loading cycles of the teeth during the movement of mobile machines constitute the i-th interval

$$N_{\text{ост}\, i} = R_{\text{ост}\, i}/\sigma_{Hi}^6$$

or the equivalent in kilometers covered

$$S_{\text{ост}\, i} = \frac{2\pi r_k Z_{\text{ц}} N_{\text{ост}i}}{(Z_{\text{к}} n_w)}, \text{ km}.$$

Vibration condition monitoring of technical condition of the planetary gear unit motor-wheel (RMK) dump truck BelAZ in operation. Testing of methodology for residual life assessment was conducted for gear: Central wheel — satellite second stage, the Calculation of contact stresses in a gear pair proceeded on the basis traction and dynamic performance of the dump truck BelAZ capacity 130 tons [13].

Analysis of the mean square values (RMS) accelerations (fig. 9, a) shows that at the time of the dump truck to 200000 km, this value remains almost constant. It further increases, at the same time begin to grow peak accelerations and PEAK-factor (fig. 9, b).

As between the amplitude of the shock pulse and the peak value of vibration accelerations there is a linear relationship, with increasing peak values increases the dynamic factor K_{Hv}. The table gives the values K_{HVi}, defined according to the results of vibration monitoring, and designed for them contact stresses σ_{Hi} values usage ΔQ_{Hi} for each i-th interval achievements ΔS_i truck during his working career on the rise. The number of loading cycles N_i of Central wheel when passing a truck route ΔS_i is calculated by the formula

$$N_i = \frac{\Delta S_i}{2\pi r_{\text{к}}}\frac{z_{\text{к}}}{z_{\text{ц}}} n_w$$

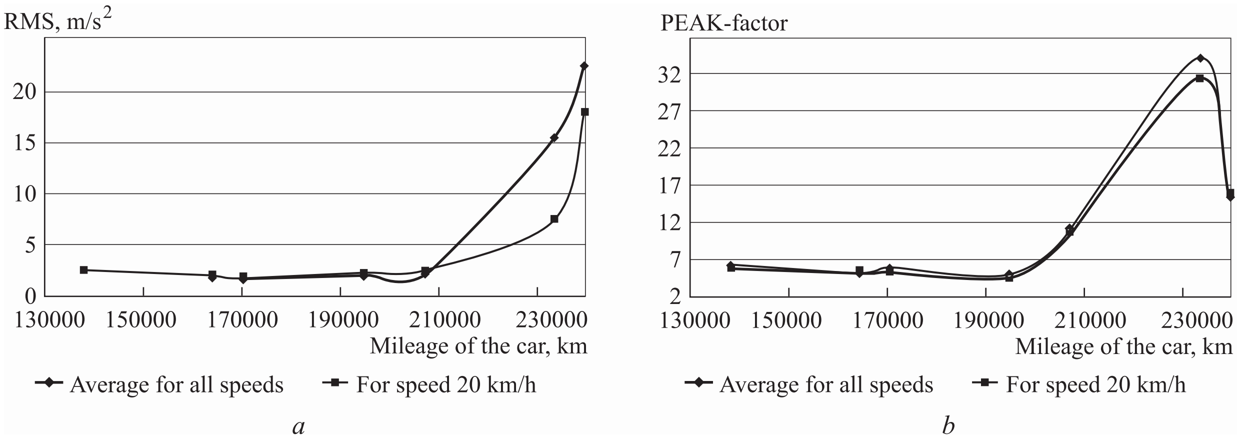

Fig. 9. The Dependence of the RMS (*a*) and PEAK factor (*b*) acceleration from mileage

Expenditure resource ΔQ_{Hi} for each *i*-th operating time interval ΔS_i dump truck

The *i*-th interval developments	1	2	3	4
Operating ΔS_i, km	0–77 924	77 924–82 846	82 846–93 423	93 423–95 804
σ_{Hi}, MPa	1 221	1 246	1 522	1 569
Number of cycles, N_i	$0{,}9812{\cdot}10^8$	$0{,}062{\cdot}10^8$	$0{,}132{\cdot}10^8$	$0{,}03{\cdot}10^8$
The consumption of the resource, Q_{Hi}	$3{,}25{\cdot}10^{26}$	$0{,}232{\cdot}10^{26}$	$1{,}643{\cdot}10^{26}$	$2{,}581{\cdot}10^{26}$

where $z_ц$ — number of teeth of the Central wheel, $z_к$ — number of teeth of the crown wheel of the second stage, n_w is the number of satellites, $r_к$ is the radius of the wheels of the truck.

Next, calculate the resource consumption of gear at each operating time interval of the truck by the above formula. The results of the calculation are summarized in the table.

Calculations show that the residual resource was equal to zero when the total mileage (accounting for the motion of the descent in career and transport mode) $S = 2{,}281{\cdot}10^5$ km. Disassembly of the gearbox with the replacement of one of the satellites due to the chipped tooth has confirmed the preliminary diagnosis (fig. 10).

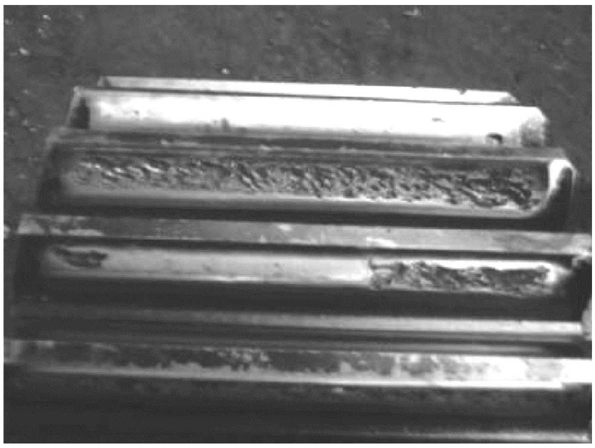

Fig. 10. Damaged work surfaces of the teeth of the Central gear of the second row RMK after run 238 000 km

Conclusion. Establishing relationships between the parameters of the impulse and vibroacoustic signal, and a comparison of their spectral characteristics allow us to identify harmonic components that are multiples of subcool frequency, which coincides with the region of the resonance mechanism and excite in him an intense vibrations. On the changes of the values of these components in the process of operation can break damage gear.

The main advantages of the developed methodology for the technical condition assessment and prediction of residual life of gear — carrying out vibration monitoring and diagnostics of the drive gears in operation. The use of such systems in the farms will allow us to move from preventative system maintenance automotive vehicles and equipment to service on their actual condition, to improve the quality of equipment, greatly reduce the cost of its repair.

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