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Research Article

The Contact Mechanics of Novikov's Surface-Hardened Gearing during Running-in Process

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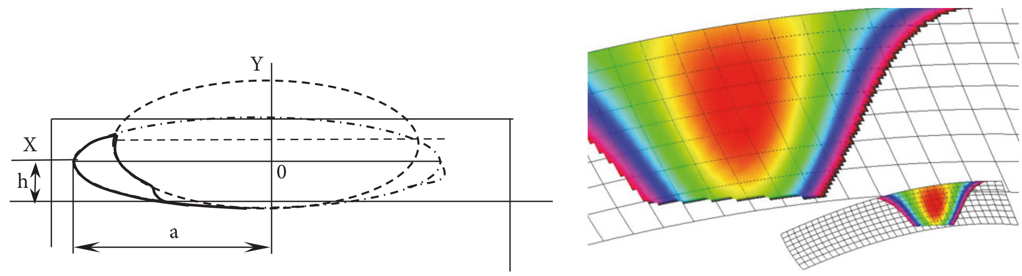
The article is devoted to the analysis of the state of the contact surfaces of the higher kinematic pair in the general case of relative motion, that is, in the presence of rolling, sliding, and twisting, which is characteristic of Novikov's circular-screw gears. The purpose of the work is to assess the impact of friction forces, the state of contact surfaces after tool processing, and the localization of the instantaneous contact spot on the level of contact—fatigue durability of gears. Power contact in the presence of geometric slippage of the mating surfaces leads to a significant change in the initial geometry and the mechanical properties of surface layers. In the existing methods of calculations of contact strength, the effect of running-in is investigated insufficiently, which leads to an incorrect result, especially for gear with high hardness of the teeth. In this work, the conditions of contact interaction close to the real requirements are studied on the basis of experimental material, numerical solution of the contact problem, determination of the terms of the contact areas of slip, and adhesion within the instantaneous spot. The shape of the instant contact spot has asymmetry and can be approximated by an ellipse with the introduction of a correction factor. The running-in period is of a plastic nature with cold deformation and reduction of the roughness of surfaces. As a result of the run-in period, the area of actual contact (tooth height) is increased by 2 or more times. It is not desirable to spread the area of contact at the area of adhesion that initiates the formation of pitting. The presence of defective surface area on the level of contact strength does not have significant influence, because of the running-in period, but increases the risk of spalling and brittle fracture.

1. Introduction

Running-in is the process of transition of the properties of the contacting surfaces of the parts from the initial to the operational. There is a change in size and macro - and microgeometry, as well as physical and mechanical properties of the material of the interacting surfaces to optimize their parameters in a relatively short period. For Novikov's transmissions, this process is of particular importance, because due to its initial point contact passes into contact on the surface, which is due to a sharp increase in the contact strength of such transmissions. However, this is a qualitative indicator. The burnishing process is influenced by many factors—geometric, kinematic, power, lubricant properties, and contact surfaces. It is clear that instantaneous contact spots, both calculated and determined experimentally, differ significantly (Figure 1).

The quasielliptic instantaneous contact spot (IPC) obtained from the initial geometry by the numerical solution of the contact problem [1] and experimentally and numerically on models [2, 3] in the course of processing turns into a kind of curvilinear trapezoid, significantly increasing in size. Thus, for transmission, given in a peer-reviewed article, when p_0= 2013 MPa chordal arc length of the active portion of the profile of the head of the tooth L (Figure 2) due to earnings increased from (1,58...1,6) mm to more than 2,15 mm.

However, if the results of the study are evident, its mechanism is a subject of discussion. Thus, confirmed by many subsequent studies, for example [5], it is considered, in particular,



(a) (b)

Figure 1: The spots of contact in Novikov's gear: (a) unprocessed single gear line (SGL), model on Plexiglas [2], and (b) computer modeling [3, 4].

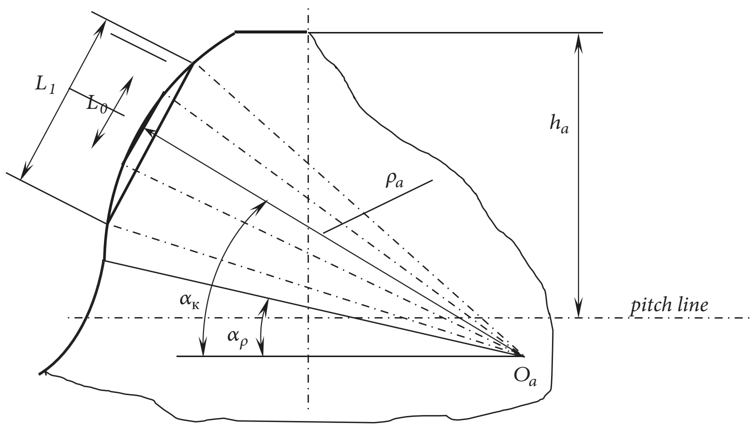


Figure 2: Changing the chordal length of the arc L of the active part of the Novikov gear in the process of running-in. L_0: after running-in at the moment on the output shaft T_2=1000 Nm; L_1: as a result of running-in at the time of failure—a breakdown on the tooth of the tooth fillet at T_2 = 3550 Nm with the operating time of $7.2 \cdot 10^6$ loading cycles; ρ_α: radius of curvature of the tooth; α_κ: pressure angle at the theoretical contact point; α_ρ: the minimum angle of the profile of the active part of the tooth.

a consequence of positive hydrodynamics—increase in the thickness of the lubricant film due to the increased speed of movement of the contact area of the teeth in stable rolling conditions. The opposite point of view is presented in [6], where good workability is due to "bad" hydrodynamics due to a sharp increase in the reduced radius of curvature of the contacting surfaces. In confirmation, it is declared of "low transmission Novikov resistance to scoring." The latter is doubtful because there are no cases of bullies for more than half a century of experience in using Novikov's gears (with different initial contours) and the references given in [6] also do not say anything about the low luminosity of such transmissions, especially about their "congenital defect." When operating in the boundary friction mode on the contacting surfaces, there would be traces of adhesive action or microcutting, but the worked surfaces are carbonized and mirrored. The opinion of Yakovlev [7] on the plastic character of run-in under conditions of a stable elastic hydrodynamic contact, when the total wear is several times sufficient for a qualitative run-in, is sufficient (in the cited paper up to 5), and there are fewer scallops of initial roughness.

Running-in process experiments were conducted on a pin-on-disk tester in [8]. The attractors of friction vibration were investigated [7] by the chaos theory, and the evolvement mechanism of the friction vibration chaotic attractors in the running-in process was analyzed. The experimental results indicate that the friction vibration has a stochastic nature, and the chaotic attractor in the phase space is an always open trajectory with a specific hierarchy and structure.

Running-in attractor was investigated as a stable and time-space ordered structure formed in running-in process in [9]. To establish prediction models of the running-in attractor, orthogonal experiments were performed by sliding pins against a disc during the running-in process. The models proposed in [9, 10] make it possible to predict the operating conditions, properties of tribological contact, and surface roughness, which gives a reference to the operating mode.

The dual-disk model testing concept, shown in [11], provides valuable information during running-in process on the applicability of new technologies, such as surface structuring, coatings, alternative fluids, or modern materials, in real machine elements.

A model for the simulation of wear particles formation and running-in in mixed lubricated sliding contacts was developed in [12]. The simulations based on a previously developed half-space algorithm coupled with a numerical elastohydrodynamic lubrication solver utilizing the load-sharing concept.

The effect of running-in on surface characteristics of spur gears and on their development during subsequent efficiency testing is studied in [13]. Micropitting was associated with surface asperities and their plastic deformation; higher running-in load gave more micropitting, also after identical efficiency tests. Running-in increased unequal compressive residual stresses in both profile and axial directions, while after efficiency testing they approached equal levels.

The conditions of running-in and wear are not exactly identical but interrelated [14]. Models for friction during running-in should include the effects of wear since wear affects the surface topography as well as the formation of transfer films, mechanically mixed layers, and third-body agglomerates.

2. Materials and Methods

Experimental studies [15, 16] and experience in the operation of surface-hardened Novikov gear (mainly type GOST 30224-96) showed that, in the group of medium quality (8...10 degree of accuracy), they are superior to their equivalent involute not less than 35...40%. At design pressures up to 2000 MPa, there were no cases of surface contact failure. The work surfaces were smooth, shiny, no pitting. However, many failures-end chips, fractures on the background of the development of deep contact cracks, to some extent, are the result of contact interaction and require more reliable information about the nature of the stress state in real contact.

The general case of relative motion—the combination of rolling, slippage, and rotation—is characteristic of Novikov gearing. The instant contact area is different from the ellipse. The effect of the cantilever application of the load to the tooth is obvious, but little has been studied. Therefore, the characteristics of the stress-strain state (SSS) in the contact area are determined numerically. That is justified in the research plan, but in the solution of applied engineering problems, it is advisable to enlist analytical models, which are corresponding to the practical problem only approximately, significantly simplify the solution, and enhance the universality of the results obtained. Hence, the assessment of the possibilities and conditions of using existing analytical solutions is a necessary stage in the development of engineering methods of calculation. The study of contact interaction provides the determination of the mode of lubricating action in the conditions of actual contact, determination of the characteristics of the contact area, and determination of the components of the stress-strain state (SSS) in the contact area.

Hardening thermal or chemical heat treatment (CHT) makes significant adjustments that affect the changes in the conditions of contact and the properties of the material of the contacting elements. The existing regulations of traditional involute transmissions are not applicable since the stress state in the contact area is different even for Novikov transmissions with different initial contours. This paper analyzes the results of the assessment of SSS in the contact area of gear Novikov-based solutions to problems of solid mechanics—analytical and numerical—and the results of bench testing carbonitriding transmission Novikov with the initial loop of SWG-5 (base for GOST30224-96) "Reducer" (Izhevsk).

3. Results and Discussion

3.1. Contact Mechanics Study. Tests were carried out in single-stage gearboxes CU-160 on stands with a closed flow of power, mechanical loading, the number of revolutions of the drive shaft n_1=1500 min^{-1}, and the circulating oil lubrication system MS-20. The transmissions with parameters were tested: center-to-center distance: a_w = 160 mm; module: m = 3.15 mm; number of gear teeth and wheels: z_1 = 32, z_2 = 65; angle of inclination of teeth: β = 17,28390; the bias coefficients are $x_1 = x_2 = 0$; width of the rim: b_1 = 50 mm and b_2 = 60 mm; material: steel with 0.25% C, 1% Cr, 1% Mn, and 0.2% Mo; CHT: nitrocarburizing with $h_{eff} \approx$ 0.8mm (up to He = 550HV1); finishing is not provided; accuracy on the scale of ND-4. The average roughness of the nontreated surfaces is R_a = 0.45 μm (grade of cleanliness - 7b) and the working surfaces are R_a = (0.30... 0.35) μm (cleanliness class - 9a). A total of 13 pairs were tested in the range of nominal torques on the drive shaft: T_1 = 1500 ... 2000 Nm.

The study of contact interaction provides the solution of the following tasks: determination of the mode of lubricating action in the conditions of actual contact; determination of the characteristics of the contact site—**E** and the pressure distribution—p and the components of the stress-strain state in the contact area. The solution of the first problem determines the possibility of solving subsequent ones. In the conditions of boundary lubrication, the contacting takes place according to the so-called "contour area" that depends on the waviness of the surfaces, load, and mechanical properties of the contacting bodies and not exceeding (10-15) % of the nominal, that is, on which the bodies would touch if their surfaces had a perfectly smooth geometric shape. In hydrodynamic (liquid) lubrication, the decisive factor is internal friction in the lubricant volume. The combined effect of the rheological properties of the lubricant and the elastic properties of the bodies determines the elastic hydrodynamic lubrication regime. Characteristic mode of friction is the parameter λ, the relative thickness of the film layer between the contacting surfaces: $\lambda = h/(R_{a1} + R_{a2})$, where h is the absolute film thickness; R_{a1} and R_{a2} are an arithmetic average deviation of the profile of asperities from the baseline. Recommendations on the criterion values of λ differ, but most often the regimes for λ <2, are classified as the boundary, at λ=2...5, as elastic hydrodynamic, and at λ > 5, as hydrodynamics. Under the conditions of the elastic hydrodynamic regime, the pressure plot differs from the Hertz plot for the "dry" contact by the presence of the input zone and the perturbation at the output of the contact. The difference in the maximum pressure in the contact zone differs from that determined by the formulas, following from the theory of Hertz, and does not exceed 20 %. That makes it possible to use the dry contact dependencies in the first approximation to determine the pressure in the lubricant. The thickness of the lubricant film in the elliptic contact was determined by the known formula [17, 18]

$$h = R_x \left[(1.82 - 0.68X) \left(\frac{\mu \alpha v}{R_x} \right)^{0.75} \left(\frac{p_0}{E^*} \right)^{-0.25} \right] \quad (1)$$

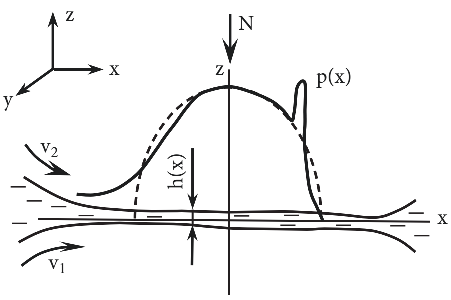


FIGURE 3: Scheme of the pressure distribution in the contact area of the cylindrical surfaces in elastohydrodynamic regime.

where $R_x = R_{x1}R_{x2}/(R_{x1} \pm R_{x2})$ is the reduced radius of curvature; R_{x1} and R_{x2} are the curvature radii of the first and second surfaces in the section passing through the direction of rolling or sliding (Figure 3, the x-axis) and a common normal to the surfaces (Figure 3, the z-axis); $X=R_{\min}/R_{\max}$ is measure of the curvature of the surfaces in the yz cross section; $0< X <1$. R_y is given the radius of curvature of the surfaces in the yz cross section; μ and a are dynamic viscosity and piezoelectric effect of the base oil at a temperature T at the entrance to the contact zone; $E^* = E_1/(1\text{-}\gamma^2)$ is reduced modulus of elasticity; γ is Poisson's ratio; $v = (v_1 + v_2)/2$ is average speed of movement of the surfaces of parts; v_1 and v_2 are speed of movement of the first and second surfaces; p_0 is the maximum pressure in contact with the Hertz.

3.2. The Parameters of the Tested Gear. Center distance is a_w= 160 mm; module is m = 3,15 mm; number of gear teeth and wheels is z_1= 32 and z_2= 65; angle of inclination of the teeth is β = 17.28390; the coefficients of the displacement is x_1= x_2= 0; the width of the bases is b_1= 50 mm b_2= 60 mm; material is steel with C=0.25%, Si=0.17-0.37%, Mn=0.9-1.2%, Cr=0.9-1.2%, Ni<0.3%, Cu<0.3%, and Mo=0.2-0.3%; chemical-thermal treatment nitrocarburizing is c $h_{\text{eff}} \approx 0.8$ mm (to He= 550HV); finish machining is not provided; the accuracy is on a scale ND-4. The average roughness of the untreated surfaces is Ra= 0.45 μm (purity class - 7b), worked surfaces are Ra= (0.30...0.35) μm (purity class - 9a). Test conditions are as follows: torque on the drive shaft: T_1 = 1500-2000 Nm; mode: 100%; drive shaft speed: n_1 = 1500 min^{-1}; lubrication with MS-20 oil.

Tested gear is as follows: R_x= 716.55 mm; R_y= 22.86 mm; p_o= 2013 MPa; E^*= 2.15e+5 MPa; γ = 0,3. Speed v is determined by [19]

$$v = \omega_1 \left\{ \left[r_1 + 0.5L_t \left(u^{-1} - 1 \right) \sin \alpha_t \right]^2 + \left[-0.5L_t \left(u^{-1} - 1 \right) \cos \alpha_t \right]^2 + \left(r_1 ctg\beta \right)^2 \right\}^{0.5} \tag{2}$$

where $r_1 = mz_1/2 \cos ß$ is the dividing diameter of the gear and $u = z_2/z_1$ is the gear ratio; L_t is the pressure angle and the distance from the pole to the theoretical contact point in the end plane: v = 27.9 m/s. At a temperature of T=1000 C (almost not reached in the tests), μ =1.84e-4 Pas and a = 14.5e-7 Pa^{-1}. In this case (even without taking into account a multiplying factor X) the minimum value of the thickness of the layer h = 2.07 μm and (when R_{a1}= R_{a2}= 0.45 μm) λ = 2.3. Consequently, the transmissions worked in a stable elastohydrodynamic regime.

At the same time, similar calculations were carried out in accordance with the works of Hamrok-Dawson [18], but the calculated hydrodynamic parameter never fell below 2.5, which is even more favorable.

3.3. Characterization of Contact Area E and Pressure Distribution p without regard to Running-in. When the elastic characteristics of the materials of interacting bodies are equal, the jump of normal displacements determining the contact area depends only on the normal load [19], which allows separating the problem of determining S and p from the problem of determining the SSS and solving them sequentially.

Problem 1. In the orthogonal coordinate system $OX_1X_2X_3$, where X_3=0 is the common tangent plane, E, an unknown contact region, is lying in this plane. The problem is to find the solution of the homogeneous equilibrium equations of the elastic medium in the half-spaces $X_{3(1)} > 0$, $X_{3(2)} < 0$ satisfying the boundary conditions

$$\begin{aligned} & u^+(x_1, x_2) - u^-(x_1, x_2) \\ & \quad = \delta - r^+(x_1, x_2) - r^-(x_1, x_2) = F(x_1, x_2), \\ & p(x_1, x_2) \geq 0, \\ & (x_1, x_2) \in \mathbf{E} \\ & u^+(x_1, x_2) - u^-(x_1, x_2) > F(x_1, x_2), \\ & p(x_1, x_2) = 0, \\ & (x_1, x_2) \notin \mathbf{E} \end{aligned} \tag{3}$$

Problem 2. The solution obtained in the previous problem—the definition of the contact area and the density of the normal load [**E**, $p(x_1, x_2)$]—allows us to calculate the components of the stress state, in which under the simultaneous action of the normal $p(x_1,x_2)$ and tangent $t_{x1}(x_1, x_2)$ loads can be, based on the solution of Boussinesq-Cherutti, presented as

$$\tau_{i,j} = \varphi_2 \left(x_3, \frac{\partial V}{\partial x_1}, \frac{\partial V}{\partial x_2}, \frac{\partial V}{\partial x_3}, \frac{\partial^2 V}{\partial x_k \partial x_3}, \frac{\partial^2 V}{\partial x_i \partial x_j}, \frac{\partial^2 V}{\partial^2 x_k}, \frac{\partial^3 W}{\partial x_i \partial x_j \partial x_k}, \frac{\partial^3 U}{\partial x_i \partial x_j \partial x_k} \right) \tag{4}$$

$$\sigma_{i,j} = \varphi_1 \left(x_3, \frac{\partial V}{\partial x_3}, \frac{\partial^2 V}{\partial x_1 \partial x_2}, \frac{\partial^2 W}{\partial x_1 \partial x_2} \right), \quad i = 1, 2, \ j = 1, 2, \ k = 1, 2, \tag{5}$$

V, W, and U are, respectively, Newtons, logarithmic Boussinesq, and biharmonic potentials of a simple layer.

Tangential loads are caused by friction forces, which are generally determined by the kinematics of the interacting

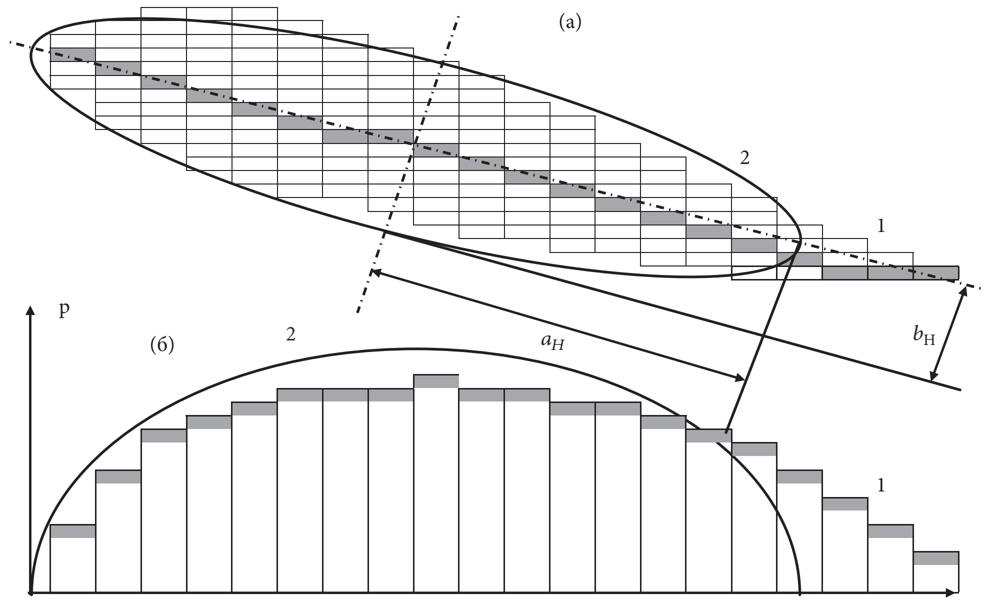


Figure 4: Contact area (a) and pressure distribution (b) in Novikov's cylindrical gear: 1: contact of real surfaces; 2: equivalent elliptical contact.

bodies as absolutely rigid, as well as by elastic deformations that violate the "ideal" kinematics of rolling. Thus, the instantaneous contact spot can represent either the area of total slip $\mathbf{E} = E_+$ or a combination of slip and clutch zones E_0 with zero relative slip speeds, i.e., $\mathbf{E} = E_+ \cup E_0$; this model of friction on Amonton-Coulomb is realized only in E_+.

The solution of the problem of determining the tangential loads in the contact of a higher kinematic pair (in particular, with respect to Novikov transmissions) was carried out on the basis of the variational approach, which formulated it as the problem of minimizing the Kalker [20] energy functional in Spector and Fedorenko [21]. The error in the numerical results (relative to the analytical ones) on the grid was 0.62% and 1.06% for the intensity of the octahedral stresses.

The theoretical and experimental studies carried out showed the following:

(1) The shape of the instantaneous contact spot (Figure 4(a)), obtained by a numerical solution of the contact problem, has a pronounced asymmetry and qualitatively corresponds to the results of the experiments.

(2) For the same compressive forces, the numerically obtained values of $p_{\max}$ of the maximum contact pressures are lower than when using the analytical solution for elliptical contact (Figure 4(b)). The discrepancy did not exceed 5%, and as the load increased, it decreased.

(3) But, despite the smaller values of $p_{\max}$, the equivalent stresses (in this case, the intensity of the octahedral stresses) in the central region of the contact area, determined numerically, are somewhat higher than those obtained analytically. The excess is small (up to 4%) and in the calculation methods was taken into account by the introduction of the corresponding coefficient.

(4) Under the conditions of an out-of-pole contact, the distribution of tangential frictional stresses is signposted (in the case of an elliptical contact close to an ellipsoidal one) with a certain shift of the maximum in the rolling direction (Figure 5(a)). Under these conditions (complete slippage), the frictional forces are increased and proportional to the magnitude of the friction coefficient at the contact surface, but they quickly decay as they move away from the surface.

(5) The presence within the contact area of the adhesion region with zero relative slip (which is possible with the extension of the contact to the pole zone) dramatically changes the distribution of tangential frictional stresses. On the interface between E_0 and E_+, zones of local concentration of these stresses appear (Figure 5(b)). Insignificant in absolute value, they indicate the probability of pitting.

3.4. Metallographic Analysis of Samples. Metallurgical studies checked the quality of the near-surface area of the tested gears. Fractographic analysis, metallographic analysis of the core and hardened layer, measurement of microhardness from the surface to the core, and measurement of the roughness of the contact surface were carried out. The measurements were carried out both on the working side and on the nonworking side of the tooth.

The microstructure is quite stable (Figure 6): at a depth up to 0.14 ... 0.18 mm trooco-or-latent-leaf martensite; further small-and-middle-needle martensite (points 4, partially 5), residual austenite (scores 3 ... 5). At depths of 0.4...0.6 mm are bainite martensitic structures, and deeper is bainite. Ferrite in the core is not detected, and the dark component is absent.

Despite the fact that all samples were made from the same batch steel and HTO passed simultaneously, the spread of hardness values reached 150 HV1 units within one tooth and 250 HV1 for the same points of different samples (Figure 7). Measurements were performed on both the working and the nonworking sides of the teeth at the load application site. No significant change in hardness due to run-in was detected. In

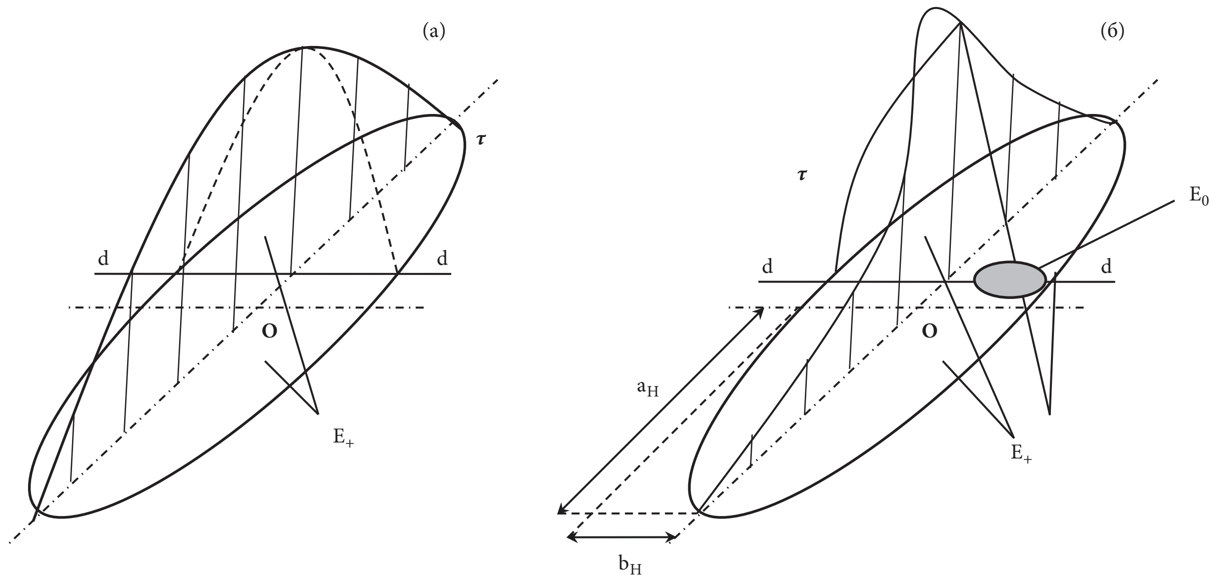


Figure 5: The distribution of tangential stresses over the contact surface in the Novikov transmission: (a) - $X_1 = X_2 = 0$: complete slip; (b) -$X_1 = -0.3$; $X_2 = + 0.3$: slippage and grip. X_1 and X_2: the coefficients of displacement, respectively, gears, and wheels.

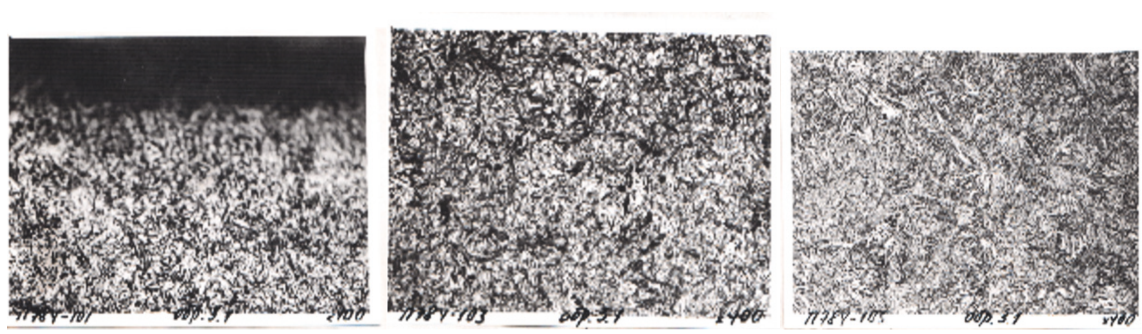


Figure 6: Microstructure of the reinforced layer and core (Sample 3.1). (a) near-surface zone; (b) effective area; (c) the core.

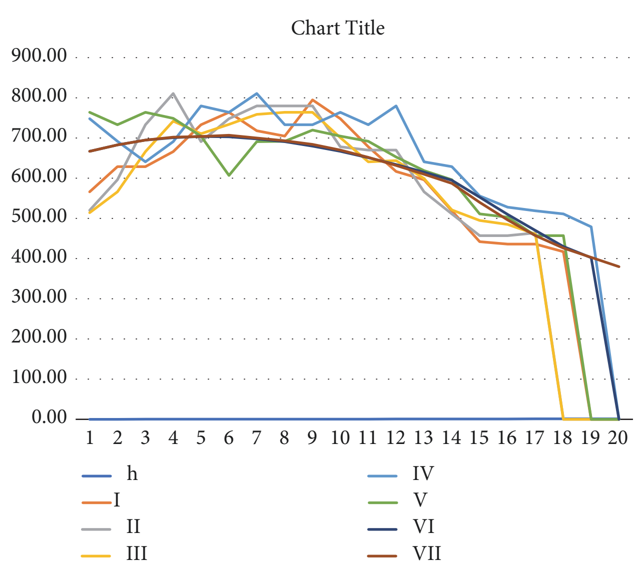


Figure 7: Hardness distribution along the thickness of the hardened layer. 1-3: actual distribution of hardness of samples of one batch (3 samples); 4: probabilistic curve of the actual hardness distribution (for 14 samples with a probability of 90%); 5: distribution according to the approximation.

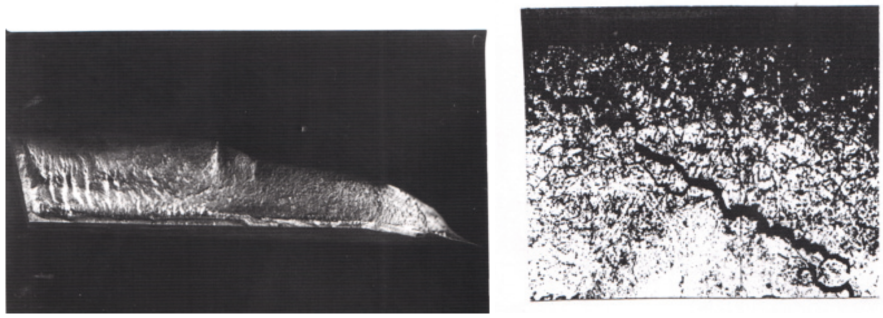


Figure 8: Surface of failure (sample 3) and sample microslip.

65% of cases hardness on the working side was slightly higher and in 20% the opposite picture was observed, i.e., there was "loosening."

The destruction of teeth in the majority of cases had the character of fatigue kinks (Figure 8), with more or less pronounced zones of slow and accelerated development of cracks developing from surface foci. Defects, witnessing about them, as a result of contact interaction are not found. An exception is sample 12, in whose section (Figure 8) a crack is found that develops from the depth at an angle to the surface of about 320. The microstructure indicates a significant overheating: in the effective zone a large needle (up to 16 micrometers) at a depth of 0.30...0.35mm (more than 100HV lower than at a depth of 0.50...0.55mm).

The qualitative side of the process of running-in is obvious, but quantitative side is practically not studied. In [22] it was proposed to introduce the coefficient of increment of teeth in height [22] for its evaluation. It is assumed that the radius of curvature of the radiused curvature Ra will be increased until the instantaneous contact patch spreads along the height of the tooth by the value $Z_l K_l L$. Here L is the chordal length arcs of the active portion of the tooth head and K_l is the coefficient of reduction of the active tooth height due to the cutting of the teeth by running-in. The value of K_1 is determined by the parameters of the initial contour, the coefficient of displacement by the module, and the number of teeth of the transmission wheels formulas (11-10-11-12), table 17 of work [22]. Based on the test results, the dependence Z_l = 0.95-3.7.10-4 (HHB-200), where HHB is the minimum hardness of the active surfaces of the teeth in the gear wheel and the wheel. For hardened and X-hardened wheels it is possible to take Z_l = 0,82.

Considering the running-in, the dependence for determination of the normal contact stress of the gear, with the original contours type GOST 30224-96 (with the center of curvature of the head of the tooth lying outside the pitch line), takes the form

$$\sigma_H = Z_H \left[\frac{T_H}{z \cos \alpha_k} \right]^{0.687} m^{-2.063} (R_x)^{-0.312} \cdot \left(Z_1 K_1 L_p \right)^{-1.063} \tag{6}$$

where T_H is estimated torque on the wheel, Nm.

Z_H = 3,7103 is for transmissions with source contours of GOST 30224-96 type (with the center of curvature of the tooth head lying outside the dividing line), with the center of curvature on the dividing line Z_H = 3.96.103.

Rx is the reduced longitudinal radius of curvature at the theoretical contact point.

L_p is the chordal length of the arc of the active portion of the head of the tooth of the lath.

4. Conclusions

(1) Run-in significantly changes the conditions of contacting. During the running-in process the contact spot spread in height from 15...20% of the active profile to 60...80%. Almost completely neutralized defectiveness (due to CHT) is near-surface zones of the hardened layer. The cleanliness of surfaces increased by 1-2 classes.

Even the unhandled transmissions worked in a stable elastic hydrodynamic lubrication regime. The evaluation of the stress-strain state at the early stages of operation can be performed on the basis of the initial one, without taking into account the rheology of the lubricant, which greatly simplifies the calculations.

(2) Chemical heat treatment (CHT) of the experimental samples was performed at an effective depth of h_{eff} ≈0.25m, which seems overestimated. The contact strength does not exert any influence, and the hardest layers of a lesser thickness will be optimal for breaking strength. Another negative point is the increase in the time of the CHT process, which leads to an increase in the austenite grain and the depth of the defective near-surface zone. The result of overheating (samples 8,12) is an increase in the surface troostomartensitic zone and the adjacent zone of medium- and coarse-grained martensite: the character of the destruction is brittle with a developed region of the dolomite. In addition, cases of a decrease in the hardness of the nitrocement layer on the working surface are noted, as compared to the nonworking layer, i.e., loosening. It is possible to recommend CHT Novikov transmissions to the effective depth h_{eff} ≈(0.18 ... 0.2) m with mandatory control of the microstructure of the surface zone.

(3) The appearance of the area of adhesion within the contact area is undesirable. A Novikov's high-speed transmission was tested with a rectilinear (in the initial contour) near-pole section, connecting the arc sections of the head and the

tooth fillet (the root), providing the possibility of its use in the process of engagement. Despite the fact that the profile angle of this site was 340, it was struck by pitting, while the extra-pole (Novikov) areas were mirrored. When designing Novikov's adjusted gears, the bias coefficients should be assigned from the condition that the contact propagation to the pole zone be excluded [22]. It can be measured with nondestructive tests [23, 24].

(4) When designing transmissions, consideration should be given to the effect of alternative types of failure. With surface hardening, the anticipatory effect of deep contact fractures (DCF) with a fundamentally different (from the surface) model of contact fatigue life is possible. Calculations for the prevention of DCF must be carried out on the original geometry, without taking into account the burnishing, because they are the leading in the initial stages of operation (up to about 107 loading cycles).

Data Availability

The paper data used to support the findings of this study are included within the article. There are no restrictions.

Conflicts of Interest

The authors declare that they have no conflicts of interest.

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References

[1] V. I. Korotkin and N. P. Onishkov, "To the problem of contact fatigue life of surface hardened Novikov gears," *Vestnik Rostovskogo Gosudarstvennogo Universiteta Putey Soobshcheniya' (Vestnik RGUPS)*, vol. 3, p. 14, 2007.

[2] A. I. Pavlov, "Contact of convex and concave surfaces in gearing," *Herald of the National Technical University "KhPI"*, vol. 10, pp. 99–102, 2002.

[3] S. V. Lunin, "New discoveries in WN gear geometries," http://www.zakgear.com, 2001.

[4] S. Lunin, *Interactive visualization with parallel computing for gear modeling*, 2012, http://www.zakgear.com/Parallel.html.

[5] Y. Ariga and S. Nagata, "Load Capacity of a New W-N Gear with Basic Rach of Combine Prifile," *Journal of Mechanisms, Transmissions, and Automation in Design*, vol. 107, no. 4, pp. 565–572, 1985.

[6] G. A. Zhuravlev, "The fallacy of the physical basis of engagement Novikov as a reason for the limited application," *Reducers and Gears*, vol. 1, 2006, http://news.reduktorntc.ru/arc/06_1/1_06_38_45.pdf.

[7] T. Liu, G. Li, H. Wei, and D. Sun, "Experimental observation of cross correlation between tangential friction vibration and normal friction vibration in a running-in process," *Tribology International*, vol. 97, pp. 77–88, 2016.

[8] D. Sun, G. Li, H. Wei, and H. Liao, "Experimental study on the chaotic attractor evolvement of the friction vibration in a running-in process," *Tribology International*, vol. 88, pp. 290–297, 2015.

[9] Y. Zhou, X. Zuo, H. Zhu, and T. Wei, "Development of prediction models of running-in attractor," *Tribology International*, vol. 117, pp. 98–106, 2018.

[10] P. Bergmann, F. Grün, F. Summer, I. Gódor, and G. Stadler, "Expansion of the Metrological Visualization Capability by the Implementation of Acoustic Emission Analysis," *Advances in Tribology*, vol. 2017, Article ID 3718924, 17 pages, 2017.

[11] J. Moder, F. Grün, M. Stoschka, and I. Gódor, "A Novel Two-Disc Machine for High Precision Friction Assessment," *Advances in Tribology*, vol. 2017, Article ID 8901907, 16 pages, 2017.

[12] A. Akchurin, R. Bosman, and P. M. Lugt, "Generation of wear particles and running-in in mixed lubricated sliding contacts," *Tribology International*, vol. 110, pp. 201–208, 2017.

[13] D. Mallipeddi, M. Norell, M. Sosa, and L. Nyborg, "Influence of running-in on surface characteristics of efficiency tested ground gears," *Tribology International*, vol. 115, pp. 45–58, 2017.

[14] P. J. Blau, "On the nature of running-in," *Tribology International*, vol. 38, no. 11-12, pp. 1007–1012, 2005.

[15] V. I. Korotkin, N. P. Onishko, and Y. D. Kharitonov, "Stress state of Novikov gearing teeth under the conditions of real multipair engagement," *Spravochnik. Inzhenernyi zhurnal*, pp. 11–17, 2015.

[16] V. I. Korotkin and D. A. Gazzaev, "Modeling of the contact interaction of the teeth of the gear wheels of Novikov," *Journal of Machinery Manufacture and Reliability*, vol. 43, pp. 104–111, 2014.

[17] "Friction, wear and lubrication," Handbook. Moscow, Mechanical engineering, vol. 2, 1979. 358 p.<?cmd?>.

[18] B. J. Hamrock and D. Dowson, "Isothermal Elastohydrodynamic Lubrication of Point Contacts, Part III," *Journal of Lubrication Technology, Transactions of ASME*, vol. 99, pp. 264–275, 1977.

[19] A. N. Beskopylny, N. P. Onishkov, and V. I. Korotkin, "Assessment of the fatigue durability of the rolling contact," *Advances in Intelligent Systems and Computing*, vol. 692, pp. 184–191, 2018.

[20] J. J. Kalker, "Rolling with slip and spin in the presence of dry friction," *Wear*, vol. 9, no. 1, pp. 20–38, 1966.

[21] R. V. Goldstein, A. F. Zazovskii, A. A. Spektor, and R. P. Fedorenko, "The solution of spatial contact rolling problems with slip and clutch by variational methods," *Advances in mechanics*, vol. 5, no. 3, pp. 61–100, 1982.

[22] V. I. Korotkin, N. P. Onishkov, and Yu. D. Kharitonov, "Novikov gears. Achievements and development," *Machine Building*, p. 384, 2007.

[23] A. N. Beskopylny, A. A. Veremeenko, E. E. Kadomtseva, and N. I. Beskopylnaia, "Non-destructive test of steel structures by conical indentation," in *Proceedings of the MATEC Web of Conferences*, vol. 129, 2017.

[24] A. N. Beskopylny, A. A. Veremeenko, and B. M. Yazyev, "Metal structures diagnosis by truncated cone indentation," in *Proceedings of the MATEC Web of Conferences*, 2017.