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Improvement of tribological characteristics of coupling parts "shaft-sleeve" with polymer and polymer-composite materials

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Abstract

The article provides an analytical justification of the flow of tribological processes of coupling of "shaft-sleeve" parts, which simulates the functioning of sliding bearings and cylindrical joints of machines. The main attention is paid to such characteristics as contact pressure, static and dynamic forces, the criterion of the product of the total pressure on the sliding speed, the work of friction forces and its transition into thermal energy in the friction zone for polymer (based on polyamide P-68) and polymer-composite coatings (based on P-68 with kaolin filler) on the working surfaces of the parts. A comparative analysis of the functioning and tribological characteristics of the couplings of parts without coatings is presented.

Experimentally, on the basis of tests of samples on the МИ-1М friction machine, a significant reduction in wear and an increase in the relative wear resistance of samples with polymer-composite coatings in the modes of friction without lubrication (by 1.3...1.4 times) and marginal friction (in 1.2...1.3 times), as well as a decrease in the temperature in the friction zone (365 K and 347 K) compared to the polymer coating.

Keywords: tribological characteristics, polymer material, polymer-composite material, conjugation of parts, heat resistance

Introduction

The coupling of parts, nodes, systems and aggregates of machines in agricultural production [1,2] work in difficult conditions of sign-changing cyclic and dynamic load, increased dustiness, interaction with active and aggressive working (technological) environments, and therefore do not produce the planned resource and 80...90% of their failures are caused by friction and wear. The development of methods and measures to increase their wear resistance, such as tribotechnical systems, control of processes and states, require the identification of patterns of interaction, the mechanism of friction and wear, a set of characteristics and properties of materials.

Among the main directions of increasing the wear resistance of tribocouplers of parts, the following deserve attention: improvement of the design of parts and their joints; use of advanced materials and working (technological) environments; development of effective technologies for strengthening, restoration and modification of materials of parts, modification of working (technological) environments with substances (fillers, additives, additives, etc.) and treatment with physical fields (electric, magnetic, electromagnetic, laser and ultrasonic radiation, etc.); improvement of technologies of accelerated practice of conjugations of parts; improvement of operating conditions and selection of rational modes of their operation; development of new tribotechnical methods and technologies for ensuring reliable operation of parts, their couplings and machines as a whole [3].

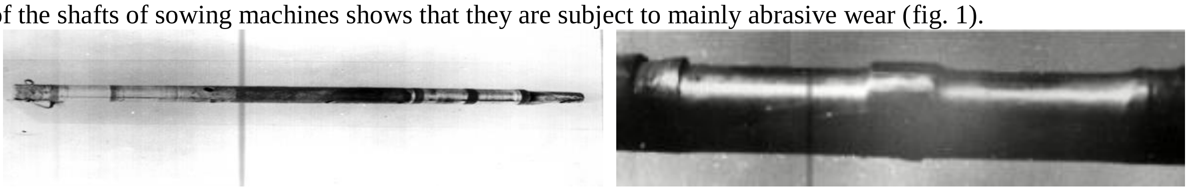
The performance of restored external cylindrical surfaces of machine parts with polymer (PC) and polymer-composite coatings (PCC) largely depends on ensuring reliable heat removal from the friction zone into the part and the environment, because the operational heat resistance of polymers is very low (kapron – 383 K, polyamide P-68 – 403 K, fluoroplastic-4 – 413 K) [4,5].

The intensity of heat dissipation is determined both by the geometry and phase ratio of the PCC components, and by the thermophysical characteristics of the coating materials and conjugated parts [6].

At the same time, the task of PCC efficiency is realized from both theoretical and experimental points of view, i.e. the friction conditions and such tribotechnical characteristics of the working surfaces of the parts as wear, friction coefficient and temperature in the friction zone are determined. It matters on which surfaces of the parts' conjugations the PC and PCC were formed. Most often, PC and PCC are applied to the working surfaces of the coupling of "shaft-sleeve" parts, that is, direct and reverse pairs. If PC or PCC is applied to the shaft, the tribocoupling is a reverse pair. The theoretical substantiation of the evaluation of the tribotechnical characteristics of the surfaces of the tribocoupling parts requires the use of the theoretical provisions of B.I. Kostetskyi [7,8], as well as the establishment of the balanced mode of tribocoupling "shaft-sleeve" friction.

Literature review

It is known [9-11] that sowing machines work in conditions of direct contact with the soil and high dustiness, which causes intensive wear of their parts: quick failure of cuffs and seals is observed; shafts coupled with bushings [1,2]; axes of ninety- and forty-toothed cogs; tension sprocket and marker, drive shafts of fat seeding machines; conjugation "thirteen-tooth gear shaft"; gear mechanism shafts, etc. Analysis of the nature of the wear of the shafts of sowing machines shows that they are subject to mainly abrasive wear (fig. 1).



a b

Fig. 1. General view of the operation of the drive shafts of the fat seeder (a) and the transmission mechanism (b) when the ССТ-12Б 480 ha seeder is working

According to the statistical data on the wear of the shafts of sowing machines, it was determined that the distribution of the amount of wear most closely corresponds to the normal law and the Weibull-Hnedenko law [12, 13]. The limit value of the wear of the shafts leads to the occurrence of excess force during their rotation in the bushings, as a result of which the cotter pins are cut or the connecting brackets of the drive of the seeding device are broken, which leads to the complete loss of performance of the seeding machines. The analysis of studies on the amount and nature of wear of the shafts of sowing machines [7, 14] shows that they have low wear resistance and require strengthening during manufacture.

The wear resistance of moving couplings of parts, including the "shaft-sleeve", is one of the most important factors limiting the reliability of machines. The speed of their wear depends on the following factors: load (contact pressure), temperature (volume average and contact), type and mode of movement, rotation frequency, aggressive action of working (technological) and external environments. To ensure their efficiency, the rational choice of the material of the parts, the physico-chemical and mechanical properties of their surface layers, the rheological and physico-chemical characteristics and properties of the working (technological) environment, etc., are of decisive importance.

The analysis of the constructions of movable conjugations of parts that create direct and reverse pairs - tribotechnical systems [15] allows us to identify four conditions of their existence (table 1).

Table 1

Conditions of existence of direct and reverse TTS and their characteristics

The type of movable conjugations of parts	The ratio of the hardness of the materials of the parts*	The ratio of areas of friction zones of parts surfaces*
Straight	$H_p > H_н$	$S_p > S_н$
Back by materials	$H_p < H_н$	$S_p > S_н$
Inverse by geometry	$H_p > H_н$	$S_p < S_н$
Reversible in terms of materials and geometry	$H_p < H_н$	$S_p < S_н$

*H_p, $H_н$, S_p, $S_н$ – *hardness* and areas of friction zones of moving and stationary parts .

Direct tribocoupling of parts is a widespread construction, in which the material of the moving part has a higher hardness and a larger area of the friction zone, and the stationary part has a correspondingly lower hardness and a smaller area of the friction zone. The intensity of wear of hard and soft materials of reverse tribocoupling parts is the same in terms of materials and geometry. They are characterized by the "cutter" effect, that is, the process of introducing a solid layer with a smaller friction area. Schemes of reverse and direct tribocoupling of parts, when they are strengthened by RS and RSS, are shown in fig. 2.

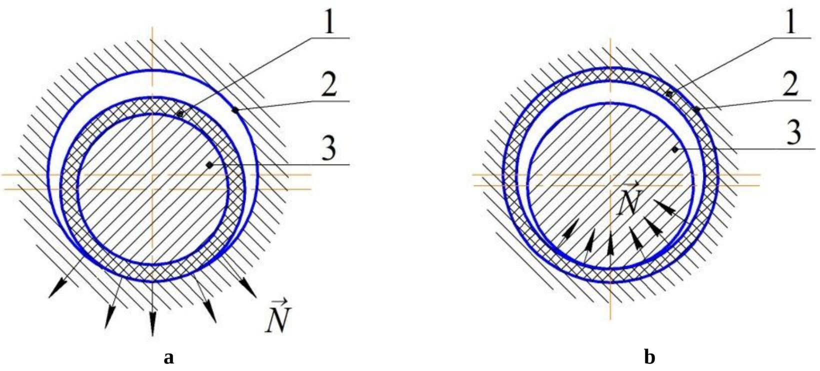


Fig. 2. Schemes of reverse (a) and forward (b) tribocoupling of parts: 1 – layer of PC or PCC; 2 – sleeve; 3 - shaft.

The displayed different nature of force influence in the reverse and forward tribocouplers determines the difference in the conditions of their friction and wear. This is evidenced by the analysis of the operation of "shaft-hub" couplings, characteristic of seeders [1,2] and other machines [16-21] .

Purpose

The purpose of this work is a theoretical-experimental comparative tribological study of the coupling of "shaft-sleeve" parts reinforced with polymer (polyamide P-68) and polymer-composite coatings (based on polyamide P-68 with kaolin filler).

Results

The tribocoupling of the "shaft-hub" parts, which simulate such friction nodes as sliding bearings and cylindrical joints, is schematically depicted in fig. 3.

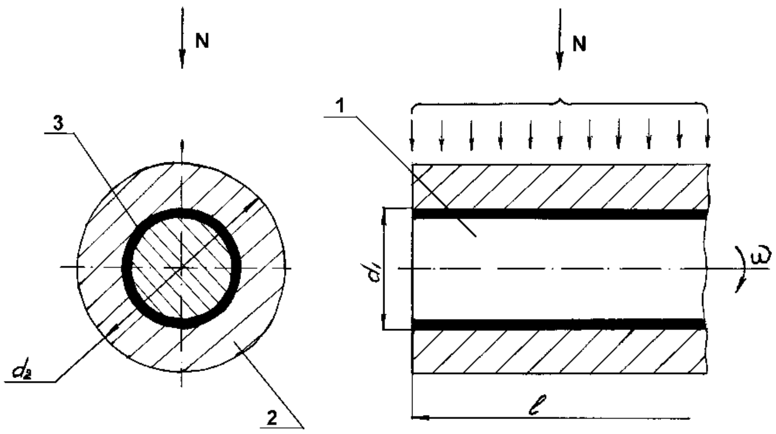


Fig. 3. Scheme of contact tribocoupling of parts with a polymer coating: 1 - shaft; 2 - sleeve; 3 – polymer coating.

The contact pressure in the tribocoupling of parts can be estimated by the formula [22,23]:

$$P_{st} = 0{,}8\left(\frac{N}{l \cdot K_\Sigma} \cdot \frac{\delta}{d_1(d_1 - \delta)}\right)^{1/2}, \tag{1}$$

where N – the normal load; d_1 – the diameter of the cylindrical surface; l – the length of the cylindrical surface; δ – thickness of the polymer layer; K_Σ – the total elastic constant for the case of contact of deformable conjugate parts.

$$K_\Sigma = K_1 + K_2, \tag{2}$$

where K_1, K_2 – the elastic constant for the material of the shaft and sleeve, respectively.

$$K_1 = \frac{1-\mu_1^2}{E_1};\; K_2 = \frac{1-\mu_2^2}{E_2}, \tag{3}$$

where μ_1, μ_2, E_1, E_2 – are Poisson's coefficients and modulus of elasticity, respectively, of the material of the shaft and sleeve.

Substituting (2) and (3) into (1), we obtain the value of the static force in the friction node:

$$P_{st} = 0{,}8\left(\frac{N\cdot\delta\cdot E_1\cdot E_2}{l\cdot d_1(d_1+\delta)\cdot\left[E_1(1-\mu_2)^2+E_2(1-\mu_1)^2\right]}\right)^{1/2}. \quad (4)$$

During the oscillating motion of the tribocoupling of parts (fig. 4), loaded by an external force, it can be exposed to the conditioned loads generated by the inertia of the oscillating masses.

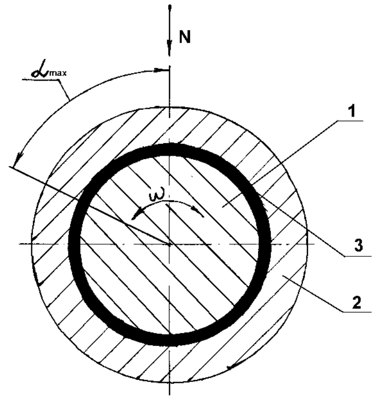


Fig. 4. Scheme of tribocoupling of parts in the oscillation mode: 1 – shaft; 2 – sleeve; 3 – coating.

When increasing the frequency and amplitude of the oscillating motion, the tribocoupling parts are loaded with inertial forces comparable in magnitude to the specified external load.

The lines of action of static and dynamic forces practically coincide [24,25]. At the same time, the amount of the total load acting is equal to:

$$P_\Sigma = P_{st} + P_a \sin\omega t, \quad (5)$$

where P_{st} – the value of the static force determined by expression (4); P_a – the amplitude value of the inertial force.

The relative speed of displacement of the working surfaces of the tribocoupling parts reaches its maximum value when the inertial forces are equal to zero, that is, the phase shift between speed and load fluctuations is equal to 90 °.

Accordingly, the sliding speed is described by a function

$$V = V_a \cos\omega t = \cos\alpha, \quad (6)$$

and the product of the total load on the sliding speed, respectively, is a function of α:

$$P_\Sigma V = V_\alpha\left(P_{st}\cos\alpha + \frac{P_a}{2}\sin\alpha\right). \quad (7)$$

It is worth noting that when using the criterion $P_\Sigma V$, it is necessary to take into account the fact that when the value of the friction coefficient depends on the sliding speed, this affects the heat release in the friction zone, and, therefore, the temperature of the friction surfaces.

To determine the value of the angle $\alpha = \alpha_m$ at which the criterion $P_\Sigma V$ reaches its maximum value, function (7) is differentiated by α_a , and the resulting expression is set to zero and we obtain:

$$\alpha_a = \arcsin\left[-\frac{P_{st}}{4P_a} + \left(\frac{P_{st}^2}{16P_a^2} + \frac{1}{2}\right)^{1/2}\right]. \quad (8)$$

introduce the coefficient η that characterizes the ratio of the product of the load on the sliding speed, taking into account inertial forces, to the product of the static load on the amplitude value of the sliding speed:

$$\eta = \frac{(P_\Sigma V)_{\max}}{P_{st}V_{\max}} = \cos\alpha_a + \frac{P_a}{P_{st}}\sin 2\alpha_a. \quad (9)$$

According to expression (9), when testing bearings for wear, their parameter data were limited so that the magnitude of the inertial force did not exceed one third of the load. At the same time, the criterion $P_\Sigma V$ increases by less than 5%, and therefore it is possible to accept $\eta \approx \cos\alpha_a$.

The magnitude of the amplitude value of the inertial (dynamic) force acting on the tribocoupling unit of the parts is determined by the formula:

$$P_a = \frac{I \cdot \omega^2 \cdot \alpha^2}{S_a}, \tag{10}$$

where I – the moment of inertia of the parts set in motion by the friction node relative to the rocking axis; S_a – the amplitude of swinging of the tribocoupler of parts during oscillations.

In the general case, considering that the friction zone in the tribocoupling of parts extends to part of the surface of the shaft coating S_{tr}. At the same time, the force of friction is equal to:

$$F_{tr} = P_a \cdot f_{tr} \cdot S_{tr}, \tag{11}$$

where f_{tr} – coefficient of friction. With dry friction in the coupling of polymeric materials with steel, the coefficient of friction for kapron is 0.115; for polyamide P-68 - 0.115, and for fluoroplastic-4 - 0.105.

The maximum surface area of the friction zone for a full period is equal to:

– for rotational movement:

$$S_{tr} = \pi d_1 l; \tag{12,a}$$

– for oscillating motion:

$$S_{tr} = 2d_1 \cdot l \cdot \alpha_a, \tag{12, b}$$

where α_a is the amplitude of the angle of deviation from the initial position (Fig. 4), expressed in radians.

Based on the previous one, with a fixed friction process $F_{tr} = const$, the work of friction forces for a full period can be estimated by the formula:

$$A_{tr} = \Omega \cdot d_1 \left(\frac{N \cdot \delta \cdot E_1 \cdot E_2 \cdot l \cdot d_1}{(d_1 + \delta) \cdot \left[E_1 (1 - \mu_2)^2 + E_2 (1 - \mu_1)^2 \right]} \right)^{1/2}, \tag{13}$$

where Ω – the coefficient that takes into account the nature of the movement ($\Omega = 7{,}9$ – for rotational movement; $\Omega = 3{,}2\alpha_a^2$ – for oscillatory movement, α_a, rad).

Based on the second law of thermodynamics, the work of external friction is described by the equation:

$$A_{tr} = Q + \Delta E, \tag{14}$$

where Q – the thermal energy into which the work of external friction was transferred; ΔE is the amount of energy absorbed by the surface layers of conjugated parts.

Well known [26,27] that in a real process, the work of external friction is not completely transformed into thermal energy, and therefore the energy is ΔE not equal to zero. The ratio $\frac{\Delta E}{A_{tr}}$ is, in general, a value that depends on the properties of the constituent tribocouples of materials, the nature of their interaction, the mode of friction and the working environment.

When the friction conditions change, the energy characteristics of this coupling of parts and their ratio also change. If $\frac{Q}{A_{tr}} \to \max$, or $\frac{\Delta E}{A_{tr}} \to \min$, then this characterizes the transitional process from unsteady to steady friction, and, therefore, wear processes when the friction surfaces are not worn. If the friction surfaces of the restored parts are worked in and given the necessary physical and mechanical properties, then already at the first stage of the friction process, the following ratios are realized:

$$\frac{Q}{A_{tr}} \approx 1; \qquad \frac{\Delta E}{A_{tr}} \to \min.$$

The regime of stable friction is also observed when the tribotechnical characteristics of tribocouplers have linear dependencies [7,8]. At the same time, the rate of wear is minimal and sinusoidally fluctuates near some constant value. A dynamic balance of friction and wear processes is also observed here. The specified conditions are the most desirable for tribocoupling parts of nodes, systems and machine assemblies. Under these conditions, their service life will be maximum. Modes of sharp changes in friction characteristics are also observed, in which the work of friction forces increasingly turns into energy absorbed by the surface layers of materials of tribocoupler parts. At the same time: $\frac{\Delta E}{A_{tr}} \to \max$ or $\frac{Q}{A_{tr}} \to \min \to 0$. This leads to a complete change in their physical properties, the nature and type of connections between the conjugated parts. As a result of established friction, wear of parts and dynamic self-regulation in the system of formation and destruction of secondary structures occurs.

Given the given materials of the tribocoupled parts, the nature of the interaction and the working medium, there is a region of change in the friction mode, in which the integral over the volume of the deformed surface layers takes a minimum value:

$$\int_V \frac{\Delta E}{A_{tr}} dV \to \min \tag{15}$$

For polymer-metal couplings, stable friction is observed almost at the beginning of the process, that is, it can be assumed that $A_{tr} \approx Q$ in the friction zone in one revolution, according to expression (13), the amount of thermal energy will be released:

$$Q = \Omega \cdot d_1 \left(\frac{N \cdot E_1 \cdot E_2 \cdot d_1 \cdot \delta}{l(d_1 + \delta) \cdot \left[E_1 (1-\mu_2)^2 + E_2 (1-\mu_1)^2 \right]} \right)^{1/2}. \tag{16}$$

First of all, let's consider the process of heat removal from the tribo-coupling zone of "shaft-sleeve" parts with the corresponding materials (fig. 5). We also assume that the isothermal surfaces in the triboconjugation of parts and the temperature will be a function of only one coordinate *h* - along the normal to the isothermal surfaces.

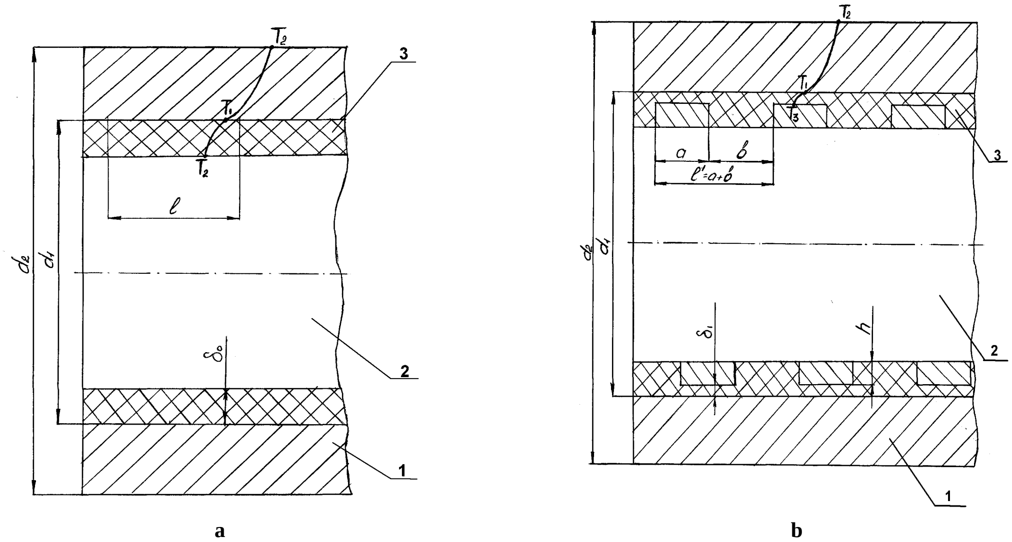


Fig. 5. Scheme of removal of thermal energy from the friction zone for PC (a) and PCC (b): 1 - sleeve; 2 - shaft; 3 – coating.

According to the Fourier law [28,29], the heat flow through the cylindrical surface of the sleeve can be determined by the expression:

$$Q = -\lambda \frac{dT}{dh} \cdot S(h), \tag{17}$$

where $Q = const$ for any isothermal surface; $S(h)$ – heat energy removal surface; λ – thermal conductivity of the part material.

Having integrated equation (17), having previously divided the variables, within the limits of h from h_1 to h_2 and to T – from T_1 to T_2, we have:

$$Q = \frac{\lambda (T_1 - T_2)}{\int_{h_1}^{h_2} \frac{dh}{S(h)}}, \tag{18}$$

where T_1, T_2 – the temperatures in the friction zone and on the outer surface of the sleeve, respectively.

Let's enter the notation

$$\int_{h_1}^{h_2} \frac{dh}{S(h)} = N_{h_1}^{h_2}, \tag{19}$$

where $N_{h_1}^{h_2}$ is the thickness of the wall along which heat dissipation is observed.

Considering (19) in (18), we have:

$$Q = \lambda(T_1 - T_2)/N_{h_1}^{h_2}. \quad (2\,0)$$

If equation (17) is integrated from h_1 to h and from T_1 to T :

$$\frac{Q}{\lambda}\int_{h_1}^{h}\frac{dh}{S(h)} = -\int_{T_1}^{T} dT, \quad (21)$$

then we get the following temperature distribution:

$$T = T_1 - \frac{Q}{\lambda} N_{h_1}^{h}. \quad (2\,2)$$

Substituting the value Q from expression (20), we have:

$$T = T_1 - (T_1 - T_2)\frac{N_{h_1}^{h}}{N_{h_1}^{h_2}}. \quad (23)$$

If we enter the dimensionless temperature $\theta = \dfrac{T - T_1}{T_1 - T_2}$, then the temperature distribution according to expression (23) will take the form:

$$\theta = 1 - \frac{N_{h_1}^{h}}{N_{h_1}^{h_2}}. \quad (24)$$

Let the stationary process of heat conduction be carried out in a cylindrical wall (sleeve) with an inner radius r_1 and an outer radius r_2 and through a polymer coating on the cylindrical surface of the shaft (Fig. 5).

Boundary conditions are set on the sleeve surfaces

$$T|_{r=r_1} = T_1;\; T|_{r=r_2} = T_2, \text{ or } \theta|_{r=r_1} = 1;\; \theta|_{r=r_2} = 0. \quad (25)$$

Based on equations (21) and (22) and fig. 4, we have:

$$h = r; h_1 = r_1; h_2 = r_2; S(h) = 2\pi r l. \quad (26)$$

According to expression (19), the reduced thicknesses of the heat sink walls are equal to:

$$N_{h_1}^{h} = \int_{r_1}^{r}\frac{dr}{2\pi r l} = \frac{1}{2\pi l}\ln\frac{r}{r_1}, \quad (27)$$

$$N_{h_1}^{h_2} = \frac{1}{2\pi l}\ln\frac{r_2}{r_1}. \quad (28)$$

Taking into account the obtained parameters (27) and (28), the expression for the heat flow diverted from the friction zone by the bushing has the form:

$$Q_b = \frac{2\pi l \lambda_1 (T_1 - T_2)}{\ln\dfrac{r_2}{r_1}}. \quad (29)$$

Then the distributions of temperature (23) and dimensionless temperature (24) will be converted into expressions, respectively:

$$T = T_1 - (T_1 - T_2)\frac{\ln(r/r_1)}{\ln(r_2/r_1)}; \tag{30}$$

$$\theta = 1 - \frac{\ln(r/r_1)}{\ln(r_2/r_1)}. \tag{31}$$

If you enter dimensionless coordinates:

$R = \frac{r}{r_1}$, and $\frac{r_2}{r_1} = K_R$, expression (31) can be written in the form:

$$\theta = 1 - \frac{\ln R}{\ln K_R} = 1 - \frac{2\lambda \ln R}{2\lambda \ln K_R} = \frac{\ln R}{2\lambda \xi_{to}}, \tag{32}$$

where $\xi_{to} = \frac{\ln(K_R)}{2\lambda}$ is the linear thermal resistance of the cylindrical wall.

The amount of thermal energy from the friction zone of the tribocoupling parts is removed both through the sleeve and through the polymer coating on the cylindrical surface of the shaft.

Estimate these heat flows as you can according to the obtained expressions (29) and (16), based on the thermal energy released in the friction zone.

Let the amount of heat be removed through the sleeve

$$Q_b = \gamma Q, \tag{33}$$

where γ – the coefficient that takes into account part of the heat removed through the sleeve from the friction surface. Part of the heat dissipated through the polymer coating on the cylindrical surface of the shaft is equal to:

$$Q_{pc} = (1-\gamma)(1-\beta)Q, \tag{34}$$

where β – is the coefficient that takes into account part of the thermal energy dissipated in the tribo coupling of parts.

Let us have a section of the shaft l with a uniform polymer coating of thickness δ_p (fig. 3).

Then, according to expressions (18), (29) and (34), the heat flow is transferred to the shaft with a polymer coating:

$$Q_{pc} = (1-\gamma)(1-\beta)\frac{2\pi l_p \lambda_2 (T_1 - T_2)}{\ln \frac{d_1}{d_1 - 2\delta_p}}, \tag{35}$$

where λ_2 – the thermal conductivity of the polymer material.

In case of combined PCC:

$$Q_{pcc} = (1-\gamma)(1-\beta)\frac{2\pi(a\lambda_3 + b\lambda_2)(T_1 - T_2)}{\ln \frac{d_1}{d_1 - 2\delta_{PCC}}}, \tag{36}$$

where a – the length of the metal section of PCC with thermal conductivity λ_3, b – the length of the polymer section of PCC with thermal conductivity λ_2, δ_{PCC} – the thickness of PCC.

On the МИ-1М friction machine, in accordance with the standard methodology [ ] , studies of the wear characteristics of the friction surfaces were carried out (PC – pure polymer coatings (polyamide P-68), PCC – polymer-composite coatings based on P-68, cast iron СЧ18, steel 45) without lubrication and with extreme friction from specific load and sliding speed.

Experimental values of the values of linear wear depending on the duration of tests without lubrication and with extreme friction are presented in Figs. 6 and 7.

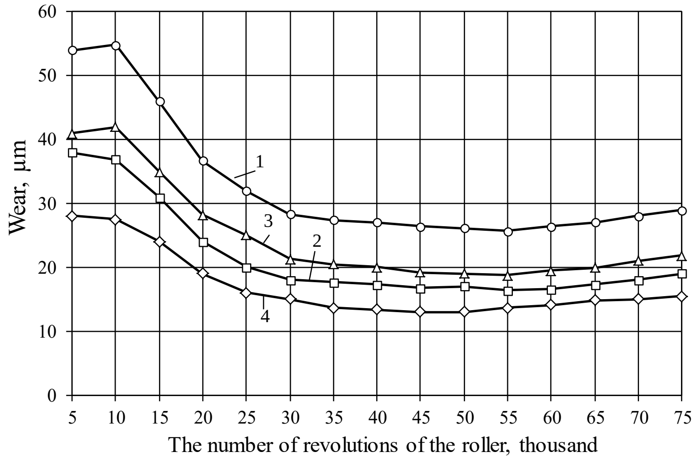


Fig. 6. Dependence of the amount of wear of tribo-coupling friction surfaces of "roller-pad" samples without lubrication depending on the duration of the test (P = 1.5 MPa, V = 0.5 m/s). Tribocoupling of materials: 1 – "cast iron-PC", 2 – "cast iron-PCC", 3 – "steel-PC", 4 – "steel-PCC".

The tests were carried out at a specific load of 0.5 MPa and a sliding speed of 0.5 m/s without lubrication and at extreme friction. The linear wear of the samples was determined after every 5000 revolutions, which corresponded to 628 m of friction path. The total duration of the tests included both a run-in period and a period of steady wear.

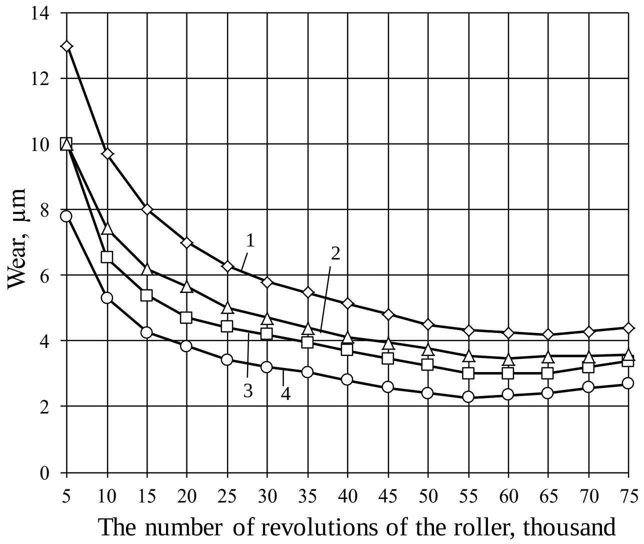


Fig. 7. Dependence of the amount of wear of the "roller-pad" tribocoupling surfaces at the limit friction depending on the duration of the test (P = 1.5 MPa, V = 0.5 m/s). Tribo-coupling of materials: 1 – "cast iron-PC", 2 – "cast iron-PCC", 3 – "steel-PC", 4 – "steel-PCC".

In each test, linear wear, friction moment and temperature in the friction zone were determined. In comparative tests, it was determined that after 50,000 revolutions, the wear process both under conditions of extreme friction and without lubrication occurs with constant intensity. Therefore, the wear results obtained after 50,000 revolutions of the roller were used for a comparative assessment of the wear resistance of the surfaces of the studied samples.

In order to evaluate the wear resistance during the period of established wear, the criterion is the relative wear resistance, which is determined by the ratio:

$$\gamma = \frac{U_e}{U_a},$$

where U_e – the linear wear of the reference surface for a certain number of revolutions of the roller, µm; U_a – absolute wear of the tested surface for the same number of roller revolutions, µm. The indicated values were determined during the period of established wear, that is, from 50,000 to 75,000 roller revolutions, according to the data shown in the graphs. Cast iron was used as the standard for determining the relative wear resistance of tribocoupling samples. The results of relative wear resistance at friction without lubrication and at limit friction are presented in table 2.

Table 2

Relative wear resistance of surfaces in friction without lubrication and in extreme friction during the period of established wear (P=1.5 MPa, V = 0.5 m/s)

Surface	Relative wear resistance,γ	
	without lubrication	marginal friction
PC	3.11	2.00
PCC	4.66	3.33
Cast iron СЧ18	1.00	1.00
Steel 45	1.36	1.25

The results given in the table. 2 show that the relative wear resistance of PCC samples in the mode of friction without lubrication is 1.5 times greater than the relative wear resistance of PC samples and 3.4 times greater than that of steel 45 samples. For limit friction, the relative wear resistance is respectively greater than 1, 3 and 2.7 times.

Differences are also observed in the nature of the wear of the conjugated surfaces. The results of experimental studies of the wear of the roller and pad for the tested samples under friction without lubrication and under extreme friction are given in table 3.

Table 3

The nature of the wear of the coupled surfaces in friction without lubrication and in marginal friction during the period of established wear (P=1.5 MPa, V = 0.5 m/s)

The studied surface	Amount of wear, μm					
	without lubrication			marginal friction		
	roller	pin feather	couple	roller	pin feather	tribocoupling of samples
Cast iron-PC	27	32	59	5	11	16
Cast iron-PCC	18	26	44	3	9	12
Steel-PC	20.8	24.6	45.4	3.8	8.5	12.3
Steel-PCC	13.9	20	33.9	2,3	6.9	9.2

The analysis of the experimental data showed that it is characteristic for the studied samples: the pad wears out more intensively compared to the roller. The total wear of "cast iron-PCC" and "steel-PCC" friction pairs is 1.3...1.4 times less than the wear of "cast iron-PC" and "steel-PC" pairs in friction conditions without lubrication and 1.2 ...1.3 times less when working in conditions of extreme friction.

The presence in the friction zone of a polymer material with a high molecular weight and a low activation energy of mechanodestruction [5,30-32] has a favorable effect on the reduction of the working-in time of the couplings, which increases the degree of dispersion of the micro-uniformities of the surface being worked-in.

Thus, polymer materials applied to the surface to be restored help speed up the run-in process, reduce initial wear, and, as research shows, surface roughness decreases.

The study of temperature changes in the friction zone was carried out by comparing heat dissipation with pure polymer and polymer-composite coatings. The temperature was measured according to the method described in [3]. In addition, the intensity of heat removal from the friction zone was studied.

The results of measurements of the temperature of the examined surfaces of the samples and its theoretical assessment are given in table 4.

Table 4

Temperature in the friction zone (P=1.0 MPa, V = 0.5 m/s)

Surface	Temperature, *K*			
	Experimental data		Theoretical evaluations	
	without lubrication	marginal friction	without lubrication	marginal friction
PC	378	364	376	361
PCC	365	347	363	344
Cast iron СЧ18	388	375	384	371
Steel 45	385	372	383	368

According to the data in the table, the heat resistance of PCC is higher than pure polymeric PC, which indicates the feasibility of using the developed technological process to increase the wear resistance and heat resistance of the range of parts that work as sliding bearings.

For polymer materials, there is a fairly clear relationship between the friction coefficients and the temperature in the contact zone: lower temperatures correspond to a lower value of the friction coefficient and vice versa. The temperature arising as a result of friction changes the elastic and strength properties of the polymer surface layers of tribocouples of samples and parts. This affects the change of the actual contact area of the surfaces and the force of friction, and, therefore, the coefficient of friction.

The change in temperature observed in the friction zone is due to the more intense heat dissipation of polymer-composite coatings compared to pure polymer ones.

Conclusions

1. Analytical expressions for static, dynamic and total forces were obtained for tribocoupling of "shaft-sleeve" parts. The assessment of the work of forces and thermal energy in the tribocoupling of parts is determined for the established friction process.

2. From a theoretical point of view, the effectiveness of heat dissipation from a polymer-composite coating in comparison with polymer coatings has been proven. Patterns of change in thermodynamic and dimensionless temperature changes were found for the investigated coatings.

3. A study of the wear of the working surfaces of the "roller-pad" samples was carried out on the МИ-1М friction machine. Polymer coatings, polymer-composite coatings, cast iron СЧ-18, steel 45 were subjected to research. The dependence of the wear of the conjugation of the samples on the number of revolutions of the roller in the modes without lubrication and with extreme friction was determined. It is shown that the relative wear resistance of PCC samples in the mode of friction without lubrication is 1.5 times greater than that of PC and 3.4 times greater than that of samples made of steel 45. For extreme friction, the relative wear resistance is correspondingly greater by 1.3... 2.7 times.

4. The results of temperature measurements in the friction zone of the sample joints indicate that the heat resistance of the polymer-composite coating based on polyamide P-68 with kaolin filler is higher than the pure matrix polymer material both in the mode without lubrication and in the mode of marginal friction.

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Аулін В.В., Лисенко С.В., Гриньків А.В., Пашинський М.В. Покращення трибологічних характеристик спряження деталей "вал-втулка" полімерними та полімерно-композиційними матеріалами

В статті дано аналітичне обґрунтування протікання трибологічних процесів спряження деталей "вал-втулка", яке моделює функціонування підшипників ковзання і циліндричні шарніри машин. Основна увага приділена таким характеристикам як контактний тиск, статичному та динамічному зусиллям, критерію добутку сумарного тиску на швидкість ковзання, роботи сил тертя й її переходу в теплову енергію в зоні тертя для полімерних (на основі поліаміду П-68) і полімерно-композиційних покриттів (на основі П-68 з наповнювачем каоліну) на робочих поверхнях деталей. Наведено порівняльний аналіз функціонування і трибологічних характеристик спряжень деталей без покриттів.

Експериментально, на основі випробувань зразків на машині тертя МИ-1М, доведено істотне зменшення зносу та підвищення відносної зносостійкості зразків з полімерно-композиційними покриттями в режимах тертя без змащення (в 1,3...1,4 рази) і граничному терті (в 1,2...1,3 рази), а також зменшення температури у зоні тертя (365 К і 347 К) у порівнянні з полімерним покриттям.

Ключові слова: трибологічна характеристика, полімерний матеріал, полімерно-композиційний матеріал, спряження деталей, термостійкість.