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Theory and experiment of tribological test methods

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Abstract

Based on the analysis of the current state of test methods for wear of friction pairs and the need for methods with certain operating conditions, the problem of developing a theory of test methods for wear of friction pairs according to the ball-cylinder scheme was solved, which makes it possible to determine the parameters of wear models and the general characteristics of the wear resistance of materials. It is shown that the type of wear within the range of properties of the friction pair parameters ensures compliance with the basic conditions in contact: materials, lubrication, pressure, speed, temperature, type of movement, and other less significant factors. It has been established that one of the fundamental issues in the development and conduct of wear tests is to take into account the influence of the stationarity of the wear mode at a point. Test methods in steady and unsteady modes (transient mode) are different. The test method should be based on the solution of the contact problem for a friction pair sample-counter-sample. Based on the solution of wear-contact problem for the "ball-cylinder" scheme, a theory has been developed for identifying the parameters of the wear pattern. To solve the inverse problem, a method of approximating function is proposed and implemented. The power approximation of the experimental function after substitution into the resolving equation gives simple expressions for calculating the model parameters. The results obtained make it possible to predict the intensity of wear of tribocouples under given initial operating conditions: according to loads, sliding speed, characteristics of lubricants and structural materials.

Keywords: wear, friction pairs, tribological tests, contact task

1. Modern approaches to the technique of tribological tests. Experiment, theory, methods, equipment.

A review of research on modern highly cited periodicals in the field of friction and wear regarding the problems of tribological testing is proposed.

The article [1] presents the development, manufacture and testing of an experimental bench for assessing the fretting fatigue of conductor materials in power lines. The friction machine has three independent drives that allow testing under a controlled load. This makes it possible to study the influence of relevant parameters on the fretting fatigue process (such as normal, tangential and body forces, wear level and wear surface morphology).

In [2], the technological problem of testing jet engine parts under conditions close to operational ones is solved. A bench for testing blade friction is proposed to study the occurring phenomena and the behavior of rotor blades during contact interactions between the blades and the engine housing. A feature of the presented installation is the possibility of increasing the number of working elements of the chamber, which makes it possible to produce more than one contact per revolution. This article discusses the test setup and its components, the experimental procedure, and examples of results.

In [3], laser microscopy was used to quantify wear mechanisms. The results showed that for some mechanisms it is possible to obtain quantitative data for the classification of wear and tear in accordance with ASTM G98. Some suggestions for modification and improvement of the standard are offered.

In work[4]. The study of the brake pad for friction and wear is carried out on an inertial brake dynamometer at various temperatures (200°C, 250°C and 300°C). The actual field tests are carried out on four different trucks, namely a city bus (CB), a high speed bus (HSB), a highway truck (HWT) and a dump truck (TL). This study will be useful for developing the composition of the friction material and predicting the actual life of the brake pad for various trucks.

The article [5] presents experimental setups for conducting research on real machine components and obtaining more realistic results than conventional tribological test benches. Slope pad plain bearings, as well as gears and complete gearboxes for advanced industrial applications, can be tested using the benches described in the article. A double disc machine and a closed loop gear test bench are used to investigate the various wear mechanisms that occur in gears.

The authors of the study [6] study the problem of reducing friction and wear in metal-metal systems, as well as in some mechanical components. This paper presents a computational study based on Archard's linear wear law and Finite Element Method (FEM) modeling for dry sliding wear analysis in pin-on-disk testing. The Archard wear model was implemented in a FORTRAN user routine (UMESHMOTION) to describe sliding wear. The modeling of wear particle formation mechanisms was taken into account using the overall wear factor obtained from experimental measurements. In such an implementation, a step-by-step calculation of surface wear based on nodal displacements is considered using adaptive mesh tools that rebuild local nodal positions. Numerical and experimental results were compared in terms of wear rate and friction coefficient. In addition, during numerical simulation, the distribution of the stress field and the change in the surface profile across the worn track of the disk were analyzed.

In the study[7], based on the classical theory of adhesive wear by Archard, a three-dimensional finite element model was created to simulate the process of destruction of self-lubricating spherical plain bearings under conditions of oscillatory wear. The results show that self-lubricating spherical plain bearings go through two different stages in the wear process, namely an initial wear stage and a stable wear stage. The moment of friction at the initial stage of wear is somewhat less than the moment of friction at the stage of stable wear; however, the wear rate in the initial wear stage is high. The developed finite element model can be used to analyze bearing wear mechanisms and predict bearing life.

The article [8] experimentally and theoretically investigates the effect of lubrication on stresses at high temperature. A statistical formula has been developed to predict the Striebeck curves given the interaction of asperities with lubricant on the contact surface. It has been found that increasing the temperature at the interface can significantly change the performance of the lubricant due to the influence of the lubricant additive and that the conventional assumption of a constant coefficient of friction for a rough contact is not practical. As the temperature increased, the boundary layer formed by the lubricant molecules dissolved, and a soft solid layer was formed by lubricant additives at the contact boundary. It was also found that in the boundary lubrication regime, friction depends on the sliding speed due to the formation-destruction cycle of a soft hard layer,

It is noted in [9] that the characteristics and service life of structural materials at high temperatures are largely affected by the degradation of the material during wear and corrosion. To evaluate material performance at high temperature, it is necessary to understand the processes and their interactions to select and develop new advanced materials. In the present study, an experimental setup for testing and high-temperature abrasion is presented, taking into account the oxidation process.

In [10], the change in temperature over time in a deep groove ball bearing in an oil bath lubrication system was experimentally and analytically studied. The test device is a radially loaded ball bearing equipped with instruments to measure the friction torque as well as the transition temperature of the outer ring, oil and housing. The developed mathematical model provides a comprehensive thermal analysis of a ball bearing, taking into account frictional heat release, heat transfer processes and thermal expansion of the bearing components. Experiments are carried out for various speeds and loads to test the model. It has been established that the predicted temperatures at various loads and speeds agree well with those measured experimentally.

The article [11] discusses the optimization of the tribological characteristics of lubricating oils used in the processing industry. Optimum lube oil performance is determined by multivariate optimization of viscosity, temperature and oil film thickness. Experimental test results are shown for the oil used in the real plant and the new oil obtained from the simulation.

Reference [12] presents a methodology for deriving pressure-dependent friction laws from experimental tribometric testing of rubber-to-metal contacts. Tribometric tests are simulated to analyze the contact area and obtain an approximation of the contact pressure distribution. The proposed methodology is recommended for evaluating the experimental results of tribometric tests carried out according to the "flat surface" and "cylinder surface" schemes. Planar and cylindrical materials have been fabricated with very similar surface morphology and therefore theoretically the same pressure dependence of the law of friction.

Article [13] contains a description of the test bench and the first experimental data concerning the mechanical and thermal characteristics of a rotor supported by air-lubricated bearings under external pressure. The stand allows you to load the rotor radially and axially in both static and dynamic modes, as well as measure the orbits on the supports. Rotating unbalance response at various speeds has been measured up to 60,000 rpm. The rotor is stable and has no vortex instability.

The authors of this work analyzed the methods for identifying the parameters of wear patterns for various wear models and test schemes [14-17].

The above review of the literature confirms the versatility of tribological testing methods in terms of techniques, test conditions, materials, design of benches and samples, modeling procedures and processing of experimental results.

2. Basic principles of tribological testing

2.1 Duration of tribological tests

Wear reduction is carried out mainly in three directions: 1) the creation of wear-resistant surfaces based on the laws of tribomaterials science; 2) creation of lubricants, lubricating additives and lubricating systems; 3) creation of methods for reliable prediction of wear of components based on theoretical tribology.

With regard to friction units, the stages of a rational test cycle are formulated as follows: 1) selection of combinations of materials based on a priori information; 2) identification of the boundaries of compatibility of a friction pair in terms of the determining parameter; 3) modeling on samples of the operating conditions of tribocouplings; 4) full-scale simulation on the stand, taking into account the main operational parameters of the tribocouplings; 5) full-scale modeling, taking into account the typical operating conditions of the friction unit in the machine during operation.

If wear tests of a friction pair are carried out strictly under the operating conditions of the unit, taking into account the statistical nature of wear, then the duration of the testing process for a statistically representative number of samples can last for months and even years. There are various methods for forcing tests in order to speed up the process. It is shown that without building reliable mathematical models of nonlinear wear processes, the use of accelerated wear testing methods can lead to errors. Based on the considered models, the following methods of accelerated forced tests are proposed: 1) extrapolation in time; 2) extrapolation by load; 3) the principle of requests; 4) interpolation on intervals; 5) extrapolation on intervals; 6) double consecutive extrapolation; 7) double parallel extrapolation, etc.

The analysis of the methods shows that the main success in the application of various accelerated methods is determined by the reliability of the mathematical model underlying the process. The friction unit of any machine, regardless of the designer, technologist or operator, goes through at least two stages during operation: the running-in stage and the stage of steady wear. When developing methods for accelerated wear testing of materials, it is necessary to take into account the presence of transient processes and running-in at the initial stages.

2.2. Test schemes and conditions in contact

The decisive role in the validity of the laboratory test method is played by the geometry of the contacting elements. Geometric schemes are systematized or simply given in many reference books and articles. In most of the frequently used test schemes, for example, ball-plane, etc., none of the publications provides formulas for calculating contact pressures, the main parameter of test conditions. This is in most cases one of the main causes of scatter and non-reproducibility of test results. On the other hand, the lack of information about the change in pressure conditions during the test devalues the results, making them estimates, as in the case of tests on a four-ball.

Can be offereda list of factors affecting the difference between the rated contact voltages and those calculated under standard conditions. Consideration of this list gives an idea of the reasons for the dispersion of wear test results according to different test schemes: 1) subsurface stress concentrators: grain boundaries, twins and other concentrators; 2) the nature of the surface: topography, texture, residual stresses; surface energy level; microstructure, pollution; 3) surface defects: inclusions and particles of the secondary phase; dents and scratches; grooves and risks; corrosion, rust; traces of etching; fretting traces; slip marks; 4) discontinuities in contact geometry: ends of line contacts; wear particles in the contact area; 5) load distribution inside the bearing: elastic deformations; distortion of parts; clearance in contact; 6) elastic-hydrodynamic lubrication; 7) tangential forces: no slip; sliding rolling.

The first three main factors that determine wear are: pressure - P, sliding speed - V and contact temperature T. Accounting for these three factors requires a three-factor experiment. The planning of such tests can be carried out according to the well-known mathematical theory of experiment planning.

The environment in which the friction unit operates has a decisive influence on the scheme and test method. When testing friction pairs in the presence of lubricants, their ball circuit is used. The criterion for the quality of the lubricant here is the bulk temperature at which the contact oil layer is destroyed, which is fixed by a sharp increase in the friction coefficient. However, this method, in terms of physical meaning, is the furthest (in comparison with other methods) from reproducing the actual pattern that occurs in contact during friction. It is proposed to give preference to a flat contact as the most specific and corresponding to real nodes.

Obviously, to a large extent, negative conclusions about the four-ball scheme are explained by the lack of methods for estimating contact pressures and neglecting wear. Improvements in the accuracy of determining the anti-wear properties on a four-ball machine can be obtained using the pre-print method.

It is experimentally shown on a four-ball machine that the volumetric wear of a spherical sample is a single-valued linear function of the work of the friction forces. A linear dependence of wear on pressure was also obtained. Obviously, the results obtained are of a particular nature and are valid for specific test conditions.

BThe large discrepancy in determining the results of wear tests by various methods in the absence of lubrication and abrasive effects, according to studies performed in the USA, is due to factors:

1) heterogeneity and instability of surface morphology and wear marks;
2) the presence of particles on the sliding surface as a third body;
3) dynamic rigidity of the installation, as a factor leading to load fluctuations;
4) the shape of the sample or test design leads to different coefficients of variation of the results.

2.3. Wear measurements, testing machines

The wear measurement method is an integral part of the wear test method. The literature describes in detail the methods of artificial bases: the method of imprints, the method of punched imprints, the method of cut holes, and the concept of the method of radioactive isotopes is given. All methods are divided into periodic or discrete and continuous without stopping the machine. Among discrete methods, methods are distinguished: 1) micrometric measurements; 2) weighing; 3) profiling; 4) artificial bases. Continuous methods: 1) according to the content of wear products in used oil; 2) radioactive isotopes; 3) pneumatic micrometering; 4) according to the flow rate of the working medium through the slots between the rubbing surfaces; 5) lever devices and indicators; 6) mesdoses; 7) inductive sensors: 8) strain gauge micrometering.

Any testing machine can be considered as a system consisting of two main parts: 1) testing unit: sample, counter-sample and their fastening; 2) everything else: drive, gears, electrical part, loading, etc.

The study of the parameters of wear particles provides such information about the nature and intensity that it is difficult to overestimate. Obviously, this branch of tribology is still waiting for its intensive development. Photographs of the surfaces of wear particles show that their structure and geometry practically repeat the surface layers of the bodies. By the type of particles, one can judge the stage of wear. Information about particle morphology can be obtained using a bichromatic microscope. The shape of the wear products themselves corresponds to the implemented friction mechanisms. For normal wear, the particles are lamellar; for abrasive - the form of microchips; Fatigue fracture mechanisms correspond to particles of rounded geometry or even spherical ones.

At present, wear particle analysis is widely used to monitor the condition and predict the failure of lubricated components in machines. The parameters of the wear particles reflect both the nature and degree of wear of the rubbing surfaces. Methods and devices for diagnosing the state of machines based on wear particles distinguish between external and built-in. An analysis of patents and publications points to the intensive development of built-in ferrodiagnostics devices. In this case, a variety of principles are used: spectral analysis, ferrography, light scattering, in-line ultramicroscopy, activation methods.

2.4. Wear contact tasks

Contact pressures are one of the first three main factors (pressures σ ,temperature T , sliding speed V), determining the intensity of surface wear for given materials and lubricants.

The role of pressure here is close to the role of the stress state in the problem of ensuring strength. In problems of contact strength, contact pressures are also defined as the first stage in solving a strength problem. At the second stage, the stress state is determined, at the third stage, using the criteria of strength, ductility or fatigue, the question is decided whether the structure will fail or not. The logic of assessing the state of the friction unit in terms of wear is somewhat different. At the first stage, for a given pair of friction and lubrication, the dependence of wear is studied u_w or wear rate du_w/dS from the main factor - pressures and friction paths, the parameters of this dependence are determined.

At the second stage, the contact problem is solved taking into account the wear for the considered friction unit, for example, for a plain bearing, and the dependence of the wear of this unit on the load dimensions and the found model parameters is determined.

At the third stage, this dependence, taking into account the parameters of the model, calculates the wear of the assembly and compares it with the allowable wear. This stage can be performed in a deterministic setting based on average values or in a probabilistic setting with an estimate of the reliability of the node.

In the general case, the statement of the contact problem for two bodies, taking into account wear, consists of the following equations and conditions. The first basic basic relation is the condition of continuity in contact in algebraic form:

$$u_e(x,y,s)+u_w(x,y,s)=u(x,y),$$

where $u(x,y,S)$ -full elastic normal contact displacements of both bodies; x,y -coordinates, s -friction path.

The second basic equation in the formulation of the problem is the equation of equilibrium of contacting bodies:

$$Q = \int_F \sigma(x, y, S) dF ,$$

where Q -the total load acting on the bodies (the projection of the load on the vertical axis); $\sigma(x, y, S)$ - projection of contact pressures on the same axis; F -contact surface area.

The third basic relation of the contact problem with wear is the equation of the wear model in differential form:

$$\frac{du_w}{ds} = F(\sigma, s)$$

or in integral form:

$$u_w(s) = \int_0^S F(\sigma, s) ds .$$

The complexity of the final equation and the success of solving the problem as a whole largely depend on the degree of complexity of the wear model.

The development of methods for solving contact problems with wear taken into account has gone from the simplest problems to the refinement of models and the complexity of solutions. The wear model is a differential (or integral) relationship between wear and the main factors. In this paper, of these factors, only the influence of pressure is studied.

The test method should be based on the solution of the contact problem for a friction pair sample-counter-sample. Rigorous solutions of this problem with allowance for compliance are complex. It is promising to use a calculation scheme in which the contacting bodies are rigid. With sufficiently large wear, neglect of elasticity is permissible.

Based on the analysis of the current state of test methods for wear of friction pairs and the need for methods with uniquely defined operating conditions for contact pressures, the following task was set and solved in this work: development of a theory of a test method for wear of friction pairs according to the ball-cylinder scheme, which makes it possible to uniquely determine the parameters wear patterns and general characteristics of the wear resistance of materials.

2.4. Wear models

Models (law) of wear processes are dependences of the intensity of wear (or wear) on the determining factors of the process, which are used in the design of friction nodes, forecasting their wear and to optimize structural, technological and other parameters. The pattern of wear can be applied in practice, if the algorithm for determining the parameters of this pattern is known. It is the quantitative parameters that make it possible to assess the influence of determining factors (pressure, temperature speed). The parameters can be set only based on the results of experimental tests, which makes the model close to the real operating conditions of the friction unit.

The theory of determining the parameters of wear models is developed on the basis of the solution of inverse wear-contact problems (that is, when the dependences for the calculation of parameters are determined according to the accepted mathematical form of the law of wear, geometric ratios (conditions of continuity in contact), equilibrium conditions.

The more determining factors in the model, the more difficult the solution. So already two parameters (for example, pressure and speed) significantly complicate the task and require certain assumptions.

In this work, a conceptual methodology is considered, one of the main tasks of which is the development of the theory of test methods for the identification of wear parameters.

It is proposed as the most appropriate test schemes to use schemes when during the tests the contact pressure changes due to a change in the contact area, which makes it possible to have results for the pressure range based on the results of the tests of one sample. Such test schemes include: four-ball scheme, sphere - ring, cone - ring cross cylinders, cylinder - plane, sphere - plane and others.

For the tests, the scheme is adopted, which, according to geometric and technological features, most closely corresponds to the real tribo-coupling. Thus, the wear models obtained by the four-ball scheme should be used to evaluate the wear resistance of connections in which contact is made along a line or at points (gears, cam mechanisms) with small contact area sizes. For connections in which the dimensions of the contact area are commensurate with the dimensions of the contacting bodies (sliding bearings, ball bearings), other test schemes should be used: "cone-ring", "sphere-ring", which more adequately correspond to real contact.

At the initial stages of solving this problem, the theoretical foundations of test methods were considered only for the contact pressure factor. In this work, methods of the theory of tests for a larger number of determining factors for the above-mentioned test schemes with a variable contact area are developed. This made it possible to evaluate the influence of speed and temperature factors on the wear process.

Having a developed theoretical apparatus, tests of the friction unit are carried out in conditions close to real ones (in terms of materials, lubrication, temperature, etc.) and the parameters of the wear model are quantitatively calculated.

Based on the obtained wear model, you can:

- calculate (predict) the wear of the unit under different conditions based on contact pressure and sliding speed, for example, at the stage of the design calculation of the friction unit;
- to optimize the structural and technological parameters of the friction unit according to the wear criterion.

The proposed calculation-experimental approach does not give one hundred percent compliance with the real flow of the process, but it is a necessary way to create calculation engineering methods for predicting the wear resistance of friction nodes of technical tribostems.

3. Implementation of the method of tribological testing of structural and lubricant materials according to the "ball-cylinder" scheme

To simulate the wear of a spherical indenter along a cylinder, a wear regularity (model) is proposed in the form of a dimensionless complex of determining factors (contact pressure and sliding speed):

$$\frac{du_W}{dS} = fC_W \left(\frac{\sigma}{HB}\right)^n \left(\frac{VR^*}{\nu}\right)^p, \tag{1}$$

where u_W is the ball wear;
S is the friction path;
f is the coefficient of friction in the ball-cylinder pair;
σ is the normal contact pressure;
HB is the hardness of the ball material;
V is the sliding speed;
R^* is the reduced radius of the sphere and cylinder;
ν is the kinematic viscosity of the oil;
C_W, n, p are the wear resistance parameters.

Let us take the form of the worn surface of the ball in the form of a circular sector with a profile radius a (Fig. 1).

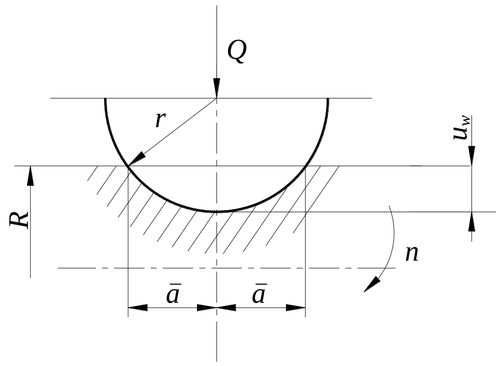

Fig. 1. Scheme of contact "cylinder-ball": Q is load; n is the speed of rotation; $\bar{a}$ is the half-width of the contact; R, r are the radius of the cylinder and balls

We also assume that the contact pressure under the wearing ball on the non-wearing surface of the cone is uniformly distributed:

$$\sigma = \frac{Q}{\pi a^2}. \tag{2}$$

In the case of ball-plane contact, the maximum wear of the ball is determined by the expression:

$$u_W = \frac{a^2}{2r}, \tag{3}$$

where r is the radius of the ball.

In the case of ball friction not along a plane, but along a cylindrical rotating surface, the dependence of wear on the contact radius will to some extent differ from that accepted for a plane. Let us estimate the magnitude of this deviation.

For friction on the cylinder, the contact area on the ball is curved and is performed along the radius of the cylinder R. In this case, with the same (with a plane) contact area, the wear value (3) is added to the wear value:

$$u_{W_1} = \frac{a^2}{2R}, \tag{4}$$

where R is the radius of the cylinder.

The relative error in determining wear from not taking into account the curvature of the cylinder, taking into account (3) and (4), is determined by the ratio:

$$\frac{u_W}{u_{W_1}} = \frac{r}{R}. \tag{5}$$

It follows that an error of 1% is provided for a ratio of radii of 1/100. Yes, at $r = 6.35$ mm the radius of the cylinder should be $R = 635$ mm, which does not seem correct for the design of the test facility.

It follows from the obtained estimate that the use of the formula for calculations when testing a ball-cylinder pair can lead to errors. Therefore, it is necessary to take into account in the calculations of the parameters of wear models the total wear over the entire contact area:

$$u_W^* = \frac{a^2}{2r_1} + \frac{a^2}{2R} = a^2\frac{r+R}{2Rr} = \frac{a^2}{R^*}.$$

where

$$R^* = \frac{Rr}{R+r}. \tag{6}$$

Thus, when testing for wear according to the ball-cylinder scheme, the dependencies for the ball-plane scheme can be used, if the ball radius is replaced by the reduced radius according to formula (6).

Let us represent the experimental dependence of the radius of a truncated sphere as a power approximating function:

$$a(S) = cS^{\beta}, \tag{7}$$

where c and β − approximation parameters, which are determined by the results of wear tests.

Integrating the wear pattern (1), we obtain the integral form of the ball wear model:

$$u_W(S) = fC_W\int_0^S\left(\frac{\sigma(S)}{HB}\right)^n\left(\frac{VR^*}{\nu}\right)^p dS. \tag{8}$$

Substituting the expression for wear into the left side of the obtained dependence $u_W = \frac{a^2}{2R^*}$, and to the right - the expression for the contact pressure (2), we get:

$$\frac{a^2(S)}{2R^*} = fC_W\int_0^S\left[\left(\frac{Q}{\pi\bar{a}^2(S)}\right)\frac{1}{HB}\right]^n\left(\frac{VR^*}{\nu}\right)^p dS, \tag{9}$$

or, taking into account (7), after integrating along the friction path, we obtain:

$$\frac{c^2 S^{2\beta}}{2R^*} = fC_W \left(\frac{Q_1}{c^2 \pi HB} \right)^n \left(\frac{VR^*}{\nu} \right)^p \frac{S^{1-2\beta n}}{1-2\beta n}. \tag{10}$$

From the condition of satisfiability of equation (10) for all values of S it follows:

$$2\beta = 1 - 2\beta n, \tag{11}$$

where:

$$n = \frac{1-2\beta}{2\beta}. \tag{12}$$

To define a parameter p tests are carried out at two values of the sliding speed V1 and V2, from which we obtain two groups of experimental data with approximating functions:

$$\begin{aligned} a_1 &= c_1 S^\beta; \\ a_2 &= c_2 S^\beta. \end{aligned} \tag{13}$$

Here we consider the problem of determining wear parameters based on the results of testing specimens with a changing contact area during wear. A change in the wear area causes a change in the values of the contact pressures. Exponent n expression (1) characterizes the rate of change of contact pressures during wear and is related to the index β experimental dependence (7), which characterizes the rate of change of the contact area during wear. Connection n and β in the accepted wear pattern (1) is uniquely described by relation (12). Since in the relations under consideration the sliding speed V does not depend on friction path S , then it does not affect the parameters n and β in the process of testing. In this case, changing the sliding speed V only affects the scale factor C_w in expression (1). The above reasoning is confirmed by the test results.

Substituting expressions (13) into (10) we obtain the system of equations:

$$\left. \begin{aligned} \frac{c_1^{\,2}\beta}{R^*} &= fC_W \left(\frac{Q}{c_1^{\,2} \pi HB} \right)^n \left(\frac{V_1 R^*}{\nu} \right)^p; \\ \frac{c_2^{\,2}\beta}{R^*} &= fC_W \left(\frac{Q}{c_2^{\,2} \pi HB} \right)^n \left(\frac{V_2 R^*}{\nu} \right)^p. \end{aligned} \right\} \tag{14}$$

Dividing the first equation by the second, after transformations we get:

$$(c_1 / c_2)^{2n+2} = (V_1 / V_2)^p. \tag{15}$$

Where:

$$p = (2n+2) \frac{\lg(c_1 / c_2)}{\lg(V_1 / V_2)}. \tag{16}$$

To determine the coefficient K_w we use one of the equations (14):

$$C_W = \frac{\beta c_1^{\,2n+2}}{fR^*} \left(\frac{\pi HB}{Q} \right)^n \left(\frac{\nu}{V_1 R^*} \right)^p. \tag{17}$$

Thus, formulas (12), (16) and (17) make it possible to determine the parameters of the wear pattern in the accepted form (1).

Methodology of experimental studies.

The tests of the bronze-steel pair were carried out according to the scheme of a spherical sample - a cylindrical counter-sample. A spherical surface with a radius of R = 6.35 mm was made at the end of a cylindrical specimen made of bronze with a diameter of 10 mm. The counterbody disk 96 mm in diameter and 12 mm thick was made of 40X steel, hardened and polished. The setup diagram is shown in fig. 2. The spherical sample 9, fixed in the body 8, is pressed against the counter-sample - disk 10 with the help of weights 7. The disk is lubricated by dipping in an oil bath 13. The disk is fixed on the shaft 4, which rotates in bearings 6 with the help of an electric motor 1 through a pulley 2 and a belt transmission 3.

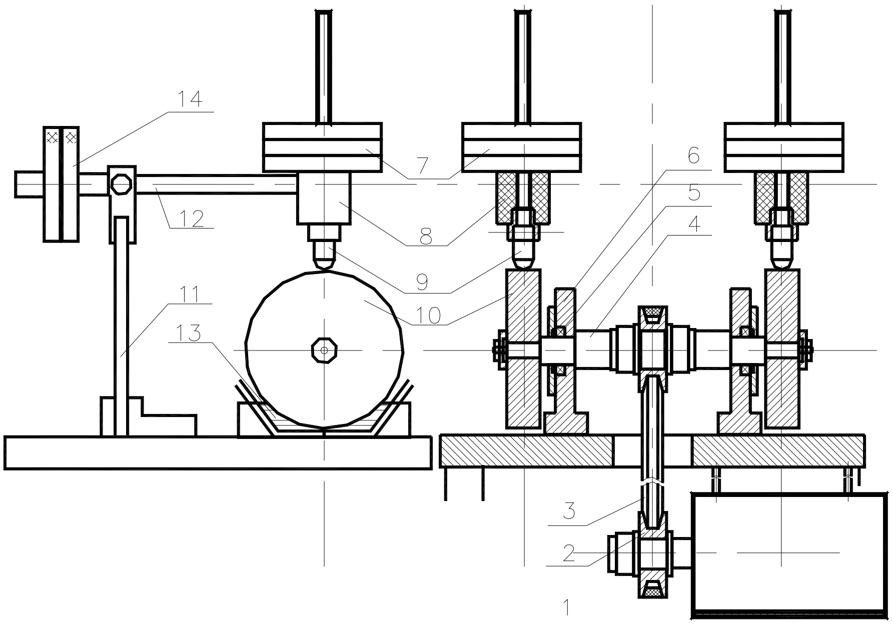

Fig. 2. Scheme of the ball-cylinder test setup: (1-motor, 2-pulley, 3-belt transmission, 4-shaft, 5-case, 6-bearing, 7-weights, 8-sample holder, 9-spherical sample , 10-counterbody (disk), 11-tension beam, 12-lever, 13-oil bath, 14-balance weight

The motor speed is adjustable from 0 to 1500 rpm. The load can be set up to 1 kg=10 N per sample. The installation provides for measuring the friction force using a strain gauge beam and lever 12.

The tests were carried out with the following initial data: the radius of the ball (sphere) of bronze - $r = 6{,}35$ mm; radius of counterbody, rotating disk $R = 48$ mm; sample load: $Q = 1$ H; the material of the ball is bronze, the disk is hardened steel; lubrication by dipping a rotating disk in a liquid lubricant; disk rotation speed 200 rpm, basic friction path of the ball surface 12 km = $12 \cdot 10^6$ mm. Lubrication was carried out with Magnum 15W–40 oil (TNK, Ukraine) with a viscosity of ν=40 mm2/s; friction coefficient f=0.08. Bronze hardness BrOF10-1 HB=90 MPa.

Reduced radius according to formula (6):

$$R^* = \frac{Rr}{R+r} = \frac{48 \cdot 6.35}{48 + 6.35} = 5.6 \text{ mm.}$$

When tested after a certain number of cycles, using a microscope, the diameter of the ball contact area is measured with an accuracy of microns. The test results are presented in table 1.

Table 1

Test results of a bronze sample during friction against a steel counterbody

No.	S, km	$a(V = 0.2 m/s)$, mm	$a(V = 0.4 m/s)$, mm
1	0.2	0.115	0.135
2	0.4	0.135	0.16
3	1.0	0.175	0.2
4	2.0	0.21	0.235
5	6.0	0.29	0.32
6	12.0	0.335	0.38

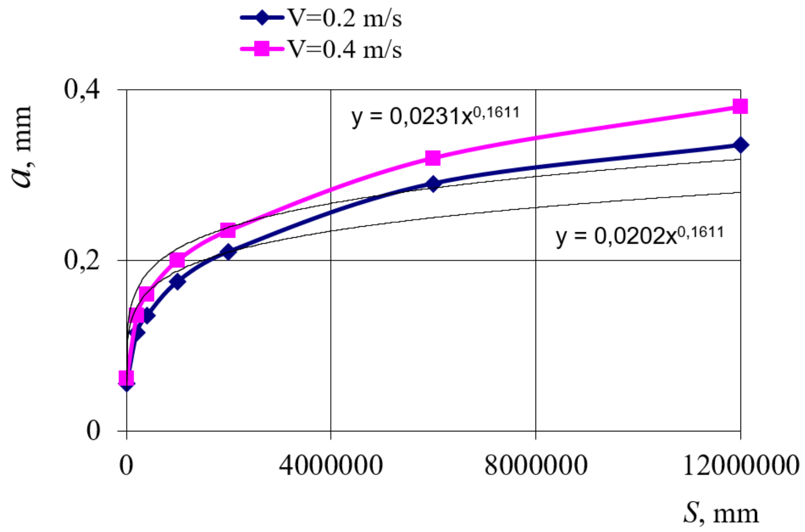

Fig. 2. Graphical interpretation of the test results and power-law approximation of the dependences of wear on the friction path a(S)/

On fig. 2, based on the test results, the dependence curves of the wear area dimensions on the friction path for two values of the sliding speed are plotted. The Excel program allows you to carry out a power-law approximation of the test results in the form (13) with a numerical determination of the parameters of this approximation, shown on the graphs in the form of equations of the trend line (9 approximating curves). The results of determining the parameters of these dependencies are shown in Table 2.

Table 2.

The results of the determination of the approximation parameters (13)

Options	V=0.2m/s	V=0.4m/s
c	0.0202	0.0231
β	0.1611	0.1611

In accordance with the procedure shown above, the parameter n can then be determined from formula (12):

$$n = \frac{1-2\beta}{2\beta} = \frac{1-2\cdot 0.1611}{2\cdot 0.1611} = 2.1.$$

According to formula (16), the parameter p is calculated:

$$p = (2n+2)\frac{\lg(c_1/c_2)}{\lg(V_1/V_2)} = (2\cdot 2.1+2)\frac{\lg(0.0202/0.0231)}{\lg(0.2/0.4)} = 1.19.$$

To calculate the coefficient C_w we use formula (17):

$$C_W = \frac{\beta c_1^{\,2n+2}}{fR^*}\left(\frac{\pi HB}{Q}\right)^n \left(\frac{\nu}{V_1 R^*}\right)^p = \frac{0.1611\cdot 0.0202^{2\cdot 2.1+2}}{0.08\cdot 5.6}\left(\frac{3.14\cdot 90}{1}\right)^{2.1}\left(\frac{40}{0.2\cdot 5.6}\right)^{1.19} = 0.001402$$

.

Thus, we have the following calculated values of the parameters in the wear pattern (1): $n = 2.1$; $p = 1.19$; $C_W = 0.001402$, which identify this dependence and make it possible to predict the wear intensity of tribocouples under given initial operating conditions: by loads, sliding speed, characteristics of lubricants and structural materials.

Conclusions

1. Based on the analysis of the current state of test methods for wear of friction pairs and the need for methods with certain operating conditions, the problem of developing a theory of test methods for wear of

friction pairs according to the ball-cylinder scheme was solved, which makes it possible to determine the parameters of wear models and the general characteristics of the wear resistance of materials.

2. Models (law) of wear processes are dependences of the intensity of wear (or wear) on the determining factors of the process, which are used in the design of friction nodes, forecasting their wear and to optimize structural, technological and other parameters.

3. Based on the solution of wear-contact problem for the "ball-cylinder" scheme, a theory has been developed for identifying the parameters of the wear pattern. To solve the inverse problem, a method of approximating function is proposed and implemented.

4. The theory of determining the parameters of wear models is developed on the basis of the solution of inverse wear-contact problems (that is, when the dependences for the calculation of parameters are determined according to the accepted mathematical form of the law of wear, geometric ratios (conditions of continuity in contact), equilibrium conditions.

5. Having a developed theoretical apparatus, tests of the friction unit are carried out in conditions close to real ones (in terms of materials, lubrication, temperature, etc.) and the parameters of the wear model are quantitatively calculated.

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Диха О.В., Старий А.Л., Гетьман М.М., Фасоля В.О. Теорія та експеримент методів трибологічних випробувань

На основі аналізу сучасного стану методів випробувань на знос пар тертя та потреби в методах з певними умовами роботи вирішувалося завдання розробки теорії методів випробувань на знос пар тертя за схемою шар-циліндр, що дозволяють визначати параметри моделей зношування та загальну характеристику зносостійкості матеріалів. Показано, що вид зношування в межах розкиду властивостей параметрів пари тертя забезпечує дотримання основних умов контакту: матеріали, мастило, тиск, швидкість, температура, вид руху та інші менш істотні фактори. Встановлено, що одним із важливих питань при розробці та проведенні випробувань на знос є облік впливу стаціонарності режиму зношування у точці. Методики випробувань в режимах (режим перехідних процесів), що встановився і невстановлений, різні. В основі методу випробувань має лежати вирішення контактного завдання для пари тертя зразок-контрзразок. На основі рішення зворотної зносоконтактної задачі для схеми «куля-циліндр» розроблена теорія ідентефікації парметрів закономірності зношування. Для вирішення зворотного завдання запропоновано та реалізовано спосіб апроксимуючої функції. Ступенева апроксимація експериментальної функції після підстановки в вирішальне рівняння дає прості вирази для розрахунку параметрів моделі. Отримані результати дають можливість прогнозування інтенсивності зношування трибосполучень за заданих вихідних умов експлуатації: за навантаженнями, швидкістю ковзання, характеристиками мастильних і конструкційних матеріалів.

Ключьові слова: знос, пари тертя, трибологічні випробування, контактна задача