МЕХАНИКА
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BOGDANOVICH Alexander V., D. Sc. in Eng., Assoc. Prof.
Professor of the Department of Theoretical and Applied Mechanics
E-mail: bogal@tut.by
Belarusian State University, Minsk, Republic of Belarus

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ABOUT ESTIMATION OF THE INFLUENCE OF NON-UNIFORMITY OF INITIAL STATES ON LIFETIME OF FRICTION PAIRS AND TRIBO-FATIGUE SYSTEMS

The article analyzes relationships of the non-uniformity of the initial states and the lifetime of friction pairs and tribo-fatigue systems based on the results of the inspection of the technical condition after operating time and laboratory tests. The non-uniformity of the initial states is determined by the variation within the tolerances of the geometry of the elements due to the manufacturing technology, and its influence on the wear rate, dynamic characteristics and lifetime of mechanical systems is estimated on the example of automobile sliding friction bearing pairs and rolling bearings. A laboratory experiment to assess the performance of a model of a tribo-fatigue system "shaft (steel 45) — roller (25ХГТ (25KhGT) steel)" due to local violation of the shaft geometry during rolling is analyzed. The correlation equations are given that link the integral characteristics of the local process of wear-fatigue damage, based on measurements of the circular contour of the test specimen (coefficients of asymmetry and non-uniformity), with the relative number of cycles until the limiting wear of the tribo-fatigue system "shaft (steel 45) — liner (silumin)" under sliding friction is reached.

Keywords: *friction unit, tribo-fatigue system, bearings, uneven wear, tests, wear-fatigue damage*

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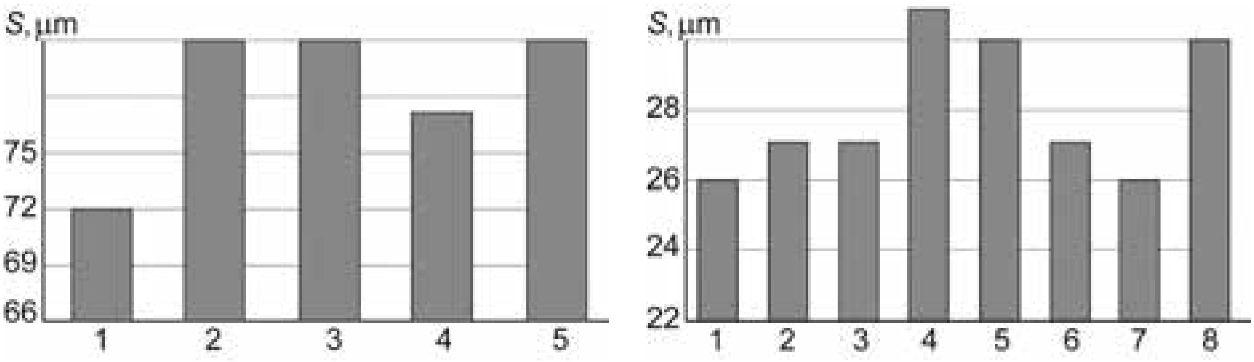
Introduction. A modern machine is, as is well known, a complex dynamic system that includes interconnected aggregates and mechanisms, which, in turn, consist of joints and individual parts. Approximately 20–25 % of all assembly units determine the reliability, durability, safety and efficiency of the machine as a whole. After the same operating time in the same machine interfaces of the same model, as a rule, there is a significant difference in the values of their wear indicators. For example, values of the coefficient of variation V of wear of the similar (identical in design and name) elements of the KamAZ-740 engines of the KamAZ-5410 vehicles after a run of 120,000 km are in the range of 0.15...0.68, and the coefficient of variation of the engine life is from 0.32 to 0.39, which, according to the authors [1], indicates their functional dependence.

The average values of V can be even higher if we consider the totality of the results of micrometering the wear of the similar interfaces of a large group of engines of the same model that have been repaired (Figure 1). The durability of rolling bearings of the same size from the same batch, operating under comparable operating conditions, can differ by 40 times [2, 3].

The non-uniformity of the initial states significantly depends on the micro- and macrogeometry of the treated surfaces (roughness, waviness, angularity, ovality, etc.), which are associated with the finished part with the type and modes of finishing operations, as well as on preliminary mechanical, electromechanical, thermal and other treatments of the material, which is due to the transfer of individual properties of the processed object (blank) from previous operations to subsequent ones [4]. The type of surface treat-



a *b*

Figure 1 — Average values of wear *S* of the crank (*a*) and rod (*b*) journals of the crankshafts of the ZMZ-53 engines of cars with a mileage of 100,000 km [1]: *a* — $S = 78$, $V = 0.61$; *b* — $S = 28$, $V = 0.57$; 1–8 — journal numbers

ment of the friction unit element significantly affects the wear of the counterbody associated with it, which indicates the influence of the technology of preliminary and final treatments on their ability to adapt to each other during the friction process and work stably under specified conditions [4].

The authors of a number of works [5–7] believe that the elimination or reduction of the influence of the most significant factors that cause wear non-uniformity of the similar elements, contribute to a more complete use of the potential life of the main units of the vehicle. At the same time, there are only single publications where the life dependencies of the similar elements on the conditions of their interaction and the initial geometric shape are given.

Thus, the significantly different technical condition of the similar machine elements after a certain period of operation is a consequence of the difference in the initial values of the roughness parameters, deviations in geometric shapes and the location of surfaces, mechanical and thermal stress, working processes related to the design, manufacturing technology and assembly. The individual (for these reasons) technical condition of each pair of friction or tribo-fatigue system determines the corresponding, individual, conditions of friction and wear, the accumulation of wear-fatigue damage.

This article analyzes the relationships between the non-uniformities of the initial states in the geometry of the working surface and the state of the working surfaces of friction pairs and tribo-fatigue systems, determined after the operational time and the results of laboratory tests.

Non-uniformity of the initial states and operational operating time. Let us illustrate the influence of manufacturing technology on the initial technical condition of the friction pair by the example of mechanical grinding of the grooves of the outer rings of ball bearings.

Analysis of the processing accuracy based on the results of waviness measurements of the processed rings showed that with an increase in the working surface runout of the grinding wheels, and, consequently, the imbalance of the "spindle — circle" system, the average values and dispersion of the waviness of the processed rings increase [8]. At the same time, the typical view of the circular patterns of the ring grooves (Figure 2) indicates that the component corresponding to the spindle rotation frequency prevails in the spindle waviness spectrum [8].

The results of experimental studies of the vibrations of ball bearings caused by the waviness of their elements showed [9] that the greatest influence on the vibration level is exerted by the waviness of the balls, and the least — by the waviness of the raceway of the outer ring (Figure 3).

The initial differences in the geometry of the bearing working surfaces largely determine their subsequent non-uniform wear, since vibrations are known to be associated with the interaction dynamics of the contacting working surfaces, and their increase contributes to the deterioration of the macro- and microgeometry of these surfaces during the operation of the tribocoupling (Figure 4), especially when the lubricant is contaminated with abrasive particles.

Apparently, the most significant initial non-uniformity affects the life of aggregates of various machines in which the similar elements are kinema-

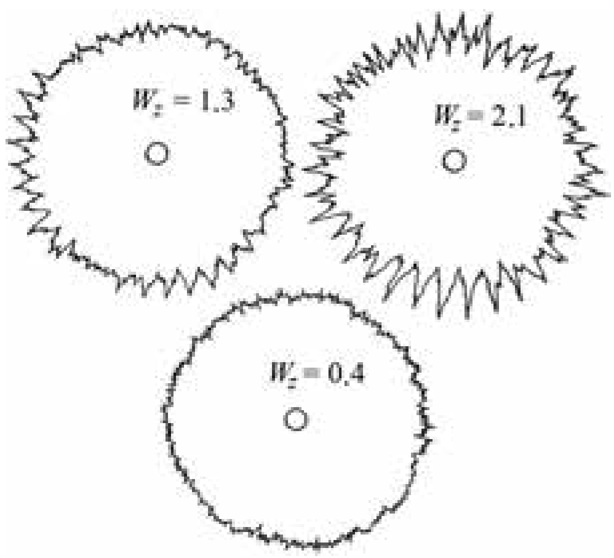


Figure 2 — Typical polar diagram of waviness of the grooves of the outer rings of the ball bearing 305 [8]

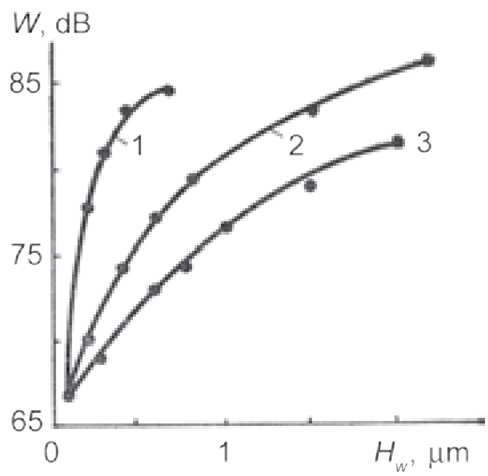


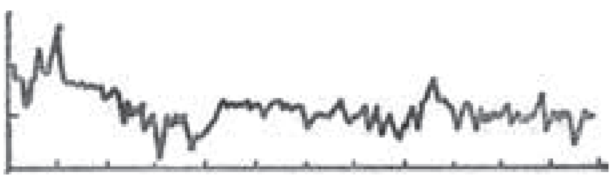
Figure 3 — Influence of the waviness H_w of the balls (1), inner (2) and outer (3) rings on the vibration level W of ball bearings 307 [9]

tically connected. The technical condition of each element directly affects the performance of the others. The corresponding interfaces are difficult to diagnose piecemeal, and a malfunction of one of them often leads to the failure of the entire unit.

As an example, we can consider the change in the wear non-uniformity of individual elements during the operation of an automobile engine. Experiments proved that during operation, the non-uniformity of wear for most of the interfaces of the cylinder-piston group and the crank-connecting rod mechanism of the engine increases. Figures 5, 6 [1] show that the wear non-uniformity (the difference between the maximum and minimum values) of cylinders, piston rings, and connecting rod bearings increases; at the same time, each part retains its position in terms of wear intensity, which was established after running-in. Only for cylinder liners, there is a decrease in the wear rate at the final stage of operation.

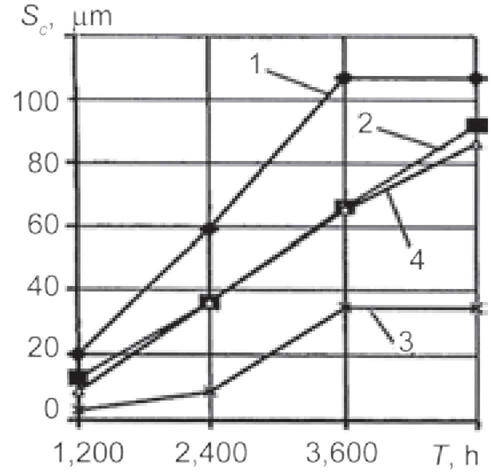
a

b

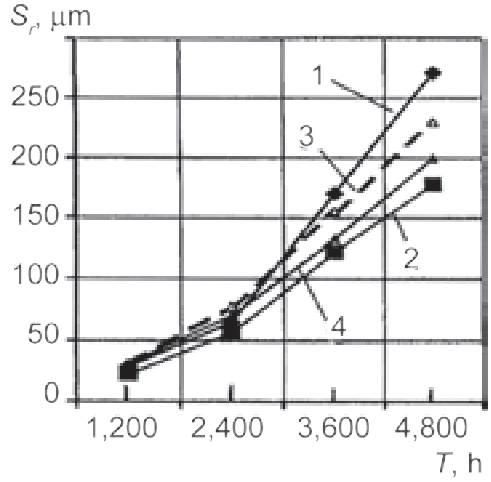


c

Figure 4 — Profile chart of the ball surfaces (vertical magnification 2·10⁴): *a* — ball surface of the new bearing; *b*, *c* — after operation at axial loads of 80 and 200 N in the presence of abrasive particles in the lubricant [10]



a



b

Figure 5 — Dependences of changes in the wear of cylinders (*a*) and piston rings (*b*) of a tractor diesel engine with a power of 27.7 kW at a crankshaft speed of 1900 rpm on the operating time *T* [1]: 1–4 — cylinder numbers

For the analytical description of the relationship between the wear non-uniformity of the similar vehicle elements and the initial non-uniformity of the values of their structural parameters, the dependencies proposed by F.N. Avdonkin are used [11]:

$$S_n = S_{n0} e^{b_n l}, \qquad (1)$$

where $S_n = S_{max} - S_{min}$ is the wear non-uniformity; S_{max}, S_{min} are the maximum and minimum wear values, respectively; S_{n0} is the value of the initial non-uniformity of the gap in the coupling after the run-in stage; b_n is the coefficient of intensification of wear non-uniformity; l is the vehicle mileage.

With the same arithmetic mean S_m values of the initial clearances in the similar bearings in the engine with a lower initial non-uniformity S_{n0} of clearances, reaching the limit value S_l of the clearance set by the engine manufacturer is the most likely event. At the same time, the life of such bearings is higher. The limit value S_l of the bearing clearance is determined experimentally and corresponds to the coupling state, in which the oil pressure in the main line

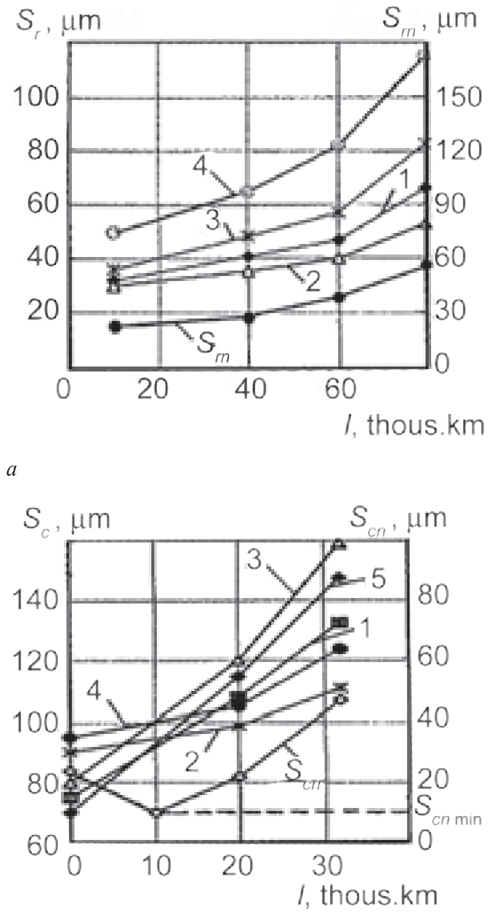


Figure 6 — Dependences of changes in the wear (S_r, S_c) of connecting rod (*a*) and crank (*b*) bearings and their non-uniformity (S_{rn}, S_{cn}) on the vehicle mileage *l* for the 4Ч (4Ch) 9.2/9.2 engine of the GAZ-2401 vehicle [1]: 1–4 — bearing numbers

and the thickness of the lubricating layer are reduced to critical values. Taking into account the non-uniformity of the structural parameters, the maximum allowable gap can be determined by the expression [1]:

$$S_{ln} = S_l - S_m - S_{n0}e^{b_n l}, \tag{2}$$

i.e. the bearing life will decrease not in proportion to the value of the initial non-uniformity S_{n0}, but in an exponential relationship. It is clear that to increase the life, it is necessary to minimize the initial non-uniformity of the structural parameters and shape deviations of the surfaces of the similar couplings. Note that the non-uniformity of the deviations of the shapes (ovality, taper) of the crankshaft journals is also exponentially dependent on the vehicle mileage:

$$\varepsilon_n = \varepsilon_{n0}e^{b_{n\varepsilon} l}, \tag{3}$$

where ε_n, ε_{n0} are the non-uniformity of the deviations of the shapes and positions of the surfaces of the crankshaft journals, respectively, at the current time and after running-in.

For self-unloading interfaces "cylinder liner — piston ring", taking into account the ovality of the liner that develops during the operation of the engine and the action of abrasive dust particles entering the gap between the piston ring and the liner with the fuel-air mixture, the dependence of the wear of the interface on the vehicle mileage is obtained:

$$S_C = \frac{S_L}{1 + ae^{-bl}}, \tag{4}$$

where S_C, S_L are the current and limit values of the liner wear; *a*, *b* are the coefficients that characterize the friction conditions. The validity of this dependence is established according to the data of operational tests of various internal combustion engines [12–14]. For example, here is a diagram of the wear of the working surface of the liner of a KamAZ-740 diesel engine with a mileage of 120,000 km (Figure 7): apparently, the places of maximum wear are not located in the plane of swing of the connecting rod, but at an angle of 20...45° to the axis of the crankshaft, which practically coincides with the results of studies of the deformations of the cylinder liner of this diesel engine during assembly (Figure 8), — the planes of maximum deformations of the liner during assembly and its wear (see Figure 7) are located identically; while the maximum deformations of the liner during assembly exceed the nominal values of ovality and taper by 2...2.5 times [15]. It can be assumed that the wear of the working surface of the liner of the KamAZ-740 diesel engine during operation is a consequence of the ovality that occurred as a result of assembly.

Some results of laboratory tests confirming the influence of non-uniformity of initial states on the durability of the "shaft — roller" and "shaft — liner" systems. The tribo-fatigue system "25ХГТ (25KhGT) steel (roller) — steel 45 (shaft)" was tested for mechano-rolling fatigue. The properties of 25ХГТ (25KhGT) steel were as follows: endurance limit σ_{-1} = 570 MPa, rolling fatigue limit p_f = 3,100 MPa; properties of 45 steel: σ_{-1} = 260 MPa and p_f = 1,760 MPa. A characteristic feature of this system was that the strength of the metal of the shaft is significantly less than that of the roller, so during testing,

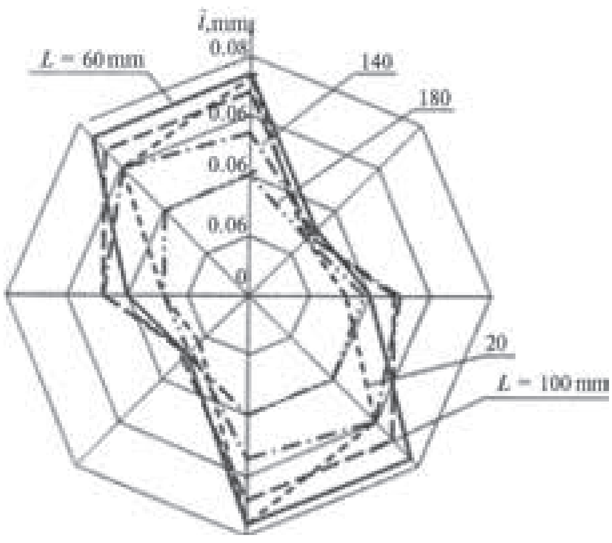


Figure 7 — Diagram of wear *i* of the working surface of the liner in its different sections along the length *L* after a run of 120,000 km [15]

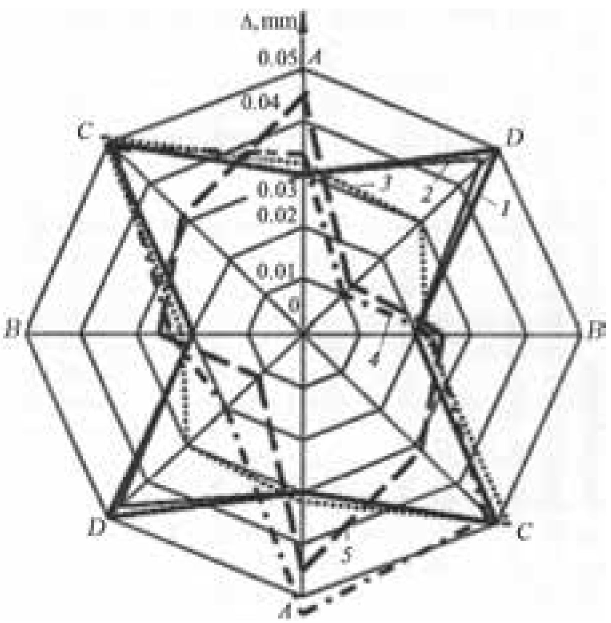


Figure 8 — Diagram of strains Δ of the inner surface of the liner in its different sections along the length *L* after assembly in the block case (1–5 — measurement courses) [15]

residual deformations and damage are detected only in the vicinity of the raceway on the shaft, while the dimensions of the roller remain practically undistorted.

The bending load $Q = 225$ N = const corresponds to the stress amplitude $\sigma_a = 225$ MPa < $\sigma_{-1} = 260$ MPa. The contact load was changed stepwise according to the program shown in Figure 9. The contact endurance limit $p_f = 1{,}760$ MPa was exceeded at the loading stage III.

The movement of the roller along the shaft became unsteady during the transition from the stage VII to the stage VIII of contact loading, i. e. after 700,000 test cycles (see Figure 9, arrow 1). At the stage IX, there was a loss of stability of movement (see Figure 9, arrow 2).

The influence of the manufacturing error of the shaft on the formation of residual surface wave-like damage as a result of the non-stationary process of elastic-plastic deformation in the zone of contact interaction with the roller (the troppi phenomenon) during rolling friction is shown in Figure 10 *a*. With deviations of Δ = 10 μm, the movement of the roller along the shaft was stable during the entire test time ($N_\Sigma = 1.2 \cdot 10^6$ cycles). But if the manufacturing error was Δ = 40 μm, the loss of movement stability occurred approximately at $N \sim 7 \cdot 10^5$ cycles. This means that the geometry of the elements and the accuracy in contact are factors that largely form the conditions for the occurrence of wave-like damage during mechano-rolling fatigue tests.

A special experiment was conducted to assess the effect of a local violation of the shaft geometry on the system's performance. To do this, a patch was performed in the working area of the shaft, which initially initiated the occurrence of vibrations in the system. The results of the rolling fatigue tests are shown in Figures 10 *b* and 11.

During the tests, we studied the time variation not only of the averaged values δ_c of the approach of the shaft and roller axes (Figure 11 *a*), but also of the discrete values δ_c measured separately at each of the 8 points fixed along the perimeter of the raceway (see Figure 11 *b*). In addition, Figure 11 *c* shows a particular implementation of the process δ_c (*t*) for a given individual point (no. 4) on the raceway.

Figure 11 *b* shows that the movement of the roller along the shaft is almost unsteady initially, and this

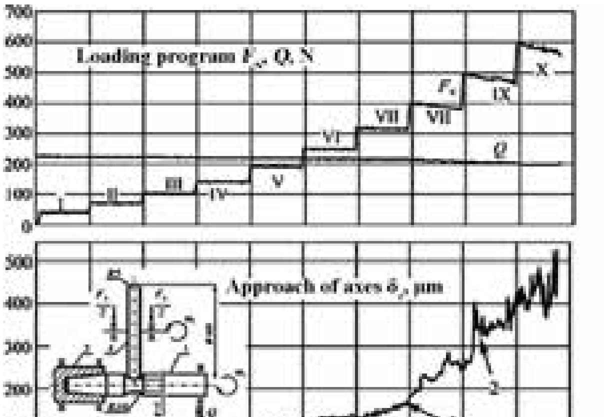
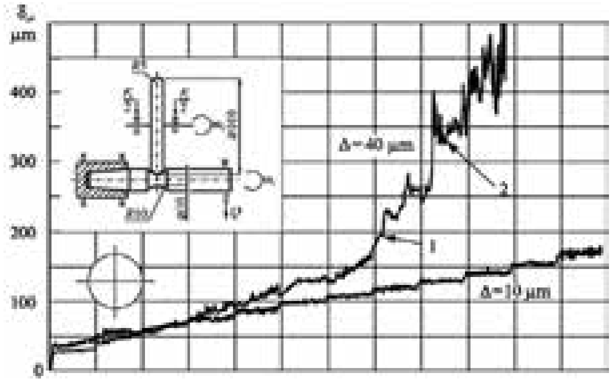
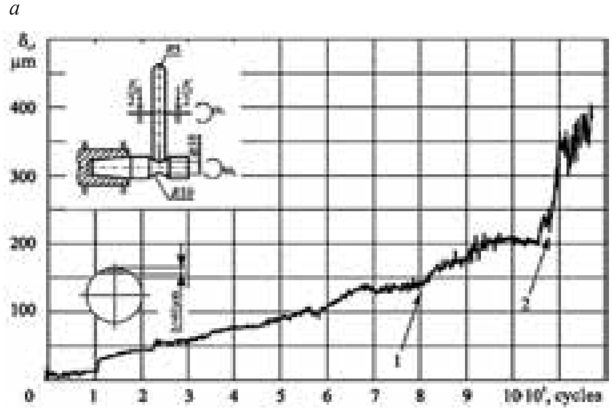


Figure 9 — Test report of the system "roller (25ХГТ (25KhGT) steel) — shaft (steel 45)" for mechano-rolling fatigue [16, 17]

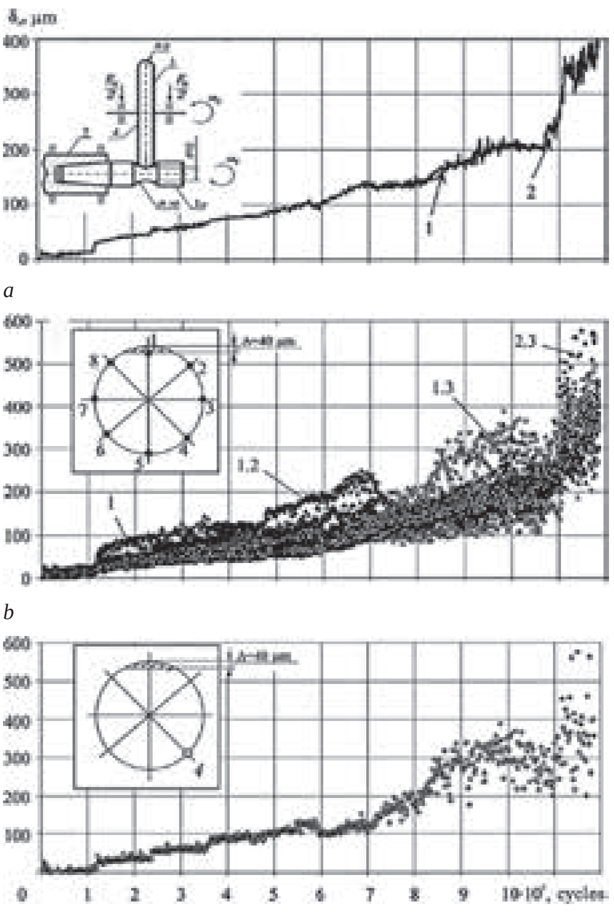


a



b

Figure 10 — Influence of the shaft manufacturing error on the troppi excitation in the system "roller (25ХГТ (25KhGT) steel) — shaft (steel 45)" under mechano-rolling (*a*) and rolling (*b*) fatigue [16, 17]



a

b

c

Figure 11 — Change in value δ_c of convergence of the axes at the rolling fatigue of the system "roller (25ХГТ (25KhGT) steel) — shaft (steel 45)": *a* — averaged values: *b* — values at 8 points; *c* — values at one of the 8 points [16, 17]

unsteadiness increases due to the residual change in the size of the contact area and the local properties of the material along the raceway.

In the vicinity of the patch (point 1), the discrete value δ_c reaches 100 µm by $2 \cdot 10^5$ cycles, and at $6 \cdot 10^5$ cycles it doubles. And this is due to the fact that after a jump on the patch, the roller impacts the metal in the vicinity of the next point 2. Then a similar process of forming wave-like holes occurs already in the vicinity of point 3, and with a durability of ~10^6 cycles, a discrete value δ_c of ~300...400 µm is achieved. It turns out that the residual wave-like damage seems to roll from point to point. When the shock-fatigue process occurs, they form (after $N > 1.1 \cdot 10^6$ cycles) along the entire length of the raceway for a very short test time (~100,000 cycles) and become deep and wide. The total loss of operability was achieved with the number of loading cycles, which is 20 % more than in the case of mechano-rolling fatigue tests (see Figure 10 *a*, *b*), and the contact load reached the value $F_N = 800$ N.

It is known that a change in the elastic properties and density of the material during the accumulation of damage leads to a change in the dynamic characteristics of the system: the frequency and shape of natu-

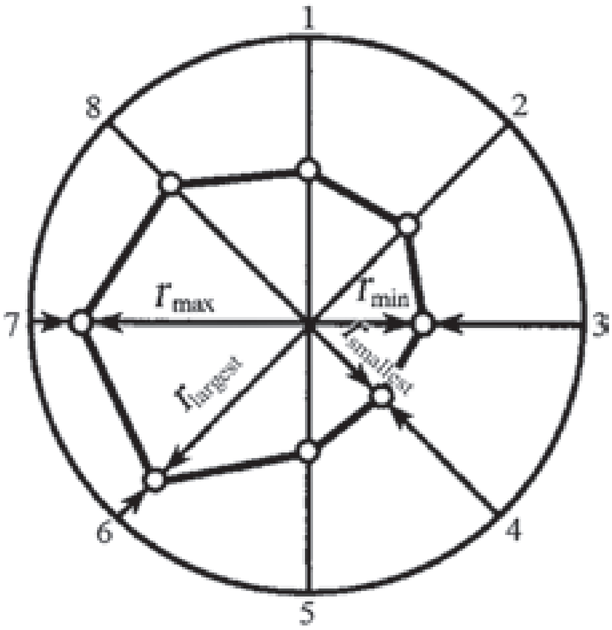


Figure 12 — Diagram of the circular contour of the test sample with the designation of radii for determining the coefficients of asymmetry and non-uniformity: 1–8 — local measurement points

ral vibrations, the frequency and amplitude of forced vibrations. These changes can be estimated by experimental or computational experiments. At the same time, a decrease in the effective modulus of elasticity also leads to an increase in the level of vibration — the amplitudes of forced vibrations increase (this criterion is used for vibration diagnostics of technical systems). The influence of the damage level on the dynamic characteristics of the system is analyzed in detail in [18].

Previously [19], special integral characteristics of the local process of wear-fatigue damage were proposed, based on measurements of the circular contour of the test sample, determined by eight local measurement points (see Figure 11): the asymmetry coefficient:

$$R_a = \frac{1}{4}\sum_{i=1}^{4}\frac{r_{\min(i)}}{r_{\max(i)}}, \tag{5}$$

where $r_{\min}$ and $r_{\max}$ are smaller and larger radii of the same sample diameter, and the coefficient of non-uniformity:

$$\eta_a = \frac{r_{\text{smallest}}}{r_{\text{largest}}}, \tag{6}$$

where r_{smallest} and r_{largest} are the smallest and largest radii of the sample during one revolution. Figure 12 shows the accepted designations of the sample radii.

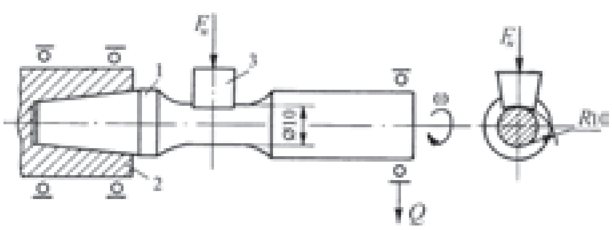


Figure 13 — Scheme of the test for mechano-sliding fatigue: 1 — shaft sample; 2 — spindle; 3 — liner counter-sample

Table 1 — Characteristics of correlations between the coefficient of non-uniformity $\eta_a = y$ and the relative number of cycles $n/N = x$ under conditions of sliding friction and mechano-sliding fatigue of the system “shaft (steel 45) — liner (silumin)”

Contact load F_N, N	Sliding friction		Mechano-sliding fatigue	
	equation	correlation coefficient	equation	correlation coefficient
250	$y = 0.123\ln(x) + 0.832$	0.734	$y = 0.141\ln(x) + 0.781$	0.967
280	$y = 0.121\ln(x) + 0.962$	0.683	—	—
300	—	—	$y = 0.15\ln(x) + 0.8$	0.974
340	—	—	$y = 0.046\ln(x) + 0.878$	0.813
350	$y = 0.14\ln(x) + 0.973$	0.799	—	—
370	—	—	$y = 0.09\ln(x) + 0.545$	0.924
410	—	—	$y = 0.1235x + 0.699$	0.678

Table 2 — Characteristics of correlations between the coefficient of asymmetry $R_a = y$ and the relative number of cycles $n/N = x$ under conditions of sliding friction and mechano-sliding fatigue of the system “shaft (steel 45) — liner (silumin)”

Contact load F_N, N	Sliding friction		Mechano-sliding fatigue	
	equation	correlation coefficient	equation	correlation coefficient
250	$y = 0.0915\ln(x) + 0.915$	0.588	$y = 0.084\ln(x) + 0.841$	0.995
280	$y = 0.082\ln(x) + 0.997$	0.623	—	—
300	—	—	$y = 0.372x + 0.550$	0.930
350	$y = 0.075\ln(x) + 0.993$	0.917	—	—
370	—	—	$y = 0.182x + 0.525$	0.732
410	—	—	$y = 0.055x + 0.887$	0.735

When testing the tribo-fatigue system “steel 45 — 25ХГТ (25KhGT) steel” for mechano-rolling fatigue by the method of stepwise variation of the bending load (see Figure 9) at a contact pressure of $p_0 = 0.7$, p_f = const, it was found that the degree of non-uniformity of local wear-fatigue damage increases accordingly to the increase in cyclic stresses (the greater non-uniformity of wear-fatigue damage is, the smaller the values of R_a and η_a are).

Tests of the mechanical system “shaft (steel 45) — liner (silumin)” for sliding friction (bending load $Q = 0$) and mechano-sliding fatigue ($Q > 0$) under regular loading according to the scheme shown in Figure 13 (test conditions and main results are described in detail in [20]), showed that with an increase in the number of cycles, non-uniformity of local wear-fatigue damage decreases (the values of the coefficients R_a and η_a increase), and most significantly at the initial stage of testing (the run-in stage), which is approximately 0.1 of the number N of loading cycles corresponding to the achievement of the maximum wear of the liner, assumed to be 100 μm.

Tables 1, 2 show the correlation equations connecting the coefficients of asymmetry and non-uniformity with the relative number of cycles n/N (n is the current number of cycles), as well as the corresponding values of the correlation coefficients obtained by processing the results of these tests at different values of the contact load F_N under conditions of sliding friction and mechano-sliding fatigue (at the same level σ_a = 160 MPa of the amplitude of bending stresses of the steel sample). As can be seen from Tables 1, 2, the correlations between the coefficients of non-uniformity and asymmetry, on the one hand, and the relative operating time (essentially, life), on the other hand, of the studied mechanical system were quite strong and are described mainly by a logarithmic, and less often by a linear equation.

Conclusion. Thus, numerous experimental data show that the life of tribo-fatigue systems and friction pairs significantly depends on the variation within the permissible values of the initial states of their elements, determined by the micro- and macrogeometry of the treated surfaces. The establishment of correlation dependencies connecting these parameters makes it possible to assign technically and economically justified requirements for the non-uniformity of the initial states, providing the required life of friction pairs and tribo-fatigue systems. If necessary, its improvement can be carried out by controlling the parameters and modes of the technological process of manufacturing and assembly, taking into account the rational choice of materials, technological equipment and controls.

References

1. Grebennikov A.S., Grebennikov S.A., Konovalov A.V. Zavisimost resursa agregata avtomobilya ot neravnomernosti iskhodnykh zazorov v odnoimennykh sopryazheniyakh [Motor-car hardware life dependence on the initial backlashes non-uniformity in its similar interfaces]. *Vestnik mashinostroeniya*, 2007, no. 6, pp. 34–38 (in Russ.).

2. Perel L.Ya. *Podshipniki kacheniya* [Rolling bearings]. Moscow, Mashinostroenie Publ., 1999. 336 p. (in Russ.).
3. Serov A.V. *Upravlenie effektivnostyu i kachestvom raboty mashin v usloviyakh ekspluatatsii* [Management of efficiency and quality of work of machines under operating conditions]. Moscow, Standartov Publ., 1979. 148 p. (in Russ.).
4. Yashcheritsyn P.I., Skorynin Yu.V. *Rabotosposobnost uzlov treniya mashin* [Efficiency of friction units of machines]. Minsk, Nauka i tekhnika Publ., 1984. 288 p. (in Russ.).
5. Nazarov A.D. *Disbalansy avtotraktornykh dvigateley: opredelenie i normirovanie* [Imbalances of automotive engines: definition and regulation]. Moscow, Mashinostroenie Publ., 1992. 272 p. (in Russ.).
6. Nepomiluev V.V., Mayorova E.A. Issledovanie vozmozhnostey povysheniya kachestva sborki putem individualnogo podbora detaley [Investigation of ways to improve assembly quality by customizing parts]. *Sborka v mashinostroenii, priborostroenii*, 2006, no. 10, pp. 43–46 (in Russ.).
7. Saninskiy V.A. Gruppovaya identifitsirovannaya sborka vkladyshey korennykh podshipnikov skolzheniya s korennymi oporami i kolenchatym valom DVS [Group identified assembly of main plain bearing shells with main bearings and an ICE crankshaft]. *Vestnik mashinostroeniya*, 2006, no. 4, pp. 31–36 (in Russ.).
8. Skorynin Yu.V., Levantsevich M.A., Shapar V.A. *Regulirovanie skorostnykh podshipnikov kacheniya* [Adjusting high-speed rolling bearings]. Minsk, Nauka i tekhnika Publ., 1989. 166 p. (in Russ.).
9. Gerasimova N.N., Sukhanova V.V. Issledovanie vliyaniya volnistosti rabochikh poverkhnostey detaley radialnykh sharikopodshipnikov na uroven vibratsii [Investigation of the influence of the waviness of the working surfaces of radial ball bearing parts on the vibration level]. *Trudy VNIPP*, 1965, no. 2, pp. 74–83 (in Russ.).
10. Tsitovich I.S., Skorynin Yu.V., Kanonik I.V., Minchenya N.T. *Kinematika i dolgovechnost podshipnikov kacheniya mashin i priborov* [Kinematics and durability of rolling bearings of machines and devices]. Minsk, Nauka i tekhnika Publ., 1977. 175 p. (in Russ.).
11. Avdonkin F.N. *Teoreticheskie osnovy tekhnicheskoy ekspluatatsii avtomobiley* [Theoretical foundations of the technical operation of cars]. Moscow, Transport Publ., 1985. 215 p. (in Russ.).
12. Denisov A.S. *Osnovy formirovaniya ekspluatatsionno-remontnogo tsikla avtomobiley* [Basics of the formation of the operational and repair cycle of cars]. Saratov, Saratovskiy gosudarstvennyy tekhnicheskiy universitet Publ., 1999. 352 p. (in Russ.).
13. Baskov V.N., Surkov A.A., Suprun A.S. Analiz tekhnicheskogo sostoyaniya agregatov transmissii [Analysis of the technical condition of transmission units]. *Povyshenie tekhnicheskoy gotovnosti avtomobilnogo transporta*, 1985, pp. 18–21 (in Publ.).
14. Grigorev M.A., Ponomarev N.N. *Iznos i dolgovechnost avtotraktornykh dvigateley* [Wear and durability of automotive engines]. Moscow, Mashinostroenie Publ., 1976. 248 p. (in Russ.).
15. Perov A.A., Sotskov D.A. Tekhnologicheskie faktory, vliyayushchie na povyshennyy iznos gilz tsilindrov dvigataley KamAZ [Processing factors, influencing on the enhanced wear of the KamAZ motor cylinder sleeves]. *Vestnik mashinostroeniya*, 2011, no. 1, pp. 22–24 (in Russ.).
16. Sosnovskiy L.A, Sherbakov S.S. Formirovanie ostatochnykh volnoobraznykh poverkhnostnykh povrezhdeniy pri kontaktno-mekhanicheskoy ustalosti [Formation of residual undulating surface damage during mechano-rolling fatigue]. *Vestnik BelGUTa. Nauka i transport*, 2005, no. 2, pp. 71–87 (in Russ.).
17. Tyurin S.A., Sherbakov S.S. Eksperimentalnoe issledovanie ostatochnykh volnoobraznykh povrezhdeniy pri initsiirovannom nachalnom iskazhenii formy obraztsa [Experimental study of residual wave-like damage with initiated initial distortion of the sample shape]. *Vestnik BelGUTa. Nauka i transport*, 2005, no. 2, pp. 88–93 (in Russ.).
18. Akhmetkhanov R.S. Vliyanie povrezhdennosti elementa konstruktsii na dinamicheskie kharakteristiki sistemy [Effect of damage to structural elements on the dynamic characteristics of the system]. *Problemy mashinostroeniya i nadezhnosti mashin*, 2015, no. 6, pp. 63–69 (in Russ.).
19. Sosnovskiy L.A., Troshchenko V.T., Makhutov N.A., Gao Wang Zheng, Bogdanovich A.V., Sherbakov S.S. *Iznosoustalostnye povrezhdeniya i ikh prognozirovanie (tribofatika)* [Wear-fatigue damage and its prediction (tribo-fatigue)]. Gomel, Kiev, Moscow, Wuhan, 2001. 170 p. (in Russ.).
20. Bogdanovich A.V., Yelovoy O.M., Lis I.N., Golovchenko Yu.A. Eksperimentalnoe issledovanie zakonomernostey obratnogo effekta silovoy sistemy stal 45 – silumin pri friktsionno-mekhanicheskoy ustalosti [Experimental study of the regularities of the inverse effect of the power system steel 45 – silumin during sliding and mechano-sliding fatigue]. *Tribofatika. Trudy 6 Mezhdunarodnogo simpoziuma po tribofatike MSTF 2010, Chast 1* [Tribo-fatigue. Proc. 6th International symposium on tribo-fatigue ISTF 2010. Part 1]. Minsk, 2010, pp. 527–532 (in Russ.).

А.В. БОГДАНОВИЧ, д-р техн. наук, доц.
профессор кафедры теоретической и прикладной механики
E-mail: bogal@tut.by
Белорусский государственный университет, г. Минск, Республика Беларусь

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ОБ ОЦЕНКЕ ВЛИЯНИЯ НЕРАВНОМЕРНОСТИ ИСХОДНЫХ СОСТОЯНИЙ НА РЕСУРС ПАР ТРЕНИЯ И ТРИБОФАТИЧЕСКИХ СИСТЕМ

В статье анализируется взаимосвязь неравномерности исходных состояний и ресурса пар трения и трибофатических систем по результатам их обследования после лабораторных испытаний и эксплуатационной наработки. Неравномерность исходных состояний определяется по варьированию в пределах допусков геометрии элементов, обусловленному технологией изготовления, и оценивается ее влияние на интенсивность изнашивания, динамические характеристики и ресурс в эксплуатации механических систем на примере автомобильных подшипниковых пар трения скольжения и подшипников качения. Анализируются результаты лабораторного эксперимента по оценке работоспособности модели трибофатической системы «вал (сталь 45) — ролик (сталь 25ХГТ)» в связи с локальным нарушением геометрии вала при качении. Приводятся корреляционные уравнения, связывающие интегральные характеристики локального процесса износоусталостного повреждения, основанные на измерениях кругового контура испытуемого образца (коэффициенты асимметрии и неравномерности), с относительным числом циклов до достижения *предельного износа трибофатической системы «вал (сталь 45) — вкладыш (силумин)» при трении скольжения.*

Ключевые слова: *узел трения, трибофатическая система, подшипники, неравномерный износ, испытания, износоусталостное повреждение*

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Список литературы

1. Гребенников, А.С. Зависимость ресурса агрегата автомобиля от неравномерности исходных зазоров в одноименных сопряжениях / А.С. Гребенников, С.А. Гребенников, А.В. Коновалов // Вестн. машиностроения. — 2007. — № 6. — С. 34–38.
2. Перель, Л.Я. Подшипники качения / Л.Я. Перель. — М.: Машиностроение, 1983. — 543 с.
3. Серов, А.В. Управление эффективностью и качеством работы машин в условиях эксплуатации / А.В. Серов. — М.: Изд-во стандартов, 1979. — 147 с.
4. Ящерицын, П.И. Работоспособность узлов трения машин / П.И. Ящерицын, Ю.В. Скорынин. — Минск: Наука и техника, 1984. — 288 с.
5. Назаров, А.Д. Дисбалансы автотракторных двигателей: определение и нормирование / А.Д. Назаров. — М.: Машиностроение, 1992. — 268 с.
6. Непомилуев, В.В. Исследование возможностей повышения качества сборки путем индивидуального подбора деталей / В.В. Непомилуев, Е.А. Майорова // Сборка в машиностроении, приборостроении. — 2006. — № 10. — С. 43–46.
7. Санинский, В.А. Групповая идентифицированная сборка вкладышей коренных подшипников скольжения с коренными опорами и коленчатым валом ДВС / В.А. Санинский // Вестник машиностроения. — 2006. — № 4. — С. 31–36.
8. Скорынин, Ю.В. Регулирование скоростных подшипников качения / Ю.В. Скорынин, М.А. Леванцевич, В.А. Шапарь. — Минск: Наука и техника, 1989. — 166 с.
9. Герасимова, Н.Н. Исследования влияния волнистости рабочих поверхностей деталей радиальных шарикоподшипников на уровень вибрации // Н.Н. Герасимова, В.В. Суханова // Труды ВНИПП. — 1965. — № 2. — С. 74–83.
10. Кинематика и долговечность подшипников качения машин и приборов / И.С. Цитович [и др.]. — Минск: Наука и техника, 1977. — 175 с.
11. Авдонькин, Ф.Н. Теоретические основы технической эксплуатации автомобилей / Ф.Н. Авдонькин. — М.: Транспорт, 1985. — 215 с.
12. Денисов, А.С. Основы формирования эксплуатационно-ремонтного цикла автомобилей / А.С. Денисов. — Саратов: Сарат. гос. техн. ун-т, 1999. — 350 с.
13. Басков, В.Н. Анализ технического состояния агрегатов трансмиссии / В.Н. Басков, А.А. Сурков, А.С. Супрун // Повышение технической готовности автомобильного транспорта: межвуз. науч. сб. / Саратовский политех. ин-т; редкол.: Ф.Н. Авдонькин [и др.]. — Саратов, 1985. — С. 18–21.
14. Григорьев, М.А. Износ и долговечность автотракторных двигателей / М.А. Григорьев, Н.Н. Пономарев. — М.: Машиностроение, 1976. — 248 с.
15. Перов, А.А. Технологические факторы, влияющие на повышенный износ гильз цилиндров двигателей КамАЗ / А.А Перов, Д.А. Соцков // Вестн. машиностроения. — 2011. — № 1. — С. 22–24.
16. Сосновский, Л.А. Формирование остаточных волнообразных поверхностных повреждений при контактно-механической усталости / Л.А. Сосновский, С.С. Щербаков // Вестн. БелГУТа: Наука и транспорт. — 2005. — № 2. — С. 71–87.
17. Тюрин, С.А. Экспериментальное исследование остаточных волнообразных повреждений при инициированном начальном искажении формы образца / С.А. Тюрин, С.С. Щербаков // Вестн. БелГУТа: Наука и транспорт. — 2005. — № 2. — С. 88–93.
18. Ахметханов, Р.С. Влияние поврежденности элемента конструкции на динамические характеристики системы / Р.С Ахметханов // Проблемы машиностроения и надежности машин. — 2015. — № 6. — С. 63–69.
19. Износоусталостные повреждения и их прогнозирование (трибофатика) / науч. ред. Л.А. Сосновский; Л.А. Сосновский [и др.]. — Гомель; Киев; Москва; Ухань, 2001. — 170 с.
20. Экспериментальное исследование закономерностей обратного эффекта силовой системы «сталь 45 – силумин» при фрикционно-механической усталости / А.В. Богданович [и др.] // Трибофатика: тр. VI Междунар. симп. по трибофатике МСТФ 2010, Минск, 25 окт.–1 нояб. 2010 г.: в 2 ч. / Белорус. гос. ун-т; редкол.: М.А. Журавков [и др.]. — Минск, 2010. — Ч. 1. — С. 527–532.