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Modeling of Wear-Out Failures and Service Life Improvement of Sealing Units

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Abstract

The article is concerned with a new method of setting up and solution of boundary value problems of reliability theory for sealing units on the basis of wear resistance criterion of sealing elements. The research group carried out theoretical study of service life for the standard "roller - shoe" friction couple on the basis of wear resistance of shoes manufactured from different sealing materials. Theoretical results were verified and the most durable sealing materials were introduced into commercial operation.
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Keywords: Sealing unit; Failure pattern; Sealing element; Wear resistance; Durability; Service life.

1. Problem Description

One of the most important issues of metallurgical production is to improve the reliability of hydraulic drives providing precise positioning and high conveying speed of operating elements of various mechanical systems. Positive-displacement hydraulic engines and hydraulic valves are widely used as actuating and control devices [1]. Their technical state determines the service life of hydraulic systems and the level of performance indicators of an industrial enterprise. In its turn, the main cause of hydraulic unit failure is deterioration of sealing elements of movable sealing units [2]. However, the instant of their failure is not very well defined, it is assessed approximately by experience or by the time when leaks of hydraulic fluid appear due to faults in air tightness caused by abrasive wear development. Practical experience shows, that the service life of sealings is very short, it does not exceed 10% of the service life of other friction couples of hydraulic cylinders and hydraulic valves. At the same time the constantly growing requirements to improvement of plant productivity and product quality as well as to reduction of

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maintenance time and emergency downtime make it necessary to significantly improve the service life of hydraulic units as far as wear resistance of their sealing elements is concerned.

That is why there are two important scientific problems to be solved on the stage of a hydraulic system design:

- development of adequate mathematical models of wear-out failures for movable sealing units;
- theoretical design analysis of feasible solutions aimed at improvement of sealing part wear resistance to provide the required level of durability of the designed hydraulic drive taking into account its configuration, application and operating conditions.

2. Development of failure pattern for movable sealing units

Analysis of traditional approaches to modeling of the wearing process of tribocouplings shows that these approaches are based on various classes of empirical and semi-empirical relationships between the wear rate (wear intensity) of triboelements and operating conditions [2-4]. Generalization of classical equations of the phenomenological [5, 6] and the conceptual [7-9] approaches, of the thermodynamic [10-14] and the kinetic [15-17] models as well as relationships developed within the frame of modern combined theories and strategies [18-20] makes it possible to come to the following conclusion.

Practical application of traditional models for estimation of service life of sealing friction assemblies on the design stage requires preliminary model or full-scale tests of engineering samples to obtain some physical quantities comprising the model. Such tests can be both expensive and time-consuming. To reduce the costs and to develop a purely analytical model of sealing element failure, the authors made use of the general methodological approach [21] to forecasting of the technical object reliability, they also used the energy-mechanical concept of stationary tribosystems wear-out [22]. This concept was developed on the basis of simultaneous solution of fundamental equations of the molecular-mechanical [7] and the structure-energy [18] theories of friction.

In this case, the physical and mathematical model of movable sealing unit failures is represented as a one-parameter boundary value problem of reliability theory of the stationary coupling class. However, it is also assumed that the sealing friction couples preserving the tightness of sealing operate most of the time in the steady state mode of fatigue wear under normal operating conditions. Practical experience shows that in this case, wear of the steel coupled element (a piston plunger, a shaft, a slide valve, etc.) can be neglected.

According to the general theory of reliability forecasting [22], the average value of the inside diameter of a seal ring (a set of rings, sleeve gaskets) $\bar{x}_t$ taking into account its contraction during assembling can be taken as the state parameter of the hydraulic unit. In the process of sealing wear with the average rate of $\bar{\dot{y}}$, its diameter increases from its initial value $\bar{x}_t = \bar{x}_0$ to the limiting one $\bar{x}_t = x_{np}$. It can be assumed that the limiting value x_{np} is equal to the minimum design diameter of the coupled element (a piston plunger, a shaft, a slide valve, etc.) to avoid development of looseness and abrasive wear of elements in future operation.

In this case the closed equation system describing physical relationships of the process of wear-out failure formation for sealing friction couples will take the following form:

- equation of movable sealing unit behavior:

$$\bar{x}_t = \bar{x}_0 + \bar{\dot{y}} \cdot t; \qquad (1)$$

- equation of transition of a friction couple to the limiting state:

$$\bar{x}_t = \bar{x}_0 + \bar{\dot{y}} \cdot t = x_{np}; \qquad (2)$$

- equations for the estimation of the average life of a hydraulic unit and its service life:

$$\bar{t} = \frac{x_{np} - \bar{x}_0}{\bar{\dot{y}}}; \quad \bar{t}_{сл} = (1 + \Pi) \cdot \bar{t}; \qquad (3)$$

- kinetic equation of the damage rate of a sealing unit (sealing wear-out):

$$\bar{\dot{y}} = \frac{\alpha^* \cdot \bar{\nu} \cdot f_{мех} \cdot p_{\max} \cdot V_{ск}}{\Delta u_{e*}}. \tag{4}$$

In order to obtain a single-valued solution to the formulated boundary value problem (1) - (4), it is necessary to lay down single valued conditions, which will single out the object under study from similar objects and which will include its characteristic features, initial and limiting conditions.

Distinguishing features of the movable sealing unit under study describe the design features and the analytical model of element loading; limiting value $x_{пр}$ of the $\bar{x}_t$ parameter; value of scheduled downtime $П$ of the hydraulic drive and others.

Initial conditions include a complex of values of six groups of parameters characterizing the initial state of the sealing unit at the initial instant $t = 0$:

- parameters of external loading (friction) including tightness/tension N in the coupling and sliding speed $V_{ск}$ of elements (group 1);
- dimensional characteristics of the coupling including the initial value $\bar{x}_0$ of the $\bar{x}_t$ (group 2);
- microgeometric characteristics of the friction surface (group 3);
- physical and mechanical characteristics of materials of triboelements (group 4);
- thermal and physical characteristics of materials including density u_{e0} of latent energy of defects of the sealing material in the initial state calculated by V.V. Fedorov equation [14] (group 5);
- tribotechnical characteristics of materials of the friction couple: τ_0, β (group 6);

Boundary conditions are represented by a set of equations, which make it possible to describe the interaction of coupling elements on the interface with environment and to determine the stationary values, which are necessary for solving the boundary value problem (1) - (4):

- A_a , $A_{T1,2}$, $\alpha^*_{1,2}$ are the nominal contact area, rubbing area and overlapping ratios [2, 7, 9];
- $p_a(p_{\max})$ is the nominal (maximum) pressure based on I.V. Kragelskiy relationships for sealing units [2, 7, 9];
- $\bar{\nu}$ is the coefficient of external energy transformation in the material of the surface layer of sealings based on B.V. Protasov equations [13];
- f is the total friction coefficient depending on the kind of the stressed state of the surface layer of the wearing element on the basis of Kragelskiy analytical dependences [7];
- T is the temperature of the sealing surface layers calculated on the basis of A.V. Chichinadze equation [10];
- Δu_{e*} and Δu_T are the critical energy intensity and density of kinetic (thermal) component of the internal energy of the material of the sealing surface layer heated to the temperature of T, calculated on the basis of V.V. Fedorov equations [14];
- $f^{у}_{мех}$ is mechanical component of the friction coefficient calculated on the basis of well known relationships between the molecular and mechanical components of friction coefficient in the steady state mode and the formula for assessment of its minimum value [7].

The combination of the set of equations (1)-(4) and relationships enumerated in the boundary conditions determines the formulation of the boundary-value problem of the reliability theory of the movable sealing unit on the basis of sealing element wear resistance. If the structure, distinguishing characteristics and the analytical model of the element loading for the given hydraulic unit are set, the set of equations given above represents the model of its wear-out failures.

3. Design analysis of sealing materials

The research group offers an algorithm of the boundary value problem solution and the results of the theoretical study of durability of the standard friction couple "roller 1 - shoe 2" given in Fig. 1 by the criterion of the shoe wear resistance. The aim of the computer experiment was to get a comparative assessment of the service life of couplings where the shoes were designed from different sealing materials recommended for application in industrial hydraulic systems by both modern domestic and foreign firms «Simrit», «Economos» and in scientific and technical literature [23].

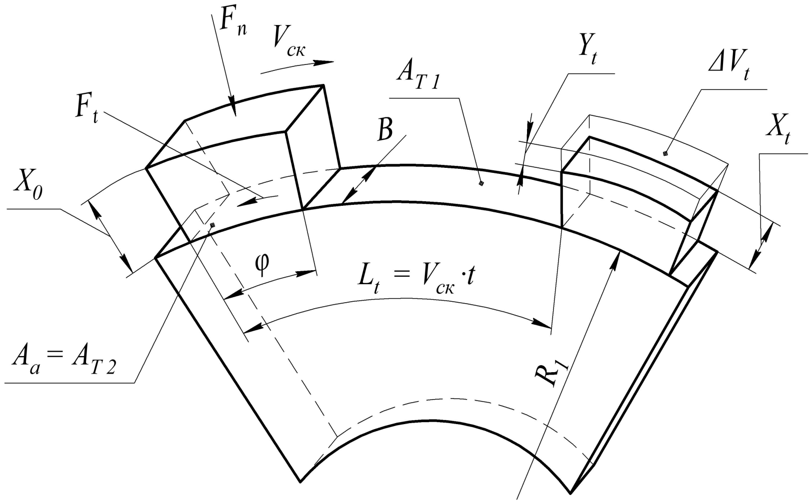

Fig. 1. Analytical model of element loading of a friction couple

The research group studied fifteen kinds of polymers divided into three groups: 1 – elastoplastics on the basis of rubber; 2 – thermal polymers (thermoplastics); 3 – thermal elastoplastics (polyurethane), Table 1, which wear out against rollers made of 40X steel grade.

Table 1 Plan of the computer experiment

Kind of the polymer	№	Roll material - - 40X steel grade
		Shoe material
Elastoplastics on the basis of rubber	1	Artificial butadiene-acrylonitrile rubber «СКН 26»
	2	Isoprene–butadiene-rubber «Compound rubber 3826»
	3	Acrylonitrile – butadiene -rubber «65NBR B210»
	4	Acrylonitrile – butadiene -rubber «80 NBR B246
	5	Acrylonitrile – butadiene -rubber «Ecoruber–H»
	6	Fluorine rubber «Ecoruber 2»
Thermoplastics	7	Polytetrafluoroethylene «Ф4»
	8	Polytetrafluoroethylene «Ecoflon 1»
	9	Filled Polytetrafluoroethylene «Ecoflon 2»
	10	Filled Polytetrafluoroethylene «Ecoflon 3»
	11	Filled Polytetrafluoroethylene «PTFE GM201»
Polyurethane	12	Polyester – urethane – rubber «94 AU V149»
	13	Polyester – urethane – rubber «95 AU V149»
	14	Polyurethane - elastoplastics «H-Ecopur»
	15	Polyurethane - elastoplastics «Ecopur»

The research group offered a block solution algorithm of the boundary value problem for the outlined analytical model, where the shoe thickness is taken as a test parameter x_t of the coupling under study.

Block 1. Initial data (Fig. 1 and Table 1).

Group 1. Parameters of external loading (friction):

F_n $V_{ск}$ are direct force and sliding speed in the place of contact.

Group 2. Dimensional characteristics of elements:

R_1, B, φ are roll radius (contact radius), its width and the angle of contact between the roller and the shoe;

$\bar{x}_0$ is the average value of the shoe thickness in its initial state;

x_{pr} is the limiting value of the parameter $\bar{x}_t$.

Group 3. - Microgeometric characteristics of the friction surface.

R_{a1} and Δ_1 are the arithmetic mean profile deviation and the complex parameter of the surface roughness of the roller;

R_{a2} is the arithmetic mean profile deviation of the shoe surface in its initial state.

Group 4. Physical and mechanical characteristics of materials of the roller and the shoe in the initial state:

$\mu_{1,2}$, $E_{1,2}$, $G_{1,2}$, $\sigma_{пц\,1,2}$, $\sigma_{T\,1,2}$, $\sigma_{B\,1,2}$, $HB_{1,2}$, $HV_{1,2}$, $\alpha_{ef\,1,2}$, $\rho_{1,2}$ are Poisson's ratio, the elasticity and shearing modulus, the proportional elastic limit, yield point and tensile strength, Brinell hardness and Vickers hardness, hysteresis losses coefficient and the density of materials of the roll and the shoe, respectively.

Group 5. Thermal and physical characteristics of materials of the roller and the shoe in the initial state:

$\Delta H_{S1,2}$, $\lambda_{1,2}$, $\alpha_{1,2}$, $c_{1,2}$ are the enthalpy of melting, coefficients of heat conductivity and heat transfer and specific heat capacity, respectively.

Group 6. Friction characteristics of materials of the coupling:

τ_0, β are specific shear strength of the surfaces and coefficient of molecular bond strengthening.

Block 2. Contact characteristics and material properties [2, 7, 9].

- Area of the contact, friction and overlapping ratio:

$$A_a = A_{T2} = R_1 \cdot \varphi \cdot B\,; \qquad A_{T1} = 2\pi \cdot R_1 \cdot B\,; \tag{5}$$

$$\alpha_1^* = \frac{A_a}{A_{T1}}\,; \qquad \alpha_2^* = \frac{A_a}{A_{T2}}\,. \tag{6}$$

- Perimeters of the friction areas:

$$u_1 = 2 \cdot (2\pi \cdot R_1 + B)\,; \qquad u_2 = 2 \cdot (R_1 \cdot \varphi + B)\,. \tag{7}$$

- Nominal pressure on the contact area:

$$p_a = \frac{F_n}{A_a}\,. \tag{8}$$

- Proportional elastic limit and shear modulus of materials:

$$\sigma_{пц1,2} = (0{,}9 \div 0{,}97) \cdot \sigma_{T1,2}\,; \; G_{1,2} = \frac{E_{1,2}}{2 \cdot (1 + \mu_{1,2})}\,. \tag{9}$$

- Elastic stiffness coefficients of the surface layer of the roller and the shoe:

$$\theta_1 = \frac{1-\mu_1^2}{E_1} \quad u \quad \theta_2 = \frac{1-\mu_2^2}{E_2}. \tag{10}$$

Block 3. External energy distribution [13].

- Coefficient of external energy absorption by the material of the roller:

$$\overline{\nu}_1 = 1-(K_\varepsilon \cdot \overline{R}_{a1}^{1/3}+1)^{-1}; \quad K_\varepsilon = \frac{\theta_1^{\frac{2}{3}}}{\theta_2^{\frac{2}{3}} \cdot R_{a2}^{\frac{1}{3}}}. \tag{11}$$

- Coefficient of external energy absorption by the material of the shoe:

$$\overline{\nu}_2 = 1-\overline{\nu}_1. \tag{12}$$

Block 4. Friction coefficient [7, 9, 10].

- Internal stress (contour pressure) in the surface layer of the shoe:

$$\sigma_{a2} = \sigma_{T2} \cdot \left(\frac{p_a}{\sigma_{T2}}\right)^{\frac{p_a}{\sigma_{T2}}}. \tag{13}$$

- The form of the stress state of the surface layer of the shoe:

$$\begin{cases} elastic\ state, if \quad \sigma_{a2} < \sigma_{m2}; \\ elastic\text{-}plastic\ state, if \quad \sigma_{y2} < \sigma_{a2} < \sigma_{T2}; \\ plastic\ state, if \quad \sigma_{T2} < \sigma_{a2} < HB_2. \end{cases} \tag{14}$$

Friction coefficient for a certain kind of the stress state of the surface layer of the shoe:

- - in elastic stressed state:

$$f = 2{,}4 \cdot \tau_0 \cdot \left(\frac{\theta_2^4}{p_a \cdot \Delta_1^2}\right)^{0{,}2} + \beta + 0{,}2 \cdot \alpha_{ef2} \cdot \left(p_a \cdot \Delta_1^2 \cdot \theta_2\right)^{0{,}2}; \tag{15}$$

- - in elastic-plastic stressed state:

$$f = 1{,}25 \cdot \tau_0 \cdot \left(\frac{\theta_2^2}{p_a \cdot \Delta_1}\right)^{\frac{1}{3}} + \beta + 0{,}4 \cdot \alpha_{ef2} \cdot \left(p_a \cdot \Delta_1 \cdot \theta_2\right)^{\frac{1}{3}}; \tag{16}$$

- - in plastic stressed state:

$$f = \frac{\tau_0}{HB_2} + \beta + 0{,}9 \cdot \left(\frac{p_a}{HB_2} \cdot \Delta_1\right)^{\frac{1}{2}}. \tag{17}$$

Block 5. Temperature [10].

- Parameters determined by the properties of the roller and the shoe:

$$m_1 = \sqrt{\frac{\alpha_1 \cdot u_1}{\lambda_1 \cdot A_{T1}}} \quad \text{and} \quad m_2 = \sqrt{\frac{\alpha_2 \cdot u_2}{\lambda_2 \cdot A_{T2}}} . \tag{18}$$

- Temperature of surface layers of the roller and the shoe in the steady-state friction mode:

$$T_{1,2} = \frac{f \cdot F_n \cdot V_{ск}}{A_{T1,2} \cdot (\lambda_2 \cdot m_2 + \lambda_1 \cdot m_1)} + T_0 . \tag{19}$$

Block 6. Stress-strain properties of materials for $T_{1,2} = const$ [24].

- Poisson ratios and elastic modulus of materials of the roller and the shoe as a function of the temperature of the surface layer:

$$\mu_{1,2}(T_{1,2}) = \mu_{1,2} \cdot e^{-0,0005 \cdot T_{1,2}} ; \; E_{1,2}(T_{1,2}) = E_{1,2} \cdot e^{-0,0007 \cdot T_{1,2}} . \tag{20}$$

- Elastic stiffness coefficients of materials of the roller and the shoe as a function of the temperature of the surface layer:

$$\theta_{1,2}(T_{1,2}) = \frac{1 - \mu_{1,2}^2(T_{1,2})}{E_{1,2}(T_{1,2})} . \tag{21}$$

Block 7. Friction coefficient in the steady-state mode [7].

- Total friction coefficient:

$$f^y = (1 \div 1{,}5) \cdot (\tau_0 \cdot \theta_2(T_2) \cdot \alpha_{эф2})^{\frac{1}{2}} + \beta . \tag{22}$$

- Mechanical component of the friction coefficient:

$$f_{mex}^y = \frac{1{,}25 \cdot (\tau_0 \cdot \theta_2(T_2) \cdot \alpha_{ef2}) + \beta \cdot (\tau_0 \cdot \theta_2(T_2) \cdot \alpha_{ef2})^{\frac{1}{2}}}{3 \cdot (\tau_0 \cdot \theta_2(T_2) \cdot \alpha_{ef2})^{\frac{1}{2}} + \beta} . \tag{23}$$

Block 8. Critical energy intensity [14].

- Density of potential component of internal energy of material of the shoe in the initial state – for elastoplastics $u_{e20} = 0$.
- Changes in the density of thermal component of internal energy of material of the shoe at the temperature T_2 of the steady-state mode:

$$\Delta u_{T*2} = \int_0^{T_2} \rho_2(T_2) \cdot c_2(T_2) \cdot dT . \tag{24}$$

- Critical density of the latent energy of material of the surface layer of the shoe (critical energy intensity) in the steady-state friction mode and fatigue wear:

$$\Delta u_{e*2} = \Delta H_{S2} - u_{e0} - \Delta u_{T*2} . \tag{25}$$

Block 9. Damaging [22].

- Numeral characteristics of the linear wear rate of the shoe by (4) for $p_{\max} = p_a$:

$$\bar{\dot{y}} = \bar{\dot{y}}_2 = \frac{\alpha_2^* \cdot \bar{v}_2 \cdot f_{mex}^y \cdot p_a \cdot V_{c\kappa}}{\Delta u_{e*2}}, \tag{26}$$

Block 10. Service life [22].

- Mean life of a friction couple by (3):

$$\bar{t} = \frac{\bar{x}_0 - x_{pr}}{\bar{\dot{y}}_2}. \tag{27}$$

- Coefficient of service life improvement of material of the shoe compared with the least wear resistant rubber СКН-26 (№1, Table 1):

$$K_{t\,i} = \frac{\bar{t}_i}{\bar{t}_1}. \tag{28}$$

This algorithm was used to calculate the mean life $\bar{t}_i$ and the coefficient of the service life improvement $K_{t\,i}$ for the whole range of materials of shoes under study ($i = 1...15$ is the number of the test in Table 1).

The results of the design calculations are given in columns two and three of Table 2.

Table 2 Results of durability prediction for friction couples

Test	Mean life (calculated)	Coefficient of service life improvement	Mean life (experimentally measured)	Average computational error
№	$\bar{t}_i, c$	$K_{t\,i}$	$\bar{t}^{э}_{i\,i}, c$	$\delta_{t\,i}$, %
1	2038	1,00	1894	-7,6
2	2447	1,20	2370	-3,3
3	2242	1,10	2083	-7,6
4	2500	1,23	2459	-1,7
5	3074	1,51	3440	10,7
6	4545	2,23	4000	-13,6
7	2852	1,40	3030	5,9
8	2852	1,40	3145	9,3
9	2935	1,44	3409	13,9
10	3036	1,49	3067	1,0
11	3055	1,50	3409	10,4
12	3686	1,81	3333	-10,6
13	4076	2,00	3750	-8,7
14	3623	1,78	3261	-11,1
15	4732	2,32	4373	-8,2

The test numbers are arranged in the order of increasing of wear resistance of materials used for shoes production (service life of friction couples under study). The most wear resistant materials of shoes, i.e. the most durable friction couples "roller - shoe" with the coefficient of the service life improvement higher than two, are highlighted in Table 2.

4. Verification of theoretical results

To assess the adequacy of the developed model and the obtained theoretical results, the research group carried out a statistically significant experiment of wear-out of shoes made of the materials under study using the friction test machine CMT-1 according to the specified scheme "roller - shoe".

Each of the 15 series of experiments was repeated 4 times and the materials of specimen and friction conditions $F_n = 200\,H$ and $V_{ск} = 1{,}5\,m/c$ were kept constant. During the tests the frictional contact of the details was constantly cooled by flowing water and its temperature was controlled by the infrared radiation thermometer «CONDTROL IR - T4».

The stability of loading conditions of the specimen during the tests was constantly controlled by means of milliamperemeter scales holding the value of loading F_n, and by means of a high-precision variable resistor holding the value of friction torque M_t. Calibration tests of scales of the devices described above were carried out beforehand to provide the accuracy of measurements.

The average time of shoe wear in each series was calculated beforehand using equation (3) taking into account (4) presetting the limiting wear of shoes as 1.5 millimeters. During this period pauses were made at regular time intervals to make records of the mass and linear wear of shoes. Its magnitude was assessed by weighting the shoes on the electronic reference scales of the 1st grade ME215S (accurate to grams of the 4th decimal place) before the tests, during the pauses and upon the completion of wear process. During the tests surface roughness of the specimen was measured on a sampling basis using the device «Perthometer S2 MAHR».

The values of the average experimental service life of the four tests are given in the third column of Table 2.

The method of the engineering estimate of coupling durability was validated by comparing the theoretical $\bar{t}_i$ and the experimental $\bar{t}_i^э$ values of the average life in each i-th set of experiments. The results of comparison were quantitatively assessed by the value of the relative error $\delta_{\bar{t}\,i} = (\bar{t}_i^э - \bar{t}_i)\cdot 100/\bar{t}_i^э$ given in the fourth column of Table 2.

Integration of the data from the Table shows that the spread of values of prediction errors of the average service life of the studied friction couples is within the range of $\delta_{\bar{t}\,i} \approx \pm 13{,}5\%$. The authors believe that such deviations prove high integrity of the theoretical results and the failure model of hydraulic units, on which these results are based.

5. Implementation of the obtained results

On the basis of the offered method, the research group developed models of wear-out failures for positive-displacement hydraulic engine systems of mill roll balance [25] and slide valves of hydraulic descaling of strip [26] for sheet hot rolling mills.

It was suggested to use the most wear-resistant materials found as a result of the tests of the friction couples "roller - shoe" to manufacture sealings of the movable sealing unit of the specified hydraulic units. These materials underwent factory tests and were introduced into production.

The research group got several useful model patents for a number of hydraulic units of new design [27, 28].

6. Conclusion

Thus, even on the stage of design of hydraulic systems, one can give a reasonably accurate assessment of the reliability level and durability of movable sealing units and analyze the ways of improving their expected service life, for example, by means of selection of antifrictional materials or by means of introducing certain changes into their design, surface microdeometry, conditions of friction on the contact area, etc.

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