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COMPUTATIONAL AND EXPERIMENTAL METHOD FOR ESTIMATING THE RESIDUAL LIFE OF GEARS BASED ON VIBRATION MONITORING DATA

A computational and experimental method for estimating the residual life of gears in the inter-repair period is proposed. The method is based on the main provisions of the theory of vibration-pulse diagnostics of spur gears, developed at the OIM of the National Academy of Sciences of Belarus. The main performance criteria are the contact and bending endurance of the teeth. Accounting for changes in the value of the coefficient of internal dynamic load in the gearing during the operation of gears is carried out on the basis of vibration analysis during their vibration monitoring. The proposed method makes it possible to estimate the residual life of the gear mechanism at any stage of operation, providing the opportunity to organize the transition from planned preventive maintenance of machines to maintenance according to the actual condition. An example of estimating the residual life of a gear pair that limits the reliability of a two-row planetary gear motor-wheel of a heavy-duty dump truck is given.

Keywords**:** gear transmission; vibration monitoring; residual life; inter-repair period; internal dynamic load in engagement

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РОЗРАХУНКОВО-ЕКСПЕРИМЕНТАЛЬНИЙ МЕТОД ОЦІНКИ ЗАЛИШКОВОГО РЕСУРСУ ЗУБЧАСТИХ ПЕРЕДАЧ ЗА ДАНИМИ ВІБРОМОНІТОРИНГУ

Запропоновано розрахунково-експериментальний метод оцінки залишкового ресурсу зубчастих передач у міжремонтний період. Метод базується на основних положеннях теорії вібраційно-імпульсного діагностування прямозубих зубчастих передач, розробленої в ОІМ НАН Білорусі. Основними критеріями працездатності приймаються контактна і згинна витривалість зубів. Урахування зміни величини коефіцієнта внутрішнього динамічного навантаження в зачепленні у процесі експлуатації передач здійснюється на основі аналізу вібрацій при їх вібромоніторингу. Запропонований метод дозволяє оцінити залишковий ресурс зубчастого механізму на будь-якій стадії експлуатації, забезпечуючи можливість організувати перехід від планово-попереджувального обслуговування машин до обслуговування за фактичним станом. Наведено приклад оцінки залишкового ресурсу зубчастої пари, що лімітує надійність дворядного планетарного редуктора мотор-колеса великовантажного самоскида.

Ключові слова**:** зубчаста передача; вібромоніторинг; залишковий ресурс; міжремонтний період; внутрішнє динамічне навантаження в зачепленні

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РАСЧЕТНО-ЭКСПЕРИМЕНТАЛЬНЫЙ МЕТОД ОЦЕНКИ ОСТАТОЧНОГО РЕСУРСА ЗУБЧАТЫХ ПЕРЕДАЧ ПО ДАННЫМ ВИБРОМОНИТОРИНГА

Предложен расчетно-экспериментальный метод оценки остаточного ресурса зубчатых передач в межремонтный период. Метод базируется на основных положениях теории вибрационно-импульсного диагностирования прямозубых зубчатых передач, разработанной в ОИМ НАН Беларуси. Основными критериями работоспособности принимаются контактная и изгибная выносливость зубьев. Учет изменения величины коэффициента внутренней динамической нагрузки в зацеплении в процессе эксплуатации передач проводится на основе анализа вибраций при их вибромониторинге. Предложенный метод позволяет оценить остаточный ресурс зубчатого механизма на любой стадии эксплуатации, обеспечивая возможность организовать переход от планово-предупредительного обслуживания машин к обслуживанию по фактическому состоянию. Приведен пример оценки остаточного ресурса зубчатой пары, лимитирующей надежность двухрядного планетарного редуктора мотор-колеса большегрузного самосвала.

Ключевые слова**:** зубчатая передача; вибромониторинг; остаточный ресурс; межремонтный период; внутренняя динамическая нагрузка в зацеплении.

Introduction. Relevance of research. Currently used methods that allow calculating the residual life of a gearing taking into account the internal dynamic load in the gearing [1] do not take into account its changes during operation, which leads to an overestimation of its resource. In addition, the use of the calculated dependences used in the design of gears by introducing correction factors when assessing the residual life of gears in operation is only a partial solution to this problem, since it does not take into account a large variety of factors that affect the bending and contact endurance of the teeth, and does not take into account the change in these factors over time [2]. On the other hand, maintenance based on the actual condition requires both the availability of accurate values of changes in the indicators of this state during the operation of gears [3, 4], and methods for determining the residual life of gear pairs based on these indicators.

Modern experience in diagnosing the condition of gears shows that of the existing methods of non-selective assessment of their residual life (predicting the resource by the mass of wear particles in oil, loss of efficiency, changes in the operating temperature of the most loaded components, backlash, etc.), the most promising are methods for controlling the characteristics of drive mechanisms by monitoring vibration parameters [5, 6].

Of particular importance are the issues of resource forecasting in the operation of unique and expensive technically complex products, the failure of which is associated with significant economic losses, accidents and other technogenic consequences (heavy dump trucks and other mining equipment, equipment of a continuous technological cycle, railway transport, etc.).

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Problem statement. It is possible to estimate the residual service life of spur gears and bearing units of new gearboxes during operation by applying the method of vibration-pulse diagnostics developed and tested in the OIM of the National Academy of Sciences of Belarus [7]. However, during repairs, as a rule, only the faulty parts are replaced with new ones, while the other part remains functioning with a partially exhausted resource, while the vibration characteristics of the gearbox may also change. Therefore, in the operational diagnostics of the drive, there is a problem of post-repair assessment of the impact of new elements on the resource indicators of each of the gears and the gearbox as a whole, and only an individual method of assessing the condition of each of the gears will increase the reliability of the transmission on the basis of timely preventive measures. At the same time, the ratio between the accumulated wear since the beginning of operation and the permissible wear according to the technical conditions allows you to predict the remaining re-source and the possible moment of failure of the parts.

For technical systems with long service lives, the assessment of the residual life of the system and the establishment of its uptime on this basis provides the possibility of developing the most effective regulations for the maintenance and repair of gearboxes based on the results of vibration monitoring and monitoring of their technical condition. This will minimize the risks of accidents, optimize maintenance schedules, increase the time between repairs, and ensure safety management. In addition, with an individual approach to predicting the failure time of transmission gears, the necessary spare parts can be pre-pared in a timely manner, which will also help to reduce the downtime of equipment in repair. The use of such methods will allow you to switch to modern progressive methods of maintenance and increase the coefficient of technical readiness of the fleet.

Earlier, the authors obtained analytical dependences that establish the relationship between the vibration parameters on the bearing unit of the gear mechanism and the load in the gear gearing [7], which allow us to estimate the level of loading and contact stresses in the gearing based on the results of measuring the magnitude of the vibration pulses [8]. On this basis, a method for predicting the exhausting of the transmission resource in operation has been developed [9]. However, during the repair of the gear mechanism with the replacement of worn or damaged gears with new ones, the working conditions and the redistribution of loads between the parts of the mechanism may change. This raises the question of how to assess the impact of repairs on the resource of the remaining elements, how will change the resource of the gearbox as a whole and how to determine it.

A computational and experimental method for estimating the residual life of a gear train in the inter-repair period.Basic provisions. The method of predicting the residual life of gears is based on periodic vibration monitoring of their technical condition during operation. In this case, the maximum value (peak) of the vibration acceleration is determined from the time realizations of the vibration accelerations to calculate the current value of the K_V coefficient, which takes into account the internal dynamic load in the gear engagement. There is a linear relationship between the value of the internal dynamic load and the values of vibration accelerations (in m/s^2). Consequently, with the growth of vibration accelerations, the coefficient of dynamism K_V increases proportionally [7].

When calculating the residual life of gear, it is assumed that the main criteria for their working capacity are the contact and bending strength of the teeth. GOST 21354-87 [1] provides dependencies that allow calculating the residual life of the gear train, taking into account the change in the internal dynamic load in the gearing during operation. At the same time, it should also be taken into account that if the same wear resistance of the gear materials is laid down in the manufacture of the gear transmission, the wear rate of the gear teeth and the wheel is not equal, since the gear wears out faster than the wheel in a proportion equal to the gear ratio [10].

In the proposed method, at the initial stage of operation, the value of the dynamic coefficient K_{V0} of the gear train is determined by calculation or experimentally at the run-in stage [7]. When using the calculation method, the maximum probability difference Δ_0 between the largest normalbase pitch of one gear wheel and the smallest of the other is calculated by means of the tolerance tables for the gear drive.

Further calculation of the contact and bending stresses of the gear train during operation involves periodic recalculation of the value of the internal dynamic load coefficient K_V. The process of determining this coefficient is based on the fact that to calculate the current value of the K_{Vi} coefficient, after each i-th measurement of the vibration accelerations of the gear, the maximum value (peak) of the vibration acceleration is determined. Processing of the results of measuring the vibration accelerations of the gear train to form a set of implementations of the vibration accelerations of the gear and wheel shafts is based on the application of the synchronous averaging algorithm [7].

When the resource of one of the gear links is exhausted, depending on the cost and labor capacity of the repair work, the task of replacing the entire transmission or replacing a failed element (gear or wheel) is solved.

For example, we consider the option of replacing one of the gears. Then the measure of the carrier capacity of a gear wheel paired with it after repair is determined by a value equal to the difference between the values of two measures – the measure of the carrier capacity of this wheel before the start of operation and the measure of damage to its teeth during operation.

Based on the results of vibration acceleration measurements during the before-repair period of the gear transmission operation, a graph is plotted (Fig. 1) of the dependence of the internal dynamic load $F_{\max}$ on the maximum value of the vibration acceleration $a_{\max}$. The construction of the schedule begins after the completion of the initial stage of operation (running-in), during which the vibration parameters are stabilized [11].

The graph shown in Fig. 1 is used in the future to determine the coefficient of internal dynamic load K_V in the inter-repair period of operation. At the same time, the value of the internal dynamic load $F_{\max i}$ is calculated from the measured value of the maximum value of the vibration acceleration $a_{\max i}$ from the graph and the current value of the K_{Vi} coefficient is calculated.

It is assumed that the coefficients of internal dynamic load when calculating the teeth for contact and bending strength are equal to each other. It should be noted that the calculated contact stresses also have the same values for the pinion teeth and the gear wheel.

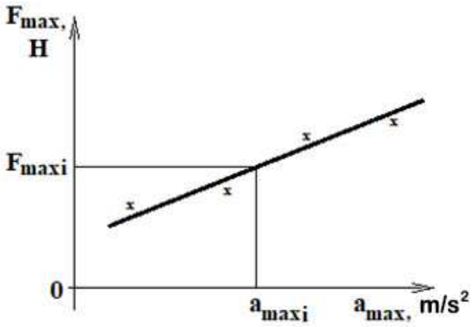

Fig. 1 – Dependences of the internal dynamic load $F_{\max}$ on the maximum value of the vibration acceleration $a_{\max}$

The need to start the procedure for assessing the residual life of gears (the end of the first operating interval) is determined by the results of vibration monitoring of their technical condition, the methodological foundations of which are set out in [7]. Clarification of the resource of gears is carried out after a steady transition of the value of their vibration from the zone of "normal vibrations" to the zone of "increased vibrations". The duration of the subsequent operating intervals is determined in such a way that at least two control measurements are carried out between the moment when the upper threshold values of the "normal vibration" zone are exceeded and the lower threshold values of the "critical vibration" zone are reached. When determining the forecast values of $\sigma^{q_H}_{H,F\,1,2(i+1)\ predict}$ related to the "subsequent – (i+1)" operating time interval (formula 18, Table 1), extrapolation methods are used (see, for example, [12, 13]).

The calculation algorithm is presented in Table 1.

In the formulas under consideration, the following indices are used:

H – to the calculations of the teeth for contact strength,

F – to the calculations of the teeth for bending strength,

1 – to the pinion,

2 – to the gear wheel,

(0) – initial stage of operation,

(i) – i-th operating interval.

Table 1 – Algorithm for estimating the residual life of the gear train in the inter-repair period

№	Parameter	Equation	Notes
The initial stage of operation– running-in (new gear, operating interval i=0)			
1	The highest probability difference between the gear and wheel normal base pitch	$\Delta_0 = 1{,}2 f_{Pb}$	f_{Pb}– limit deviation of the normal base pitch (GOST 1643-81) depending on the accuracy grade
2	Internal dynamic load of the new gear train	$F_{\max(0)} = \delta_{H,F}\sqrt{(\Delta_0 + \delta_c)Vb_w\sqrt{\dfrac{a_w}{u}}}$	$\delta_{H,F}$ – influence coefficients of gearing kind; δ_c– deformations of a pair of teeth engaged before impact, microns; b_w–facewidth, mm; V – peripherial speed of the sun gear, m/s; a_w– center distance, mm; $u = z_2/ z_1$ – gear ratio; z_1, z_2 – number of teeth of the pinion, wheel
3	Allowable stress number (contact and bending)	$\sigma_{F\lim}$ $\sigma_{H\lim}$	Reference data
4	Number of load cycle corresponding to the transition from limited life to long life	$N_{F\lim}$ $N_{H\lim}$	Reference data
5	Contact stress of the teeth of the new gear train	$\sigma_{H(0)} = Z_E Z_H Z_\varepsilon \sqrt{\dfrac{F_t}{b_w d_1}\dfrac{u+1}{u}}$	Z_E – elasticity factor; Z_H – zone factor; Z_ε – contact ratio factor; d_1 – the reference diameter of the pinion, mm
6	A measure of the carrier capacity for a pinion and wheel by contact and bending	$R_{H,F1,2} = (\sigma_{H,F\lim 1,2})^{q_{H,F1,2}}\ N_{H,F\lim 1,2}$	$q_{H,F\,1,2}$ – indicators of the degree of fatigue curve when calculating the pinion and wheel teeth for contact and bending endurance
7	The value of the maximum vibration acceleration of the gear train after running-in	$a_{\max(0)}$	According to vibration monitoring data
8	Establishing a dependency	$F_{\max} = f(a_{\max})$	Fig. 1

End of table 1

№	Parameter	Equation	Notes
	After the end of the *i*-th (*i* = 1,2, ...) operating interval		
9	Number of gear and wheel mesh contacts	$N_{1,2\,(i)}$	Operating time according to operating data
10	The value of the maximum vibration accelerations	$a_{\max\,(i)}$	According to vibration monitoring data
11	Vibration growth factor	$K_{B(i)} = a_{\max\,(i)} / a_{\max\,(0)}$	
12	The value of the internal dynamic load in the gearing	$F_{\max(i)} = K_{B(i)} F_{\max(0)}$	
13	Internal dynamic load factor	$K_{V(i)} = K_{HV(i)} = K_{FV(i)} = 1 + F_{\max(0)} / F_t$	F_t– nominal tangential load at the reference cylinder, N
14	Load factors	$K_{F(i)} = K_A K_{FV(i)} K_{F\beta} K_{F\alpha}$ $K_{H(i)} = K_A K_{HV(i)} K_{H\beta} K_{H\alpha}$	K_A – application factor; $K_{H,F\,\beta}$ – face load factor; $K_{H,F\,\alpha}$ – transverse load factor (bending and contact stress)
15	Calculated contact stress	$\sigma_{H(i)} = \sigma_{H(0)} \times \sqrt{K_{H(i)}}$	
16	Calculated bending stress of the pinion and wheel	$\sigma_{F1,2\,(i)} = \frac{F_t}{b_w m} K_{F(i)} Y_{FS1,2} Y_\beta Y_\varepsilon$	$Y_{FS1,2}$ – tip factor equal accounts of the pinion and wheel; Y_β – helix angle factor; Y_ε – contact ratio factor
17	Measure of damage to the teeth of the pinion and wheel by contact and bending stresses	$Q_{H,F1,2(i)} = N_{1,2(i)} (\sigma_{H,F1,2(i)})^{q_{H,F\,1,2}}$	
18	Predicted stress values	$\sigma^{q_H}_{H,F\,1,2(i+1)\;predict}$	By extrapolating function values $a_{\max} = f\,(N_{1,2})$ or $\sigma_{F,H\,1,2} = f\,(N_{1,2})$
19	Residual operating life of the gear and wheel according to contact and bending stresses *	$N_{H,F\,1,2ост\,(i)} = \frac{R_{H,F1,2} - \sum_{k=1}^{k=i} Q_{H,F1,2(i)}}{\sigma^{q_H}_{H,F1,2(i+1)\;predict}}$	
20	Establishing a dependency	$F_{\max} = f\,(a_{\max})$	Fig. 1
21	Checking the exhaustion of the gear and wheel resource by contact and bending stresses, making a decision on repair	$N_{H,F\,1,2\,RL\,(i)} \le 0$	If there is no need for repair, then the algorithm is performed with item 9. In the case of replacing a gear pair – with item 1. In the case of replacing one of the gears – with item 6 **

Notes:
* In some cases, it is more convenient to convert the remaining resource into hours or kilometers of mileage.
** If one of the gears is replaced:

1) for the remaining gear $R_{H,F1,2} = (\sigma_{H,F\lim 1,2})^{q_{H,F1,2}} N_{H,F\lim 1,2} - \sum_{k=1}^{k=i} Q_{H,F1,2(i)}$

2) the new value of the internal dynamic load $F_{\max 0}$ is based on the measured value $a_{\max i}$ from Fig. 1

Application example. Let us consider the calculation of the residual life of the gear pair (the sun gear – planet gear of the planetary gearbox second stage), which limits the reliability of the two-row planetary gearbox of the motor-wheel of the BELAZ heavy-duty dump truck.

1. Number of teeth:
   sun gear z_1 = 22,
   planet gear z_2 = 29;
   ring gear z_3 = 83;
2. Normal module:
   m = 10 mm;
3. Face width:
   planet gear b_{w2} = 143 mm,
   sun gear b_{w1} = 164 mm;
4. Helix angle:
   β = 0;
5. The gearing accuracy grade:
   (according to GOST 1643) – 8;
6. Number of planet gears: n_w = 3;
7. Peripherial speed of the sun gear:

V_1 = 1.21 m/s;
8. Center distance:
a_w = 266.99 mm;
9. Peripherial torque on the sun gear:
T_1=12625 N m;
10. Peripherial force on the reference diameter of the sun gear:
F_{t_z1} = 114750 N;
11. Steel grade:
20X2H4A,
12. Hardening treatment method:
cementation;
13. Tooth surface hardness:
sun gear and planet gear – 627 HB (62 HRC);
14. Dump Truck wheel radius:
r_k = 1.43 m.

Vibration acceleration measurements were carried out at the speed of the dump truck movement of 20±5 km/h, corresponding to the movement of the loaded car on the rise, since the accumulation of damage (resource exhausting) occurs in this mode. The distance that the dump truck worked in the lifting mode, for the example under consideration, is about 40% of the total operating time.

Based on the results of vibration acceleration measurements during operation, a graph was plotted (Fig. 1) of the dependence $F_{\max} = f(a_{\max})$.

Table 2 shows the results of calculating the amount of resource consumption for each *i*-th operating interval of the dump truck.

As can be seen from the table, at the third interval, the dump truck run is equal to 226058-207123=18935 km. Thus, almost at the end of the third interval, the resource of the transmission sun gear is exhausted for contact loading. In this regard, it is recommended to stop the operation of the dump truck and determine the rejection condition by wear and pitting of the working surfaces of the teeth of the gears, identify the presence of chipped teeth, and, if necessary, decide to replace the sun gear.

The total residual run of the planet gear under bending stresses by the end of the third interval is approximately 76,000 – 18935 ≈ 57,000 km. With a monthly run of the dump truck of ≈ 6000 km, this resource will be selected in about 9.5 months, and therefore the replacement of planet gears for technical indications is not relevant for this repair.

Table 2 – Residual gear life for each *i*-th operating interval of the dump truck

Parameter	Designation	Operating time interval		
		1	2	3
Total operating time (mileage) of the dump truck, km	S_i	0–194810	194810–207123	207123–226058
The value of the maximum vibration accelerations, m/s^2	$a_{i\,\max}$	1,16	2,75	23,3
Vibration growth factor	K_B	1	2,37	20
Internal dynamic load factor	K_{HVi}	1,030	1,071	1, 608
Contact load factor	K_{Hi}	1,653	1,719	2,581
Bending load factor	K_{Fi}	1,638	1,703	2,557
Residual life, km of mileage				
By contact stress for the sun gear	S_{RL}	–	81250	21000
By contact stress for the planet gear		–	820000	248000
By bending stress for the sun gear		–	561250	48000
By bending stress for the planet gear		–	792500	76000

Conclusion. A computational and experimental method for estimating the residual life of gear drives in the inter-repair period is proposed, the essence of which is to determine the value of the coefficient of the dynamic component of the load in the engagement of each gear according to the results of vibration monitoring, followed by recalculation of the residual life and identification of the gear drive, which limits the drive life as a whole. The method allows you to evaluate the technical condition and the residual life of gears at any stage of operation of the drive, including after repairs with the replacement of one or more gears.

The practical use of the proposed method makes it possible to review the principles of technical maintenance and repair of gear systems of expensive, technically complex products with a long service life (for example, BELAZ quarry equipment) and to organize the creation of information and control complexes of the "intelligent quarry" type, which ensure the transition from a planned warning system of equipment maintenance to maintenance according to its actual condition, while increasing the efficiency of machine operation and significantly reducing the cost of their maintenance and repair.

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